ENGINEERING ABSTRACTS

Section 3. Shipbuilding and Marine Engineering

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Steam Air Heaters for Marine Boilers

In this paper various difficulties experienced with gas air heaters are commented on and a case is made out for the use of a steam air preheater in conjunction with an economizer. The practical results are recorded of three years' use of steam air preheaters fitted in five ships. In conclusion, suggestions are made for the further development of steam air preheaters in conjunction with a split feed system.—*Paper by W. J. S. Glass, read at a meeting of the Institute of Marine Engineers, 11th November 1952.*

Ocean Going Tug

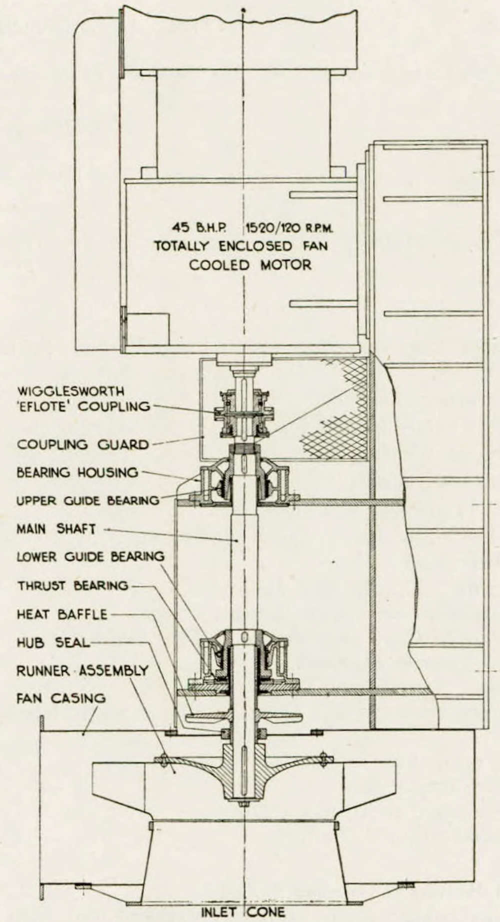
The ocean going tug *Du Guesclin,* built by the De Biesbosch Shipyard and Engineering Company, Dordrecht, for the Union des Remorques de l'ocean, Paris, and based at St. Nazaire, is classed with Bureau Veritas and is capable of performing harbour as well as deep sea service. She is also equipped with salvage and fire extinguishing apparatus and can provide 60 tons of fresh water to other craft. The principal particulars are: length overall 119ft. 8in., length b.p. 109ft. 10in., breadth moulded 25ft. 7in., depth 14ft. 11in., gross tonnage 310 tons, net tonnage 63 tons, and service speed 12 knots. Life saving appliances include two lifeboats (one with motor) slung under Welin Maclachlan davits. Steering gear is of the electro-hydraulic type. There is also an electrically driven capstan which develops a pull of 3 tons. It is of interest to note that the capstan motor is installed in the engine room. Navigational appliances include radar, direction finding apparatus and radio. The propelling machinery consists of a Crossley Diesel engine type CRL 8, having eight cylinders and developing 1,500 b.h.p. at 320 r.p.m. On account of its low height and centre of gravity, the engine is especially suited for this type of tug. In normal service, the engine develops 1,200 b.h.p. at 259 r.p.m. and on trials with the engine running at 90 r.p.m., a speed of 3·7 knots was attained. On full power a speed of 12 knots was recorded in a depth of water of 36 feet.—*Shipbuilding and Shipping Record, Vol. 80, 23rd October 1952; pp. 540-542.*

Main Engine Seatings and Bedplate under Static Bending Tests

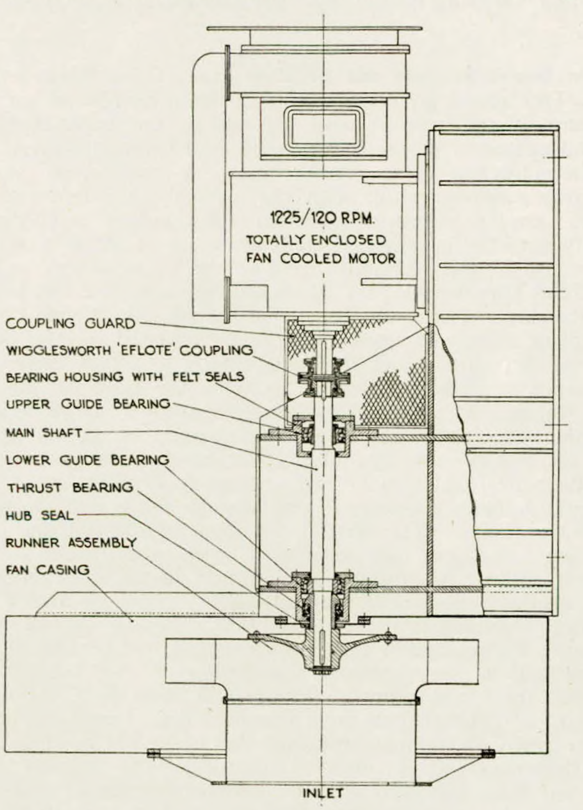
This paper gives the results of an investigation into the flexure of the bedplate and seatings of the triple-expansion steam engine of a standard type of 10,000-ton dry-cargo ship while subject to static bending tests. A brief review is made of publications dealing with the subject. No reference has been found to either experimental investigations or theoretical analyses of the problem. Details are given of the instrumentation and testing technique used in the investigation. Gauge readings were recorded for eleven different conditions of loading of the ship. The results show that the vertical deflexion of the bedplate is approximately twice that calculated for the flexure of the ship girder over the same length, and that there is no evidence of sliding movement between the bedplate base and the seating. It is shown that the comparatively large deflexion of the bedplate is due to the local changes of section in way of the engine seating. This effect is demonstrated by the mathematical solution of an analogous problem, that of a composite beam subjected to end loading and a uniform lateral elastic restraint. The vertical deflexions of the bedplate under service conditions are estimated from the results of the investigation. Assuming true alignment in the light condition, the central deflexion of the bedplate in a typical loaded condition is estimated to be 10·0/1,000 inch sag in still water. During the passage of the ship through a standard L/20 wave there will be superimposed a deflexion of $\pm 13{\cdot}7/1{,}000$ inch so that the actual central deflexions will be from 3·7/1,000 inch hog to 23·7/1,000 inch sag. Provided that there is no uneven wear-down of the main bearings the additional bending stress in the crankshaft is small. Deflexion of the bedplate causes uneven load distribution and consequent unequal wear-down between the main bearings however, and it is probable that this is the most serious effect of the flexure, resulting in high crankshaft bending stresses.—*Paper by A. J. Johnson and J. E. Richards, read at a meeting of the North-East Coast Institution of Engineers and Shipbuilders, 14th November 1952.*

Vertical Spindle Boiler Room Fans

The upper regions of the boiler casing in a modern steamship are frequently congested, particularly when high-efficiency watertube boilers, having superheaters, economizers and air preheaters are installed. Room must be found somewhere in this locality for the forced and induced draught fans and the necessity to provide crossbeams or brackets to support them is often an embarrassment to the designer. The owners of a class of advanced steamships now building at Dundee have adopted a new approach to the problem. Here the forced and induced draught-fans are of vertical-spindle pattern which can be secured to the boiler room or engineroom casing and require the minimum of trunking. The two forced draught fans to be installed in each vessel have a designed rating of 10,500 cu. ft. of air per minute (at 100 deg. F.) against 8in. S.W.G. The impellers have backwards-curving blades which provide good non-overloading characteristics. They are driven by 19/1·8 h.p. Allen totally-enclosed fan-cooled motors and the running speed is variable by series and shunt field regulation between 1,225 and 120 r.p.m., the latter figure being used when lighting up the boilers from cold. The impellers and casing are made of zinc-sprayed mild steel. Power is transmitted through a flexible coupling and the shaft is carried in ball and roller bearings. These fans are secured against the side of the engine casing at boat deck level, where they draw warm air from the upper regions of the engine room. The discharge trunking leads it through air preheaters before it passes to the two Foster Wheeler main boilers. The induced-draught fans are of similar design and have a capacity of 14,000 cu. ft. of air per minute (at 280 deg. F.) against 12·5 S.W.G. This output absorbs 41 h.p. and the fans are driven by 45 b.h.p. 1,520/600/120 r.p.m. totally-enclosed fan-cooled motors. In this case the impeller



Arrangement drawing of induced draught fan



A 10,500 cu. ft. per min. forced draught fan. The motor is rated at 19/1·8 h.p.

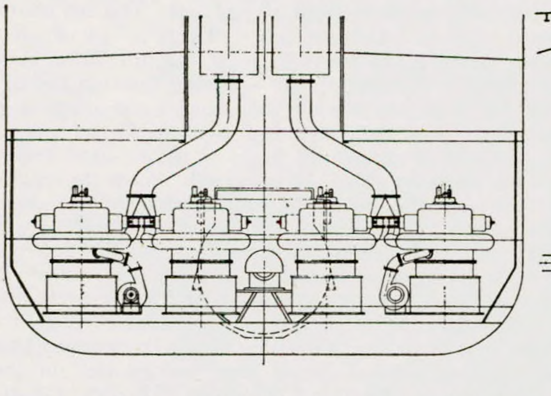
has "self-cleaning" curved radial blades. Both impeller and casings are of aluminized mild steel. Comparison of the illustrated drawings will show that the induced draught fans differ in having water-cooled Michell shaft bearings and cooling discs as it will be appreciated that the uptake gases may reach 300 deg. F. These fans are bolted to the forward bulkhead of the boiler casing at almost the level of the casing top. The inlet cones are immediately above the gas outlets from the hot passes of the air preheaters; the ducting is therefore desirably short and simple.—*The Marine Engineer and Naval Architect, Vol. 75, September 1952; pp. 411–413.*

Real Fuel Consumption of Ships' Machinery

It is generally accepted that in any comparisons between the fuel consumptions of marine Diesel machinery and geared turbines of, say, between 7,000 b.h.p. and 10,000 b.h.p. per engine, the steam machinery consumes 50 per cent more fuel than the Diesel plant. The figure is based upon the employment of the most efficient turbines, with a boiler pressure of 450 to 500lb. per sq. in. and steam temperature of 750 deg. F. to 800 deg. F. These conditions appear to represent the maximum attainable commercial efficiency of steam machinery at sea in service, and in a recent report of Pametrada it was stated that higher boiler pressures and steam temperatures do not lead to any improved economy when capital cost and higher maintenance charges are taken into consideration. This comparison of fuel consumptions is based upon average results *on trials,* which are usually taken as being from 0·37lb. to 0·39lb. per s.h.p. hr. for Diesel engines and 0·55lb. to 0·58lb. per s.h.p. hr. for steam plant, for all purposes. The quoted results, however, have little practical relevance. What is important, and in fact, what alone matters to the shipowner, is the average fuel consumption over the twenty-five years of the life of the ship. The trial trip results are determined by full-power tests. A very large proportion of the ships at sea operate at about 80 per cent of full output continuously as an average figure over the years. It is at this proportional load that the Diesel machinery is at its highest efficiency and the specific consumption at 80 per cent is always somewhat lower than at 100 per cent. With steam plant, however, the efficiency falls substantially when the output is reduced to 80 per cent and the extent of this reduction is not by any means fully realized. Secondly, the consumption of all steam plant increases with age and an average figure of 1 per cent per annum has been given as representing the generally accepted reduction in efficiency. Combine these two figures and we may expect that, over the life of a geared turbine ship built today, with machinery of the highest efficiency, the fuel consumption for all purposes will range from 0·61lb. to 0·64lb. per b.h.p. hr. or more. This was borne out in some figures published relating to the fuel consumptions of two P. and O sister cargo liners, one with Diesel and the other with steam machinery. Whereas the trial trip figures had been 0·377lb. per s.h.p. hr. and 0.5815lb. per s.h.p. hr. respectively, after two years' service the corresponding consumption had been assessed at 0·363lb. and 0·62lb. per s.h.p. respectively.—*The Motor Ship, Vol. 33, December 1952; p. 336.*

Multi Radial-engined Plant

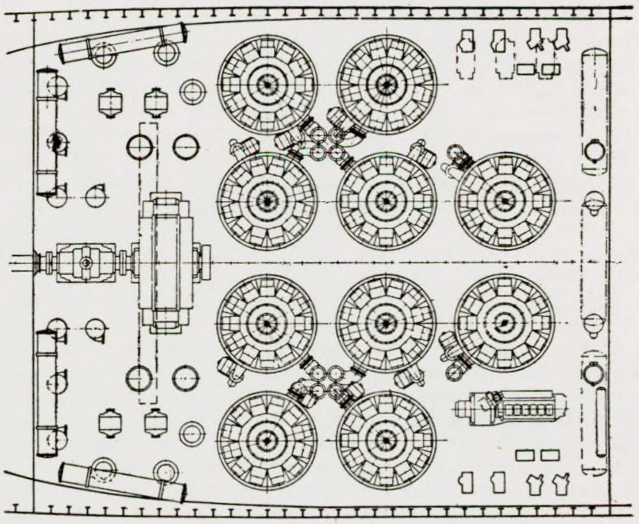
The Nordberg Manufacturing Company, of Milwaukee, have evolved an interesting proposal for a 16,000 s.h.p. single-screw Diesel electric vessel which will occupy very little head room and yet provide ample facilities for maintenance. The Nordberg 12-cylinder radial engine put forward in the scheme is of the two-cycle loop-scavenged type with a bore of 14 inches, a stroke of 16 inches, and a maximum rating of 2,125 b.h.p. at 400 r.p.m. Large numbers of this type have been built for stationary duties, either in straight Diesel form, as gas engines, or as dual fuel units. In one works alone of the Aluminium Company of America at Port Levaca, Texas, there are no less than 194 of these units. Each Diesel-alternator set consists of a 12-cylinder radial engine with governor and control gear superimposed on the alternator. In the sketch seen above, eight of these units provide propulsion power; one is for auxiliary power and the remaining unit may be used for either duty. It is claimed that the flexibility of arrangement thus makes engines of this type particularly adaptable to vessels having their machinery spaces aft.—*The Marine Engineer and Naval Architect, Vol. 75, November 1952; p. 495.*



Twelve-cylinder radial Diesel-electric installation

Turbo-charged Two-stroke Diesel Plant

The influence which the application of turbo-charging to two-stroke Diesel engines will have on the design of such machinery is likely to be greater than any other improvement in the past quarter of a century. Hence the completion of the trials of the first ship to be propelled by a large two-stroke exhaust gas turbo-charged unit is an event of the first importance. The vessel in question is the 17,000 ton tanker *Dorthe Maersk* (length b.p. 496 feet, beam 68ft. 3in. and draught 29ft. 6in.), built at the Odense Steel shipyard for A. P. Möller. In it is installed a six-cylinder Burmeister and Wain turbo-charged engine of the poppet-valve class with a cylinder diameter of 740 mm. and a piston stroke of 1,600 mm. These are the dimensions of a number of standard unsupercharged B. and W. six-cylinder engines with a normal output of 5,560 b.h.p. at 115 r.p.m. When turbo-charged, the engine is designed for a continuous output of 8,460 i.h.p. (m.i.p. about 8·0 kg. per sq. cm.), corresponding to 7,500 b.h.p., which is 35 per cent greater than the rating of the corresponding unsupercharged engine. The weight is 25 per cent less, whilst the engine is 20 per cent shorter and the fuel consumption 8 gr. per b.h.p./hr. lower (150 gr. per b.h.p./hr. or 0·33lb.). Both the hull and the engine were well under construction before it was decided to employ turbo-charging, and the hull design is not suitable for utilizing the full power economically, so that in this ship the output will be limited to 7,300 i.h.p. or 6,530 b.h.p. at 108 r.p.m., the corresponding m.i.p. being 7·45 kg. per sq. cm. The full rated output of this six-cylinder engine (7,500 b.h.p. at 115 r.p.m.) is slightly higher than the normal output of a non-turbo-charged eight cylinder engine of the same dimensions, this being 9,200 i.h.p. (m.i.p. = 6·5 kg. per sq. cm.), corresponding to about 7,380 b.h.p.—*The Motor Ship, Vol. 33, November 1952; p. 298.*



Multiple-engine Propulsion Units

An interesting and highly economical arrangement of marine power has recently been demonstrated by R. A. Lister (Marine Sales), Ltd. This power arrangement comprises four of the new Lister Freedom six-cylinder marine engines, each of which has an individual remote-control clutch, driving a single propeller shaft through S.L.M. oil-operated reverse-reduction gearing (designed and manufactured by Modern Wheel Drive, Ltd.), and giving a total of 216 b.h.p. at 1,800 r.p.m. This four-engine drive arrangement, in addition to saving space, increases the availability of the craft in which it is fitted, because of greater reliability. Each engine may be isolated from the transmission—an arrangement which ensures substantial economy in fuel, lubricating oil and general wear and tear, when only reduced power is required. In addition, in the event of breakdown of any of the main engines, the damaged unit may be disengaged, so that the vessel is able to proceed with but moderate loss of speed. The speed reduction is, in fact, only in the region of 10 per cent. Another important consideration is weight. Compared with single-engine units of the slow-speed type, multiple engines, such as the four Freedom six-cylinder units, save 2 or 3 tons for their power. This figure is based on the combined weight of fuel and engine for a range of 1,000 miles. The manufacturers claim that the price is considerably less than that for a single engine of the same horsepower—approximately 10 to 15 per cent for the total cost of the vessel. An advantage also associated with the multiple-engine drive is that the cost of spares is reduced by at least 25 per cent as compared with slow-speed engines. In a fleet of craft with multiple-engine drive, a spare engine could be carried on board, or stored ashore and exchanged, when required. Moreover, an auxiliary of the Freedom-engine type could be used, thus providing 100 per cent standardization of engine-room spares in the individual vessel.—*The Shipbuilder and Marine Engine-Builder, Vol. 59, December 1952; p. 705.*

New Lightweight Marine Diesel

The power-to-weight ratio of the new Packard marine Diesel approaches five pounds to one b.h.p. developed. Output ratings range from 300 to 800 h.p. in 6, 8, 12 and 16-cylinder magnetic and non-magnetic models, all 4-cycle, 5⅜-inch bore and 6½-inch stroke. All models have identical cylinder displacements of 142 cu. in. Production is planned later for the 8-cylinder and 60 deg. V-16, similar in design to the 6-cylinder in-line and V-12 presently being delivered. The six and V-12 are equipped with turbochargers and their continuous output is 300 and 600 h.p. respectively, at 2,000 r.p.m. The 6-cylinder weighs 2,200lb. and the V-12, 3,250lb. or 7·34lb./b.h.p. and 5·41lb./b.h.p. in that order. Aluminium alloys are used for major components in every rating. The V-12 engine block measures 30 inches across the top, 27½ inches high and almost 4 feet in length, making it the largest aluminium casting ever made in production quantities. Other vital parts of aluminium are pistons, crankcase, oilpan, intake and exhaust manifolds, valve covers, timing gear case, accessory housings, turbochargers, and smaller items. The cylinder head is made of high resisting alloy steel, and defies conventional machining. Precision cast, it contains intake and exhaust ports and valve nests. The fuel injection pumps for both the 6-cylinder and V-12 engines are of multi-plunger constant stroke design. Fuel pumps of the 6-cylinder apply to the V-11, one for the six and two for the V-12. Fuel is governed by an externally mounted hydraulic governor. An over-speed trip governor is an added safety measure. Each cylinder head is fitted with two intake and exhaust valves, driven by individual camshafts—two per bank. Valve seats and faces are hard-surfaced with stellite. Exhaust valves are sodium cooled. The nested valve springs are of damping type, caged in cylindrical barrels integral with the valve guides. Barrel type tappets fit over the springs and slide within the valve guide barrel. Valve clearance adjustment is by use of a button fitting over the valve tip and touching the inner face of the tappet. Turbochargers are mounted at the front engine end and fitted with air cleaners. Heat radiation is reduced by a water jacketed exhaust manifold which directs gases to the turbine end of the charger. A special feature of the manifold is a stainless steel tube whose diameter is such that a dead air space exists between the inner manifold surface and the tube. This design compensates for heat losses in the exhaust gas stream.—*Motorship, Vol. 33, October 1952; pp. 28-32, 38.*

Need for Improved Engine Air Filters

A recent report on air filters for internal combustion engines contains a number of interesting comments on various existing types and challenges designers to produce more satisfactory filters of radically new design. The art of designing air filters is claimed to have lagged, and existing types are claimed to flaunt far too many engineering compromises. The oil-wetted type is considered to be almost a menace. Although its initial pressure drop is low, pressure rise and efficiency fall-off are too rapid. The latter are assumed to be kept within tolerable limits by proper cleaning, but this assumption is claimed to be dangerous since it can lead to rapid engine wear from airborne impurities. The oil bath type, although it has the fewest compromises, has too high an initial pressure drop, it is stated. It weighs too much and it costs too much. The centrifugal separator or cyclone type is criticized in view of its excessively high initial pressure drop and the wide variations in its efficiency with the engine intake air flow. The strainer type of filter is considered to be the most efficient type, "but proper cleaning is almost impossible". The initial pressure drop is low, while the pressure rise is rapid, due to efficiency. Also, it takes too much space and its cost rises with its efficiency. Electrostatic filters are claimed to be too expensive, too big, and too difficult to service. They have too many parts to give trouble-free service and they have to be cleaned too often. Discussing future developments, the author of the report points to the possible development of self-charging electrostatic plastics with possible automatic washing, high-efficiency centrifuging with external energy source, or ultrasonic agglomeration of particles.—*Engineers' Digest, Vol. 13, November 1952; p. 372.*

Performance of First Werkspoor-Lugt Engine

The first Werkspoor-Lugt engine, a normally aspirated four-cylinder two-stroke, single-acting unit with cylinders 600 mm. in diameter and a piston stroke of 900 mm., the continuous service output being 1,800 b.h.p. at 162 r.p.m., is installed in the Great Lakes-type ship *Prins Frederik Willem,* owned by the Oranje Line. The fuel consumption for the main engines, running at 155-160 r.p.m., and for the two 88-kW. generators, each driven at 550 r.p.m. by a Werkspoor four-stroke engine, totals 6·3 tons per 24 hours, while the cylinder lubricating oil consumption for the same period is 40 litres, or 0·83 grams per b.h.p.-hr. No oil is reported to have been expended in the crankcase. During the first few trips, tests were made with varying classes of residual fuels, ranging in viscosities at 100 deg. F. from 627 secs. to 2,634 secs. Redwood No. 1, and eventually the latter fuel was adopted, this having a Redwood viscosity of 150 secs. when heated to 105 deg. C. before entering the main engine fuel pumps. The sulphur content varied from 1·8 to 2·2 per cent and the Conradson carbon residues were about 10 per cent. The ash content ranged from 0·085 to 0·117 per cent. When at sea or on the Great Lakes, heavy oil is normally used, but for those periods when the engine is running slowly—passing through canals and locks —or for long standby periods such as waiting at the locks, the engine has been switched over to Diesel oil, to obviate the necessity of generating steam in the oil-fired boiler to maintain the temperature of the heavy oil. With the engine running on full load, the steam for fuel heating is provided from the exhaust-gas boiler. After the first few months' operation on heavy oil, and following occasional burning of the exhaust valve seats, it was found that there was some deposit on the valve seats, on the mitre faces and lower parts of the stems and that this, partly broken off, appeared to consist of almost 70 per cent of sodium sulphate. Since the source of this deposit was obvious, extensive tests were carried out on the existing purifier and clarifier (De Laval type 1729 deg. C.) at temperatures varying from 220 deg. F. to 160 deg. F., and with flow capacities ranging from 500 to 1,000 kg. per hour—about two to four times the hourly engine consumption. As a result of these tests, and following consultation with representatives of De Laval, a larger size of purifier and clarifier was installed—type V1B 1929 C—this resulting in lower ash content values and having the effect of decreasing by a considerable extent the amount of deposit on the valves. Furthermore, special treatment and the use of better heat-resisting material for the valves, also arrangements for improving the temperature conditions of the valves, have combined to overcome these initial difficulties.—*The Motor Ship, Vol. 33, December 1952; p. 344.*

Single-acting Two-stroke Diesel Engine

The KZ 68/120 type of single-acting two-stroke Diesel engine built by M.A.N. covers a range of output between 2,000 and 4,400 b.h.p. with 5 to 8 cylinders. This new standard type has a cylinder output of 500 to 550 b.h.p. at a speed of 115 to 127 r.p.m. The mean indicated pressure of 5·5 atm. and the maximum piston speed of 5 m. per sec. are intentionally low, thus warranting an optimum of reliability even under the most strenuous conditions of continuous service. The general design has been altered but slightly, compared to the KZ 65/120 design. The main features are the following: Plain and clear design, robust cast iron construction with easy accessibility to all parts of the running gear. The entire engine frame is relieved of all combustion pressures by initially tightened tie-rods, which extend from the bedplate to the top of the cylinder block. Direct injection by individual fuel pumps, arranged in two groups above a short camshaft. The fuel valves are provided with cooled nozzles. The scavenging air is supplied by a scavenging air pump attached to the front end of the engine. The cylinders are cooled either by fresh or by salt water, whereas the pistons are mostly cooled by fresh water supplied to the pistons through telescopic pipes. The weight of the engine itself, i.e. without outer accessories such as thrust block, starting air bottles, exhaust silencer, re-cooler, etc., with a cylinder output of 550 b.h.p., is about 75 kg. per b.h.p. The specific fuel consumption amounts to 160 to 163 g./b.h.p.h., based on a suitable fuel with a lower calorific value of 10,000 Kcal./kg. at an ignition pressure of about 45 atm.—*M.A.N. Diesel Engine News, No. 26, June 1952; pp. 2-3.*

Investigation of Corrosive Nature of Flue Gases

One of the most serious problems encountered in water-tube boilers in recent years has been the corrosion of the air heater and other low-temperature surfaces by sulphuric acid, formed by the combination with water vapour of traces of sulphur trioxide produced in the boiler system and condensed out on surfaces at or below the dewpoint. The temperature-controlled mild-steel probe described in this paper was developed with the object of providing more direct information as to the mechanism of acid deposition and attack on metal surfaces. The function of the probe is to obtain a relative measure of the corrosive nature of the flue gases from a boiler rather than, at this stage, to simulate the rate of corrosion obtained on any particular boiler surface. The instrument consists essentially of a prepared metal test surface placed at the end of a long probe. This surface is inserted into the gas stream to be examined and is maintained at a constant temperature, irrespective of fluctuation in the gas temperature. After a suitable period of insertion the probe is removed and the amount of corrosion of the test surface that has occurred ascertained. Fig. 1 shows the design of the test head. The 1-inch diameter hemispherical mild steel test cap (A) fits into the heat-insulating collar (B), the lip (C) preventing air blowing directly over the external surface. A chromel and an alumel wire are welded into the cap about ½ inch apart and form the thermocouple used for measuring and controlling the temperature of the head. The unit is mounted on the end of a metal probe 6 feet long. Compressed air used for cooling passes up the centre of this probe through an inner tube (D) to the distributor. After blowing against the under-side of the cap, it discharges to atmosphere down the annulus between the tube and probe wall, so helping to keep down the metal temperature of the probe. The thermocouple in the cap is connected to a "Bristol" potentiometer pyromaster recorder. This pyromaster not only provides a permanent record of the temperature but also maintains it at any selected value by making or breaking (depending on whether the recorded temperature is above or below a pre-selected datum) two relays which operate the two circuits controlling the direction of rotation of a 24v. single-phase a.c. motor. This motor opens or closes a valve in the compressed-air line through a high-reduction gear.—*G. G. Thurlow, Journal of the Institute of Fuel, Vol. 25, November 1952; pp. 252-255; 260.*

Condensation of Steam

Experiments have been performed to examine critically the mechanism of condensation of steam, particularly when the condensation occurred in the dropwise form. This led to an investigation of the duration of life of a surface giving purely dropwise condensation, of which a brief summary is given. Having found that appreciable durations were possible, quantitative measurements of the heat transfer rates were made for high heat loads on a flat surface. Stress was laid throughout on the maintenance of either the purely filmwise or purely dropwise form. An analysis of the results is given, and an examination of the possible application to practical problems shows that steam-condenser surface areas for high heat loading can be reduced appreciably from the present value, provided little non-condensable gas is present, the presence of a small quantity of promoter is tolerable, and suitable waterside conditions can be maintained. The effects of departure from these conditions, found in practice, are discussed. It is not suggested that dropwise condensation can be attained in all apparatus using vapour condensation, but it is anticipated that the development of synthetic resins, which can be applied in very thin adhesive films, or other promoters, which are not objectionable in very small quantities in boiler feed, may lead to its wider application. A British patent, for example, claims that zirconium will give dropwise condensation without a promoter; although, with the treatment described above, this metal was found to give filmwise condensation for many hours, it did very readily give dropwise condensation without elaborate preparation when touched with oleic acid. Developments such as these justify anticipation that the application of dropwise condensation may be realizable in practice, and with this expectation, further research is continuing.—*Paper by H. Hampson and N. Ozisik, submitted to The Institution of Mechanical Engineers for written discussion, 1952.*

New Motor Coaster

The motor coaster *Dryburgh* was recently handed over to her owners, Geo. Gibson and Co., Ltd., and James Rankine and Sons, Ltd., Leith. Unusual features of the vessel are that she is equipped with hydraulic deck machinery, and that she is supplied with electric power on the alternating current system. She is also the first vessel to be equipped with a Carron-Calor gas cooking range in the galley. Designed for the transport of fruit and general cargo between Holland and the west coast of Scotland, the *Dryburgh* is fitted with forced ventilation in the holds. She is a shelterdecker with a hold capacity of 101,750 cu. ft. grain and 95,000 cu. ft. bale, the hatches being equipped with MacGregor patent steel covers. Six two-berth cabins for passengers are provided amidships. The main propelling machinery comprises a British Polar

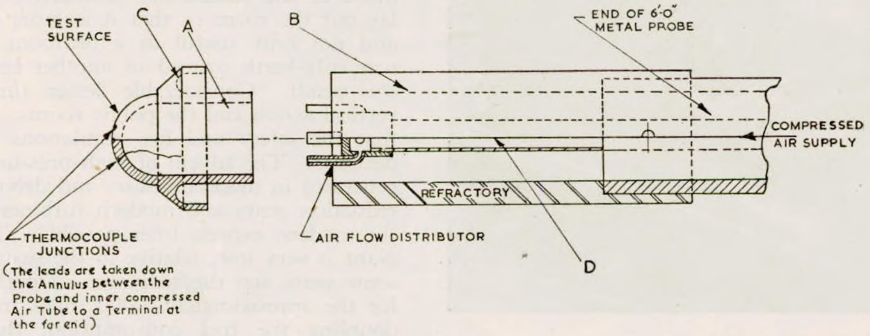
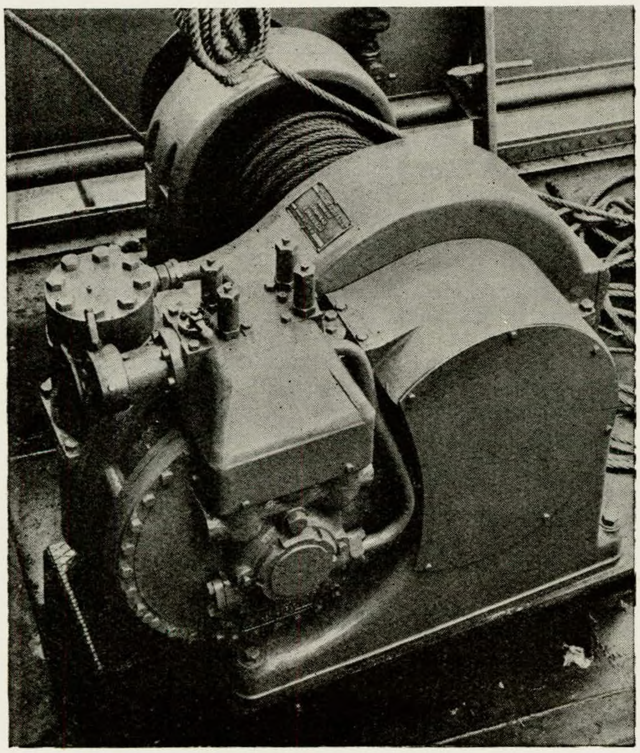


FIG. 1

8-cylinder Diesel engine developing 1,250 b.h.p. at 300 r.p.m. The principal particulars of the *Dryburgh* are as follows:—

Length overall	260ft.
Length b.p.	245ft.
Breadth moulded	38ft.
Depth moulded to shelter deck	22ft. 2in.
Depth moulded to upper deck	14ft. 8in.
Gross tonnage	1,080 tons
Deadweight (about)	1,380 tons at 14ft. draught
Speed loaded...	13·5 knots

The hydraulic power system comprises the central pumping unit, steering gear, windlass, capstan and winches, all of which were supplied by MacTaggart Scott and Co., Ltd. The main hydraulic power unit is centrally situated in the engine room close to the main control position. It consists of three hydraulic pumps driven by 18 h.p. A.C. electric motors; oil tank; steering gear accumulator; isolating valves serving the various lines; and gauge panel. The system is equipped with three MacTaggart Scott 18 h.p. multi-cylinder radial pumps situated in the engine room, vertically driven by constant speed A.C. motors. Any two of the pumps are capable of supplying the necessary power for practically any operating condition of the ship; the third pump is regarded as a standby. An automatic loading valve is fitted to each half of each pump, and controls the pump output as the pressure rises and falls. Thus each half pump is controlled to cut in consecutively, so avoiding any sudden large increases in the electric current demand. The loading valves can also be hand operated, pre-selecting the amount of pumping capacity, which then functions automatically. For instance, any one of the six half pumps can be selected to supply the steering gear when, as at sea, this is the only unit operating. The loading valves are also used when starting up a pump light. The various isolating valves are conveniently grouped so that the most suitable combination of pumps and accumulators may be selected to serve whatever deck machinery is in operation. That this selection can be made and be put into operation without the supervisor leaving the engine room makes for convenient working. Six 3-ton cargo winches are fitted in three groups of two. Each is driven by a six-cylinder radial patent selective type hydraulic motor, selection being automatically determined by the load being operated. They are capable of raising 3-ton loads at 100ft. per minute, $1\frac{1}{2}$ tons at 200ft. per minute, or a light hook at 300ft. per minute. The controls of these winches are entirely foolproof, and these speeds cannot be exceeded in any circumstances. Neither can the winch be changed from "hoist" to "lower" unless it is first stopped, thus saving much wear and tear on the cables due to shock loading. A safety or holding brake of the band type is fitted on the drum shaft and comes on automatically should the working pressure drop below a safe working value. The Calor gas installation in the galley is claimed to be the first installation of its kind on any vessel. The gas is stored at 22lb. per sq. in. in eight cylinders, each containing 83lb. of liquid gas (540 cu. ft.). These cylinders are stored in a separate compartment adjacent to the galley. Four of them are in use at a time and are expected to last for about three weeks. The gas range is of a similar pattern to the usual domestic gas cooker, except for its larger size and more robust construction to withstand use at sea.—*The Shipping World, Vol. 127, 5th November 1952; pp. 367-369.*



The 3-ton hydraulic winch

Operating Problems with Transatlantic Liners

The life of a large passenger vessel, whether it be a super-liner like the S.S. *United States* or a small passenger vessel, or in fact any ship, begins when the idea to build a ship is first conceived and divides itself into three phases as follows: (1) The design period. (2) The development of design and building period. (3) The operating period. One of the first problems facing the operator is the selection of an inspector engineer to follow through during the first two phases to ensure efficient and workable arrangements with the maintenance problem continually in mind. A second operating problem is the choice of deck and engineering officers, staff officers, and unlicensed personnel of proper qualifications with the necessary practical and theoretical experience. This ship organization fits very naturally into the operating organization ashore. The terminal operations department, which includes all pier facilities, the receiving and delivery of cargo, and the discharging and loading, including the handling of all baggage facilities connected with the embarkation and disembarkation of passengers, falls under the direct responsibility of the terminal manager, who in turn comes under the general manager. Another important point which must be considered as the ship is being built, particularly with a superliner, is a study of the port and berthing facilities at each port to which the ship will run. Although she may be expected to call at many ports of the world, her regular service, as in the case of the *United States,* is between New York, Le Havre and Southampton. If this matter is taken under consideration in the early stages, changes in the ship may be made to make her more readily fit the ports at which she will call, and at the same time changes in the ports' facilities so that they may fit the ship. The arrangement problems in the modern, safe, fireproof ship are not greatly different insofar as the general ship or stateroom arrangements are concerned. The situation today is eased somewhat by the passenger preference for showers instead of tubs, which permits a higher percentage of rooms with private facilities and enables better layouts to be made of the staterooms themselves. The attempt is made to lay out the room so that it is more suitable as a sitting room and not only useful as a bedroom. Certain rooms use the new sofa-berth instead of another bed in order to accomplish this result. Considerable design time must be spent on the vertical access and the public rooms. It is chiefly in this realm that the safety and fire regulations tax the ingenuity of the designer. The advent of high-pressure, high-temperature steam generated in modern boilers and driving the propellers through reduction gears and modern turbines has done much to make the modern express liner possible. The fuel bill with such a plant is very low, relative to the sustained sea speed. Whereas some years ago the reduction of perhaps one day in the time for the approximately 3,000 miles transatlantic passage meant doubling the fuel consumption, the present-day machinery permits this to be done for no more fuel than was formerly expended. This economical machinery is of further value in that it also permits a greatly increased radius of action at a higher speed, which is most esssential in a vessel which will be used for naval purposes, should the occasion arise. One of the essential features of today's transatlantic liner that was not included in the designs of other years is the complete air conditioning of the ship.—*Paper by N. T. Lawrence, Jr., read at a meeting of the Society of Naval Architects and Marine Engineers, 17th September 1952. Abstracted in S.N.A.M.E. Bulletin, Vol. 7, October 1952; p. 33.*

Design of Transatlantic Liners

In the S.S. *United States,* advantage has been taken of the progress in welding and the development of new materials. The latest improvements in steam machinery—higher temperatures and pressures—have been incorporated and it has the most modern electrical installation. In addition, the service experience gained during World War II and the years just prior thereto has been drawn on and applied to this ship. Because of the prevalent bad weather, "Winter North Atlantic" is associated with the greatest freeboard or least permissible draft that is assigned to any ship. Records of recent years show that it is possible to use smaller coefficients and finer ends in the North Atlantic. This is especially true as the length of the vessel increases farther beyond the length of the usual North Atlantic waves encountered. Most authorities agree that these waves are in the order of from 300 to 600 feet long with very few wave lengths, if, any, increasing to as much as 900 feet. Associated with length must be good freeboard to insure the highest percentage of the service speed being maintained. One feature of the *United States* is the fact that there is no well and forecastle forward. The promenade deck is continuous right to the bow at a high level, which makes for a dry ship forward, even when travelling at relatively high speeds. The flare at the bow high up becomes of secondary importance, as compared to the flare in the slower speed, shorter ship that more nearly approximates to the wave length and the wave period of the North Atlantic. In the North Atlantic, under the weather conditions experienced, the structure is of prime consideration. The numerous structural difficulties of large ships of the past emphasize the importance of the structural design. The structure cannot be judged by ordinary standards, as allowances must be made for dynamic effects and the severe racking strains to which the vessel is subjected. Since this must be done with the minimum weight, there is every reason to use a system of framing which will dispose the material to the best advantage. The use of welding is, of course, one of the most important reasons for the difference between the present-day transatlantic vessel and those designed and constructed prior to 1940. The saving in weight is considerable and the resulting smoothness of shell and decks contributes materially to improved performance. The use of new materials has helped to restore the balance between machinery and hull weights and centres. Structurally, this is accomplished by the use of aluminium in the topsides. This has an added advantage as the lower modulus of elasticity of the material permits the omission of expansion joints without an increase in weight.—*Paper by M. G. Forrest, read at a meeting of the Society of Naval Architects and Marine Engineers, 17th September 1952. Abstracted in S.N.A.M.E. Bulletin, Vol. 7, October 1952; pp. 33-34.*

Residual Resistance by Direct Computation

This paper presents a method for computing the form resistance or residual resistance of displacement types directly from the lines by means of an application of elementary hydraulics. The method employs no coefficients and no comparisons with existing designs or models. It is presented in detail, including a sample problem. Computations have been performed for various types and sizes from a large liner down to a small power cruiser. In all cases, agreement between computed results and values derived from trial results or model experiments has been excellent when compared with total resistance minus computed frictional resistance. The method is presented as a practical answer to a problem that has long defied solution when approached from the purely theoretical standpoint. To date, it has not been found to conflict with any of the observed and established general phenomena in connexion with form resistance. The method hinges on three conceptions which take the form of the ship, as represented by the lines, into direct consideration, and yield results which are sufficiently accurate for most practical purposes. First, a curve called the "M-Curve" is computed, from the lines. This curve represents the average of the infinite number of paths which the water takes in flowing around the hull. Second, it is assumed that, in the forebody, each section of the immersed surface of the ship's form contained between any two athwartships sections is acted upon by a jet of water equal to, and having the same shape as, the immersed area on the body plan contained between the two sections in question. Upon reaching the surface of the hull, each jet is diverted through an angle represented by the slope of the "M-Curve". Knowing the volume per second of each jet and the angle through which it is diverted, the resistance component in the direction of the ship's motion is computed. Third, it is assumed that gravity is the only force causing the water to return in the run. The potential velocity is computed for each after station, from the lines. This is reduced by an amount equal to the ship's speed, and a further reduction is made to allow for the forward component of the additional velocity required to close the water normal to the skin of the ship. The velocity remaining after these reductions are made is converted to pressure. The average of these pressures, taken at the stations at the forward and after ends of the section of the hull under computation, is considered as acting upon an area which is the difference between the immersed sectional areas of the two stations. These pressures may be positive or negative, and are algebraically combined, with sign reversed, to the summation of the resistances of the forebody. The above is an outline of the basic principles of the method. This procedure is modified, in practice, by use of the wave profile in lieu of the load water line. A tentative mathematical method for plotting the wave profile line, based on a combination of theory and observation, forms the second part of the paper.—*Paper by K. E. Mathews, read at a meeting of the Society of Naval Architects and Marine Engineers, 22nd May 1952. Abstracted in S.N.A.M.E. Bulletin, Vol. 7, October 1952; p. 35.*

Effect of Pitch and Blade Width on Propeller Performance

Results are given of tests of thirty-seven propellers, each with three blades, covering a wide range of pitch ratio 0·4 to 2 and of blade area ratio from 0·2 to 1·1. Each model was of 20 inches diameter, tested at high duty from zero to one hundred per cent slip. Thrust and torque coefficients increase appreciably with pitch and generally, but to a less extent, with blade area. The most favourable peak efficiency is 0·84 appropriate to the greatest pitch and narrowest blade. Peak efficiency is least for the smallest pitch and widest blade tested and is less than one-half of the optimum. No significant improvement in peak efficiency is expected from an increase of pitch ratio above the largest value of 2 covered by the series. More favourable peak efficiency is expected by reduction of blade area ratio below the lowest test value of 0·2 but the blades would then be exceptionally narrow. Theoretical deductions indicate that laminar or transitional flow has been avoided and that the critical Reynolds' number in this respect is ¼ million. The tests extended to a number of 4·3 millions. One upshot of this considerable advance is that the results may be expanded to ship propellers with only a small correction for skin friction resistance. Peak efficiency of ship propellers may be a little more favourable but this is subject to smoothness of finish. Departures from the results of previous series are noted which may be due to scale effect and differences in design. The investigation may prove useful as one step in the clarification of scale effect of propellers to which importance is attached by the International Conference of Ship Tank Superintendents.—*Paper by R. W. L. Gawn, read at the Autumn Meeting of the Institution of Naval Architects in Rome, 29th September 1952.*

Modern Tramp Ship

The single-screw m.s. *Stylehurst,* which recently completed trials from the Neptune shipyard of Swan, Hunter and Wigham Richardson, Ltd., is owned by the Grenehurst Shipping Co., Ltd., London, one of the Hadjilias group of companies; she is an open shelterdecker with a long forecastle and 'tweendecks suitable for the carriage of motor cars and trucks, also a space aft for special cargo. The forecastle 'tween decks are arranged for the carriage of cargo, and the cargo holds are fitted with wood shifting boards and feeders in way of the hatches for the carriage of grain in bulk. Furthermore, the tank tops in the cargo holds are specially strengthened to carry ore cargoes. She was built under the direction and supervision of Harte, Lindley and Co., Ltd., to requirements for the highest class of Lloyd's Register, and her principal particulars are: —

Length b.p.	435ft. 0in.
Breadth, moulded... ...	60ft. 0in.
Depth, moulded to shelter-deck	39ft. 0in.
Shelterdeck, above upper deck	9ft. (top of beam to top of beam)
Forecastle	8ft. (top of beam to top of beam)
Deadweight capacity ...	10,008 tons
Corresponding draught ...	26ft. 5in.
Gross register	5,750 tons
Machinery	4,400 b.h.p.

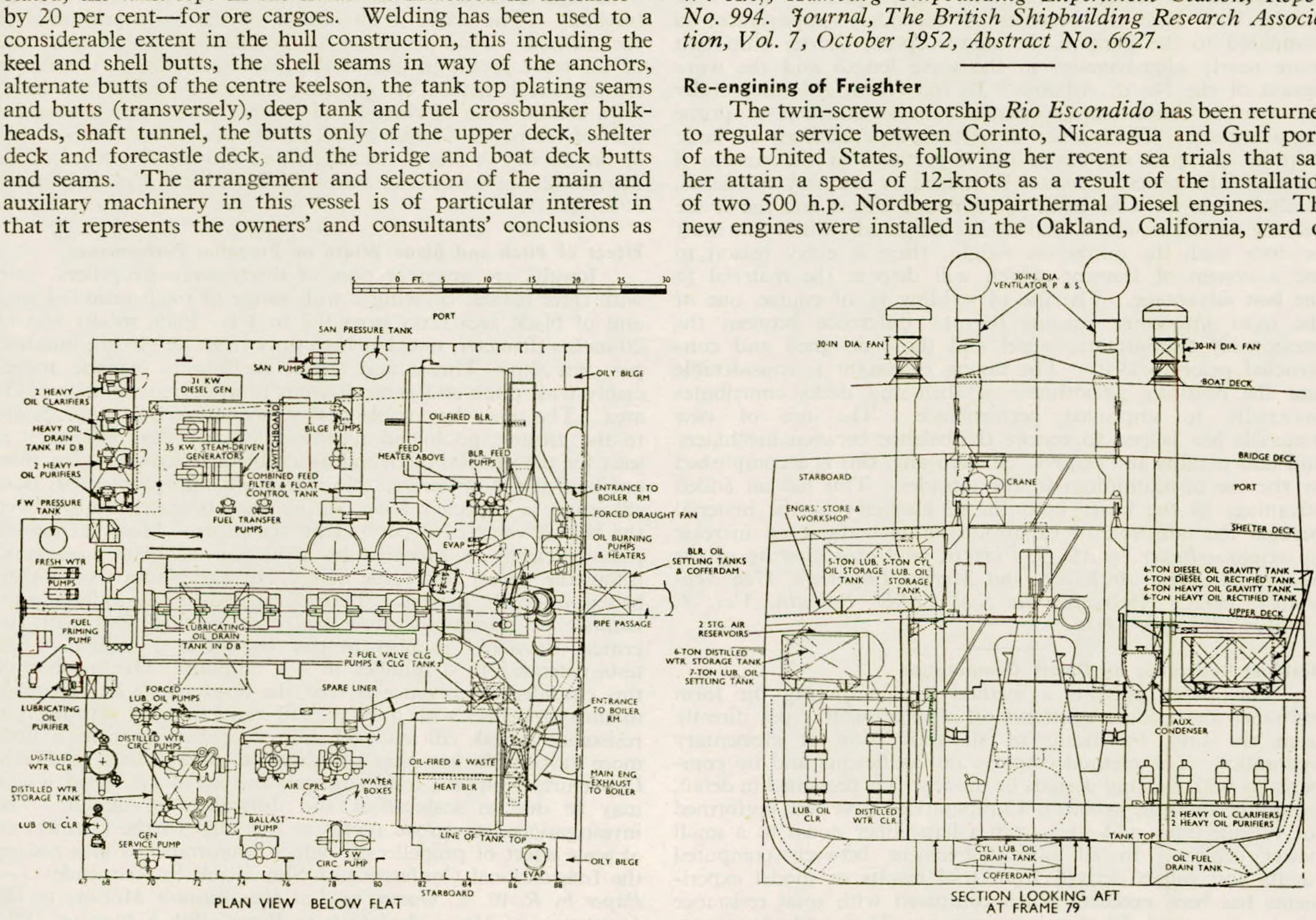
The ship is constructed with a straight raked stem, a cruiser stern, a top gallant forecastle, two masts and two derrick posts. A cellular double bottom extends aft from the forward collision bulkhead to the tunnel well. The shelterdeck stringer-plating is 10 per cent thicker than required by Lloyd's, from the after end of No. 2 hold to the after end of No. 4 hold. As already stated, the tank tops in the holds are increased in thickness—by 20 per cent—for ore cargoes. Welding has been used to a considerable extent in the hull construction, this including the keel and shell butts, the shell seams in way of the anchors, alternate butts of the centre keelson, the tank top plating seams and butts (transversely), deep tank and fuel crossbunker bulkheads, shaft tunnel, the butts only of the upper deck, shelter deck and forecastle deck, and the bridge and boat deck butts and seams. The arrangement and selection of the main and auxiliary machinery in this vessel is of particular interest in that it represents the owners' and consultants' conclusions as to the most efficient installation from the aspects of fuel and maintenance economy combined with simplicity and the avoidance of over-elaboration. For this reason, steam auxiliaries are installed. The four-cylinder Swan, Hunter-Doxford propelling engine has a cylinder bore of 670 mm. and a piston stroke of 2,320 mm.; it develops, in service, 4,400 b.h.p. at 115 r.p.m., the corresponding mean indicated pressure being 88lb. per sq. in. Arrangements have been made to enable the engine to operate efficiently on fuel having a viscosity of between 3,000-3,500 secs. Redwood No. 1 at 100 degrees F.—*The Motor Ship, Vol. 33, December 1952; pp. 340-343.*

Approximate Formulas for Calculating Wetted Surface Area

A number of approximate formulas for determining the area of the wetted surface of ships' hulls have been suggested. The majority of these are of the same general form, the length being multiplied by the sum of the breadth and the draught; and they vary only by the factors that modify the latter two dimensions in order to take the shapes of the bow and stern into account. In this paper, the author develops a further approximate formula of this type, and selects his factors by inserting the known values for the wetted area for a number of ships. The resulting formula is said to give a high degree of accuracy for a single-screw vessel with a vertical stem, length between 200 and 650 feet, and a midship-section area coefficient of approximately 0·983. A table is drawn up giving the principal dimensions of a number of vessels, their true wetted surface area, and the error in the wetted surface area as predicted by a number of these approximate formulas. The error in the result as given by the author's new formula is in the majority of cases appreciably lower than that resulting from the other formulas; in none of the examples given does the error due to the use of the new formula exceed 1·1 per cent.—*W. Alef, Hamburg Shipbuilding Experiment Station, Report No. 994. Journal, The British Shipbuilding Research Association, Vol. 7, October 1952, Abstract No. 6627.*

Re-engining of Freighter

The twin-screw motorship *Rio Escondido* has been returned to regular service between Corinto, Nicaragua and Gulf ports of the United States, following her recent sea trials that saw her attain a speed of 12-knots as a result of the installation of two 500 h.p. Nordberg Supairthermal Diesel engines. The new engines were installed in the Oakland, California, yard of



Engine room plans of m.s. Stylehurst

the Pacific Drydock and Repair Company. A converted British L.C.T., the *Rio Escondido* is 238 feet long, with a beam of 38 feet, a draught of 14 feet and has a capacity of 1,600 tons. Built as an all riveted steel ship, the *Rio* was converted into a cargo vessel in Norway. Welded sections of moulded bow and stern were attached to the existing hull to improve her lines and seagoing ability. This also provided for a raised forecastle head which gives the vessel better performance in rough weather. Basic reason behind the repowering of the *Rio* was the vessel's inability to maintain reliably schedules with her four original high speed Diesel engines which were each rated 400 h.p. at 1,500 r.p.m. and powered the twin screws through 1·5 : 1 reduction gears. The two main Nordberg engines installed in the *Rio Escondido* are right and left hand direct drive, direct reversing, four cycle units with eight cylinders of 9-inch bore and 11½-inch stroke. Each engine develops its normal rating of 500 h.p. at 500 r.p.m. Propulsion is through two new 50-inch diameter by 37-inch pitch propellers. The propeller shafts outside the hull are supported by large struts and strut bearings at the propeller end. Two rudders, located each immediately behind their respective propellers, give the vessel excellent steering qualities. The Nordberg engines are installed in the engine room at the aft end of the main hold, and are spaced on approximately 12-foot centres. By installing two propulsion units instead of the original four Diesel engines, easy access is afforded to all principal and auxiliary machinery. Maintenance cost will be substantially lower. All auxiliaries for the propulsion engines are located in the main engine room. Two motor driven air compressors, one automatic and the other with manual control, supply starting air which is stored in two 18-inch × 76-inch, 250lb. per sq. in. starting air tanks. One motor driven fuel transfer pump takes fuel oil from double bottom tanks. It then pumps it through strainers and a fuel oil filter into the main engine supply and settling tanks from which each main engine draws its fuel.—*The Log, Vol. 47, October 1952; pp. 82, 84.*

Motion of Ships at Sea

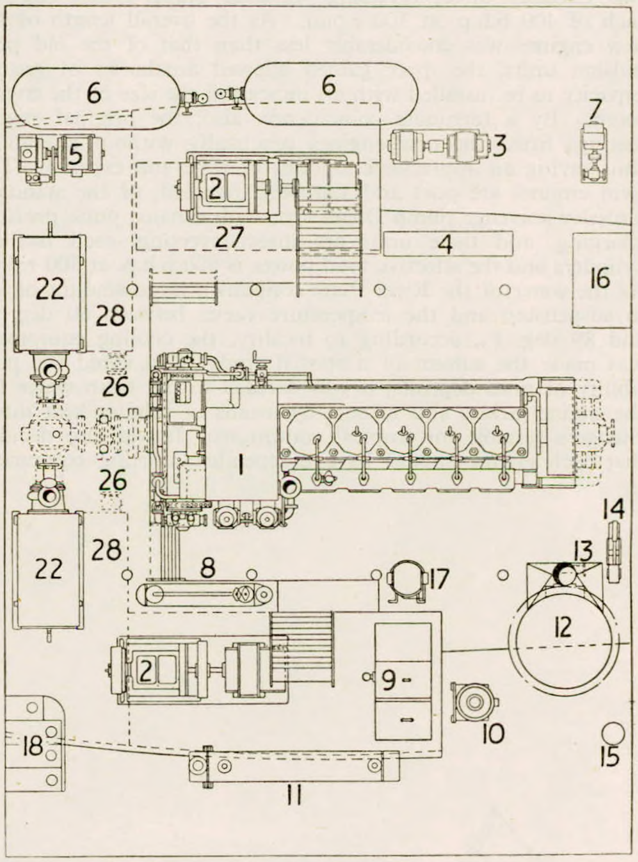
A number of sea trials have been carried out by the Admiralty in recent years to obtain data for use in the design of stabilization systems. This paper describes in general terms the trials carried out and the instrumentation used. Some interim results of the incomplete analysis of the records obtained from these trials are given, with particular reference to rolling. It is shown that under confused sea conditions the predominant rolling period is the appropriate still water period, and a statistical law governing the frequency of occurrence of roll amplitudes of various magnitudes is suggested. Some notes are included upon the analysis of curves of declining angles, and the paper concludes with some general remarks upon the nature of pitching.—*Paper by A. J. Williams, read at a meeting of the Institution of Naval Architects in Rome, 29th September 1952.*

Finnish Ferryboat

The new double-ended ferry built by Valmet OY, Helsingin Telakka, Helsinki, Suomi (Finland) is of all-welded construction and the most up-to-date features are embodied in her design. The main dimensions are as follows:—

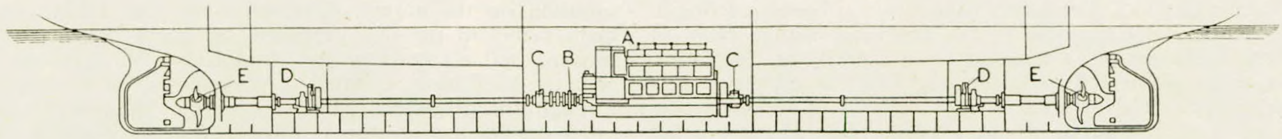
Length overall	115ft. 0in.	(35·0 metres)
Breadth moulded... ...	29ft. 6in.	(9·0 metres)
Depth moulded	13ft. 5in.	(4·1 metres)
Draught	9ft. 10in.	(3·0 metres)
Displacement (loaded) ...	300 tons	

Designed to carry 354 passengers and two large buses or trucks, the vessel has a speed of 10 knots. As the ferry is for service in Helsinki harbour the double-ended design, with a single propeller and rudder fore and aft, was chosen to avoid the difficulty of turning round in the ice which covers the harbour during some parts of the year. The application of the double-ended drive is worthy of note as it is unusual if not unique. A Crossley scavenge pump Diesel engine, type HGN5, with a continuous rating of 400 b.h.p. at 370 r.p.m., is the power unit of the vessel. This engine is a uni-directional model, being a variation on the normal direct-reversing type HRN and the main drives to the two propellers are taken direct from the forward and aft ends of the crankshaft. No reversing gear of any description is fitted. Manœuvring is accomplished through the propellers, which are of the "Kamewa" variable pitch design, manufactured under licence in Finland. Both propellers are in operation at the same time when the engine is running, and the power can be divided between the fore and aft propellers at the option of the operator. In normal conditions one-quarter of the power is absorbed by the forward propeller. For ice conditions, however, this propeller takes a larger proportion of the full load and the water can also be pumped ahead and astern alternately to clear the ice, in the same manner as an ice-breaker. All manœuvring is carried out from either of the two wheelhouses, the propellers and engine speed being remotely controlled from a control desk. To meet these extremely arduous conditions the fore and aft thrust blocks are of increased proportions, the drive to the aft propeller being carried through a Wigglesworth "Eflote" flexible coupling which is a heavy duty, all metal self-aligning coupling of the gear type. The engine is located amidships just below the main deck which at this point has to accommodate the two large buses or trucks previously mentioned. The low head-



Floor plan of engine room

1. Main engine
2. Generator set
3. Motor-converter: alternator to direct current
4. Switchboard
5. Air compressor
6. Air receivers
7. Fuel transfer pump
8. Lubricating oil cooler
9. Lubricating oil tank
10. Fire and bilge pump
11. Fresh water cooler
12. Donkey boiler
13. Automatic oil burner
14. Boiler feed pump
15. Water heater
16. Work bench
17. Cooling water pump
18. Tanks—fuel and lubricating
22. Fire pump
26. Main injection valve
27. Hotwell
28. Ice box



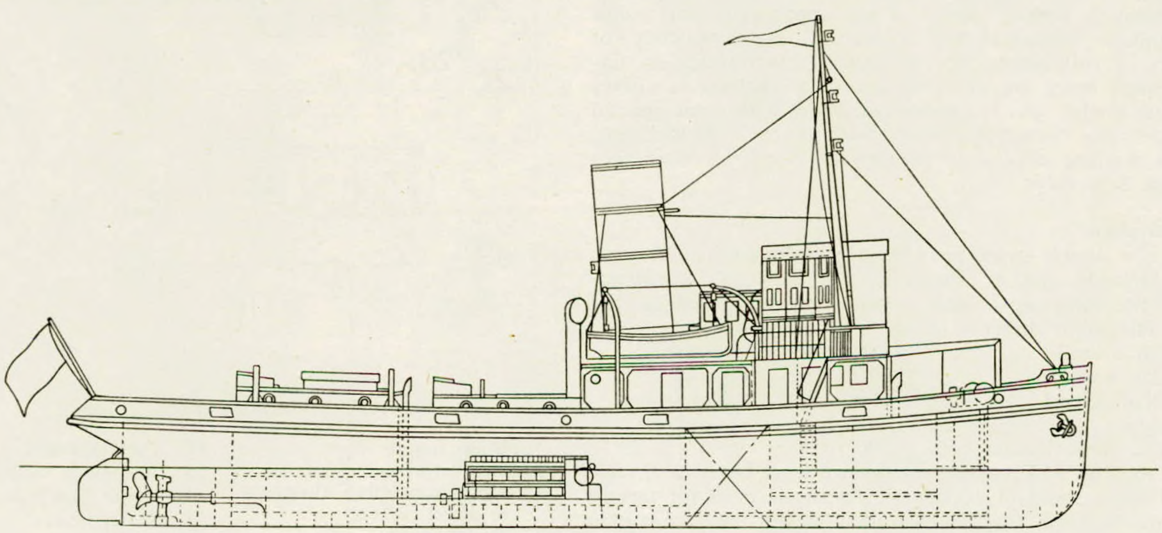
Profile and arrangement of drive to propellers

A. Main engine. B. Flexible coupling. C. Thrust block. D. Propeller operating mechanism. E. Reversible propellers.

room required by the engine is therefore an important feature of the installation. The ferry service, which is operated by Suomenlinnan Liikenne OY—Sveaborgs Trafiks AB, is between the port of Helsinki-Helsingfors and Suomenlinna-Sveaborg which is a small island at the entrance to the harbour of Helsinki.—*Crossley Chronicles, No. 161.*

Repowering Argentine Tug

The tug *Carlos Lumb* was originally fitted with two 250 b.h.p. two-cycle oil engines of the hot bulb type with crankcase compression. These engines having reached the end of their useful life, it was decided to replace them with units of modern design. The engines chosen for the conversion were two Crossley direct reversing scavenge pump Diesel engines, each of 400 b.h.p. at 300 r.p.m. As the overall length of the new engines was considerably less than that of the old propulsion units, the space gained allowed auxiliaries of greater capacity to be installed without increasing the size of the engine room. By a fortunate coincidence, also, the original engine seatings fitted the new engines practically without alteration, thus saving an appreciable amount of time and expense. The twin engines are port and starboard handled, of the standard Crossley scavenge pump Diesel type with exhaust pulse pressure charging, and these units are direct-reversing, each has six cylinders and the effective total power is 800 b.h.p. at 300 r.p.m. As the water of the River Plate contains a large amount of silt in suspension and the temperature varies between 60 deg. F. and 89 deg. F., according to locality, the cooling equipment was made the subject of a special study. To avoid the possibility of mud deposits, it was decided to use fresh water for the engine jackets and to cool by means of tubular heat interchangers suitable for tropical conditions. It was also decided that each engine should have independent cooling equipment consisting of heat interchanger, fresh water pump and river water pump, the pumps in each case being driven directly from the main engine crankshaft. The question of employing electrically driven pumps was carefully considered, but to reduce the size of the electrical generator and for other considerations it was finally decided to use mechanically driven pumps. The river water circulating pumps of the propelling engines are in duplicate. One of the pumps is normally used for the bilge, but by cross-connexions and valves either pump may be used for water circulation. Both pumps are of the ram type. High and low level water intakes are provided for the river water inlets and a multiple valvebox is arranged for convenience of operation. As a tug requires more manœuvring than the ordinary vessel, four large air receivers are provided for the storage of compressed air. For charging the receiver, each main propulsion engine incorporates an air compressor driven directly from the crankshaft. These compressors operate continuously whilst the engines are running and provide ample air to maintain the receivers recharged to the normal working pressure. When this pressure is reached an automatic cut-out comes into operation. The original steering gear was operated by a compressed air motor. This gear had proved reliable in every way but required 20 h.p. to drive the compressor, which had to run continuously whilst the tug was in service. To effect an economy in power consumption the original compressed air system was replaced by the Hyland hydraulic steering gear. This is operated by a small hydraulic pump driven by an electric motor and situated in the engine room. The steering wheel on the bridge controls a simple valve system which feeds the oil pressure to rams directly connected to the rudder shaft. Should the pump stop or fail, the hand operation gear is brought into action by a simple lever on the steering column.—*Crossley Chronicles, No. 162.*



Profile of the Carlos Lumb

Main particulars:

Length b.p.	111 feet
Breadth	23 feet
Draught	10 feet
Total engine output	800 b.h.p.

Synchronizing Marine Engines

With the synchronizing device for two marine engines, illustrated in Fig. 1, the engine shafts (8), which rotate in opposite directions, carry gearwheels (1, 2). The gears (3, 4) are concentric with the sleeves (5, 6). While the planet carrier of the differential gear (18) is stationary, and the shafts (8, 9) run at the same rate, the sleeves (5, 6) also rotate at the same speed, in the same direction. The cavity (21) is filled with oil and if a variation of the normal relative position of the ports (30, 31) occurs, oil will escape through the port (30) to the cavity (32) and flow through the connexion (34), the

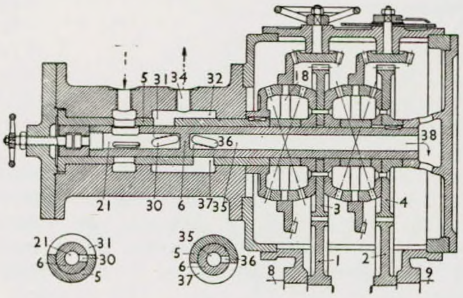


FIG. 1.

effect being to readjust the fuel lever of one engine to the extent required to regain the normal relative positions of the ports (30, 31). The cavity (35) in the hollow shaft (6) has a port (36) opposite the opening to the cavity (37) in the outer sleeve (5). When the synchronization disappears in the opposite sense to that mentioned, the port (36) is opened. Oil flows back through the connexion (34), enters the cavity (32), the space (37) and the port (36). The oil then flows freely from the cavity (35) away in the direction of the arrow (38). The effect of the discharge of oil from the regulating system is opposite that brought about by the entrance of the oil and thus the normal relative position of the ports (30, 31) is attained.—*Brit. Patent No. 678,531. N.V. Koninklijke Mij. De Schelde, Flushing. The Motor Ship, Vol. 33, December 1952; p. 374.*

Six Recent Atlantic Liners

A survey of six recent Atlantic liners, namely, the *Caronia, Orcades, Giulio Cesare, Andrea Doria, Independence,* and *United States,* leads the author to the following conclusions: Limitations on the general dimensions of liners are imposed by their ability to approach the coast. The future and present liners of moderate dimensions, i.e. 30,000 or 35,000 tons displacement, are limited in length to about 700 feet and in draught to about 30 feet by the necessity of mooring in the interior of harbours. Also the superliners which must moor in the roadstead are limited in dimensions by the width of the channels and by the dimensions of dry docks and even of floating docks. Today the 1,100 × 110 × 41½ feet of the Panama Canal locks might be an upper limit for the dimensions of the body forms and therefore for the superliner displacement, which should be not more than 60,000 tons, as the last United States giant aircraft carrier *J. W. Forrestal,* or, even disregarding the passage through Panama Canal, not more than the *Queen Mary* and *Queen Elizabeth* displacement. On the other hand, the increase of ship's speed is so far from the ever-increasing speed of aircraft that any further rise in the displacement of liners would be foolish. For a long time the resistance to the motion of the body forms has been the object of the most intensive experimental research at all tanks of the world. Their results have attained a high degree of perfection, specially for the forebody. The study of the afterbody complete with all its appendages, and of propellers, is worth further research. Also the gap between self-propulsion results at the tank and standardization trial results at sea deserves further technical inquiry. This gap is clearly revealed by the comparison of the propulsive efficiency either in its theoretical expression of efficiency EHP/s.h.p. or, if preferred, in its practical expression, i.e., the Admiralty coefficient. For instance, the value of efficiency EHP/s.h.p. at the tank is often 0·70 and even 0·72, while sometimes it is less than 0·60 on trial at sea. With the other system the value of the Admiralty coefficient at the tank is often 350 and even more, while sometimes it works out less than 300 on trial at sea. From time to time reference is made to particular body forms concerning specially the fore- and the afterbody. Sometimes attempts have been made to suppress the deadwood with a spoon-like form of the afterbody, obtaining indeed improved results at the tank both in the towing resistance and in the self-propulsion. But the vessels were not steady on a straight course and it was necessary to retain the deadwood, which may be considered a vertical stabilizing tail. Although the Maier and Yourkevitch forms have given good results in practice, it has been said that similar results could be obtained with the conventional forms, provided there is a correct distribution of longitudinal volume. From time to time the proposal of contra-propellers has arisen; the theoretical principle is sound, but until now the practical application has not been favourable. The anti-rolling devices for liners deserve consideration. The complete electrification of all the auxiliaries is advisable for the future liners. Vertical auxiliary engines are already introduced, and a greater extension of this system is advisable in order to reduce horizontal obstructions, especially in the machinery space. As regards the auxiliaries, the evaporators are worth the greatest attention and extension. In many passenger liners about 30 per cent of their deadweight is often absorbed by the fresh water for make-up of condensers and for the large domestic use. It may appear absurd that a liner surrounded by water should be a tank for a thousand tons of fresh water when one ton of fuel oil may produce forty tons of fresh water. Today the evaporators heated by steam bled from the turbines may represent an economic solution.—*Paper by E. de Vito, read at a meeting of the Institution of Naval Architects at Geneva, 26th September 1952.*

Longitudinal Strength of Ships

In 1942-43 several welded ships developed serious fractures, and at that time it was thought that there must be some fundamental difference in structural behaviour between welded and riveted ships, since the latter had not suffered to nearly the same extent. It was obvious that new problems had arisen and in consequence it was decided, almost simultaneously in the U.S.A. and in the United Kingdom, to set up research committees to investigate why welded ships were behaving differently from the riveted ships. This presented a good opportunity to obtain additional knowledge of ocean waves as well as of the detailed behaviour of some typical ships' structures. The work which these committees instigated has undoubtedly resulted in a clearer understanding of many of the factors involved in the longitudinal strength of ships. The first full-scale experiment carried out under the direction of the Admiralty Ship Welding Committee was a comparison between the behaviour of the welded tanker *Neverita* and the riveted sister ship *Newcombia* under hogging and sagging bending moments applied in still water, but no important overall difference was revealed. The next ships compared were the welded *Ocean Vulcan* and the riveted *Clan Alpine,* sister dry cargo ships of 416 feet in length, 56ft. 10½in. in breadth, and 37ft. 4in. in depth to the strength deck. They were of standard design and were transversely framed, and slight differences in behaviour were observed during the still water tests. The decision was made to fit the *Ocean Vulcan* with instruments capable of recording at sea the wave pressures, the wave profiles on the ship's sides, the wind velocities, angles of roll, pitch and yaw and the forces imposed on the ship and cargo by accelerations. The observations to be made at sea were to include the determination of ocean waves by stereophotographic survey and other methods. With these instruments the *Ocean Vulcan* made eight double crossings of the Atlantic over a period of seventeen months, and rough sea conditions were experienced on several occasions. Much valuable information was obtained from these sea trials, and in addition to the measurement of waves and wave pressures on the ship, investigations were carried out on the effects on the hull structure of vertical bending, horizontal bending, torsion, heaving and pitching, axial compression and slamming. A statistical strain gauge was also fitted to the *Ocean Vulcan* and it has been in satisfactory operation for over a year. Some interesting data have now been collated. Typical results over a period of one year are as follows: —

Range of stress.	Number of times experienced.
1 ton per sq. in. (158 kilo. per sq. cm.)	266,884
2 tons per sq. in. (315 kilo. per sq. cm.)	7,105
3 tons per sq. in. (473 kilo. per sq. cm.)	1,329
4 tons per sq. in. (630 kilo. per sq. cm.)	102
5 tons per sq. in. (788 kilo. per sq. cm.)	5
6 tons per sq. in. (945 kilo. per sq. cm.)	2

It will be noted that the maximum range was only 6 tons per sq. in., and that range on only two occasions during one year of service. It is obvious that much more severe conditions could be experienced, and these may be recorded during this investigation, which is being continued. Although ships are subject to fatigue loading, it is by no means clear that fatigue is an important factor in the longitudinal strength of ships. It may, however, be important in regard to regions of stress concentrations when a ship has been consistently subjected to an injudicious longitudinal distribution of cargo.—*Paper by J. Turnbull, O.B.E., read at a meeting of the Institution of Naval Architects in Genoa, 26th September 1952.*

B. and W. Turbo-charged Engine

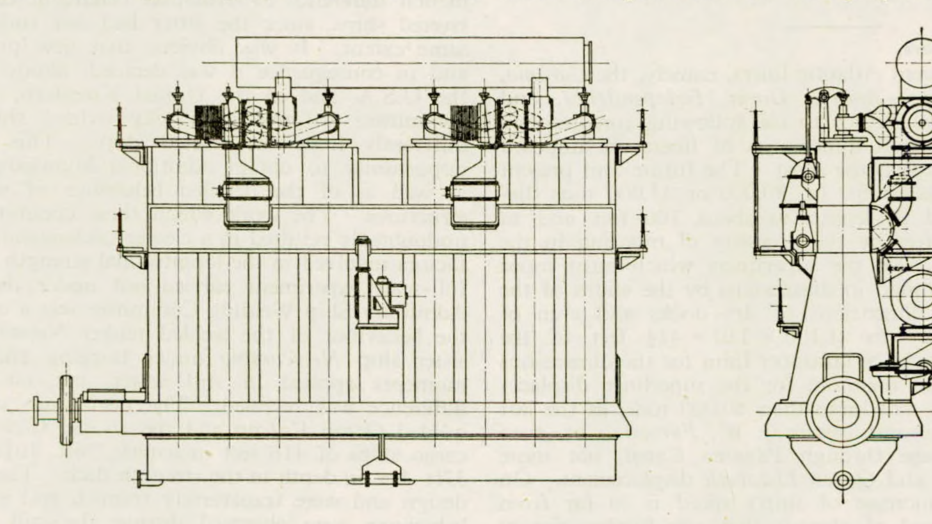
Below is shown an outline drawing of an eight-cylinder B. and W. turbo-charged engine with a cylinder diameter of 740 mm., and a piston stroke of 1,600 mm., the continuous output at 115 r.p.m. being 10,000 b.h.p. This corresponds to an increase in the effective output of 35 per cent, as compared with a non-turbo-charged engine, having the same dimensions and revolution speed. The corresponding mean indicated pressure is about 8 kg. per sq. cm. or 118lb. per sq. in. The fuel consumption from about 60 per cent to 100 per cent load is stated not to exceed 150 gr. per b.h.p. hr. or 0·33lb. per b.h.p. hr., with a minimum figure slightly below this. Two turbo-chargers are installed on the upper platform close to the exhaust valves of the engine. The small electrically driven emergency blower is at the rear of the engine.—*The Motor Ship, Vol. 33, December 1952; p. 355.*

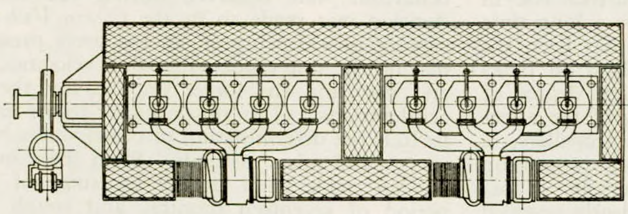
One-class Passenger Liner

With the completion by Harland and Wolff, Ltd., of the passenger and cargo liner *Braemar Castle*, the Union-Castle Line has completed its group of four one-class passenger liners The leading particulars are as follows: —

Length o.a.	576 feet
Length b.p.	540 feet
Breadth moulded	74 feet
Depth moulded	35ft. 6in.
Tonnage gross	17,040 tons
Tonnage net	9,435 tons
Deadweight	10,693 tons
Passengers...	530
Speed	17½ knots

The *Braemar Castle* has four complete steel decks, a lower deck forward and aft of the machinery space, promenade deck and boat deck. The hull is divided into ten watertight compartments by nine watertight bulkheads, all extending to the upper deck, and there is a continuous double bottom for fresh water, water ballast and oil fuel. The forward and after peaks are arranged for fresh water or water ballast. Deep fresh-water tanks are also arranged at the sides of shaft tunnel aft, and deep oil fuel tanks are fitted across the vessel forward of the boiler room. There are two cargo holds forward and three after of the machinery space, with corresponding tweendecks extending to the underside of the upper deck. The lower holds, Nos. 1 and 2 lower and Nos. 1, 3 and 5 main tweendeck spaces are arranged for ordinary cargo. Nos. 3 and 4 lower and No. 4 main tweendeck spaces are insulated and refrigerated for the carriage of fruit, etc., certain of the compartments being specially arranged for chilled or frozen produce. No. 2 main tweendeck is arranged for provisions, mails and baggage. The cargo hatches to the five holds are served by eleven tubular steel derricks worked by electric winches. The heating of the vessel is carried out electrically with heaters in the alleyways, and in the individual cabins and public rooms. The propelling machinery, made by





Outline plans of a 10,000 b.h.p. B. and W. turbo-charged engine the shipbuilders, consists of a two-shaft arrangement of Parsons' triple expansion, condensing, double reduction geared turbines, designed in accordance with Parsons' latest practice. The ahead turbines are of the all-reaction design, while the h.p. and the l.p. astern turbines incorporated in the i.p. ahead and the l.p. ahead casings respectively are of impulse type. The turbines are so disposed around the main gear wheel that simplicity of overhaul and ease of access to all parts is assured. Three oil-fired watertube boilers of Babcock and Wilcox latest design supply steam at a working pressure of 450lb. per sq. in. at the superheater outlet.—*The Shipping World, Vol. 127, 3rd December 1952; pp. 437-439.*

Asphalt-carrying Tankers

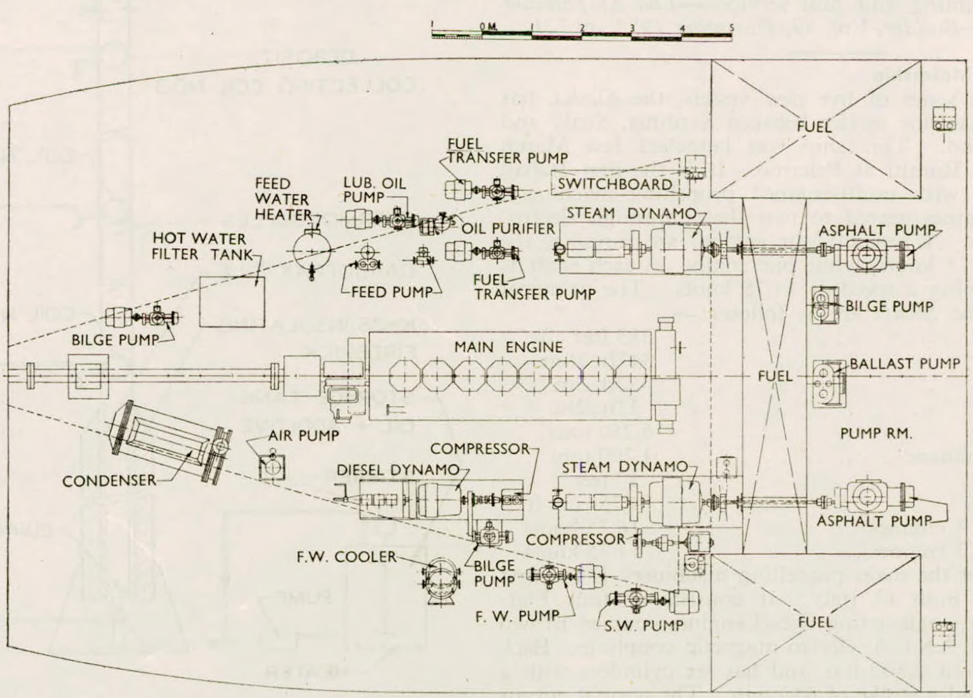
Two tankers of special construction for the carriage of asphalt in bulk were ordered some time ago in France, one to be constructed by the N.V. Scheepswerf Gebr. van der Werf, Deest, and the other by the S.A. des Anciens Chantiers Dubigeon, Nantes. The first vessel, named *Esso le Caroubier,* is owned by the Esso Standard Company, Paris. The second ship is named *Port Etienne* and she is owned by the Soflumar Company, Paris, her completion being due next year. Although the two vessels are similar in all main respects, the following description refers particularly to the *Esso le Caroubier* and the leading characteristics are given in the following table:

Length o.a.	304ft. 0in.	92·65m.
Length b.p.	279ft. 0in.	85·00m.
Breadth	42ft. 0in.	12·80m.
Depth	23ft. 0in.	7.00m.
Draught	16ft. 1in.	4·91m.
Deadweight capacity	2,730 tons	
Speed	11·3 knots	
Machinery	1,400 b.h.p.	

Heating coils are fitted in each of the five asphalt cargo tanks and the construction is designed to limit heat radiation. The vessel is classified under the Bureau Veritas I 3/3 L.1.1. A et C.P. The hull is built without a centre-line bulkhead and the machinery, as well as most of the accommodation, is arranged aft. Below the main tanks is a double bottom giving a certain amount of insulation from the sea. Although the main object of the double bottom is to avoid frames and other obstacles on the bottom of the cargo tanks so that they may be easily cleaned and drained, the centre of gravity of the cargo is raised and the metacentric height is reduced. The double-bottom tank is intended to be kept dry. The two longitudinal side bulkheads are arranged at an angle, thus reducing to a minimum the surface above the cargo exposed to the air, in order to limit any cooling effects. Another advantage of this arrangement is the relatively large surface at the bottom of the cargo tanks, thereby making it possible to install the required heating coils with sufficient space between each to allow for ease in cleaning. Corrugated transverse bulkheads are fitted between the tanks. There are two 8-inch diameter cargo pipe lines, heated by steam pipes attached to them over the entire length, including the piping in the pump room. The cargo is handled by two rotary Houttuin double-worm pumps which run at 600 r.p.m. and are driven by extended shafts passing through the fuel cross-bunker at the forward end of the engine room. These shafts are driven by Reader steam engines, one of 90 h.p. and the other of 125 h.p., coupled to 50 kW. dynamos. For propulsion, a 1,400 b.h.p. four-stroke Werkspoor engine is installed and the Büchi supercharging system is adopted. The engine has eight cylinders with a diameter of 390 mm., the piston stroke being 680 mm. Fresh water is used for cooling and there are two pumps on the starboard side of the engine room, both of the Houttuin type, one for sea water, with a capacity of 100 tons per hour, and the other for fresh water, capable of supplying 40 tons per hour. The main boiler, located above the fuel cross bunker, has a heating surface of 135 sq. metres or 14,500 sq. ft. and has two oil-fired furnaces. The second boiler, in addition to being oil-fired, takes the exhaust gas from the main engine.—*The Motor Ship, Vol. 33, December 1952; pp. 360-361.*

New Oil Tanker

The recently launched *Caltex Canberra* is an oil-tank steamship of 544ft. 4in. overall and 515ft. 0in. b.p., by 70ft. 0in. moulded, by 39ft. 9in. moulded; 17,100 tons deadweight on 30ft. 4¾in. summer draught; 15 knots speed on trial. The vessel is the first of two tankers ordered from the Furness Company by the Overseas Tankship (U.K.), Ltd., London. The *Caltex Canberra* is of the poop, bridge and forecastle type, with machinery aft, and has a well-raked stem and cruiser stern. The framing is on the combined transverse and longitudinal system. Electric welding has been extensively used, the bottom shell and decks being completely welded and the longitudinal



Plan of engine room and pump room

and transverse bulkheads being welded on the Unionmelt system, as developed by the Furness Company. There are nine treble cargo-oil tanks, giving 27 oil compartments in all, with a capacity of 16,470 tons of oil at 50 cu. ft. per ton, after allowing 2 per cent for expansion. All arrangements (including the provision of deck steam and exhaust pipes) are made for the fitting, at a later date, of heating coils in the tanks. Fuel oil is carried in bunker tanks forward and aft, and in the fore deep tanks, the total capacity being 2,300 tons at 38·5 cu. ft. per ton. The cargo-pump room, arranged between the aftermost cargo tank and the engine-room, contains three turbo-driven single-stage centrifugal pumps, each delivering 625 tons of oil per hr., and three vertical duplex stripping pumps, each delivering 200 tons per hr. In the bilge and ballast-pump room forward, there are two horizontal duplex reciprocating pumps, one for fuel oil and the other for bilge and ballast. The cargo-oil piping consists of a 12-inch diameter pipeline, with one 12-inch suction to each wing and centre tank, and a separate 6-inch diameter stripping line, with a 6-inch suction to each tank. The discharge pipes amidships are 12-inches diameter, and the stern discharge 9-inches. Steaming out, fire extinguishing and gas freeing pipes are provided, and Butterworth tank cleaning apparatus is fitted. The general outfit includes two steel masts and topmasts, three steam winches, steam windlass, and electro-hydraulic steering gear, controlled by telemotor from the wheel-house, as well as mechanically from the boat deck and by an automatic gyro pilot. All the officers' and engineers' cabins have a private toilet adjoining. A mechanical system of ventilation and heating is fitted in the accommodation. Refrigerated chambers of about 3,920 cu. ft. are fitted in the poop 'tween decks, and electric Freon-type refrigerating machinery is installed. The navigating equipment includes wireless and direction-finder, radar apparatus, gyro compass, electric log, rudder indicator, echo sounding gear, electric telegraphs, etc. The propelling machinery, constructed by Richardsons, Westgarth and Co. Ltd., Hartlepool, consists of a set of double-reduction geared turbines developing 7,300 s.h.p. at 100 r.p.m., with a maximum of 8,200 s.h.p. at 104 r.p.m. Steam at a pressure of 450lb. per sq. in. is supplied by two Foster Wheeler "D" type boilers, burning fuel oil on the Todd system. Electric current is supplied by two 300-kW., 230 volts, turbo-generators and one 75-kW., 230 volts, Diesel-driven generator. In addition, two 50-kW. motor generators are provided to give 110-volt supply for lighting and hull services.—*The Shipbuilder and Marine Engine-Builder, Vol. 59, December 1952; p. 721.*

Italian Twin-screw Motorship

The first of a series of five new vessels, the *Sicilia,* has entered the Tirrenia Line service between Sardinia, Sicily and the Italian mainland. This ship was launched last March by Cantieri Navali Riuniti at Palermo. It is the first Italian vessel to be fitted with multi-engined propelling machinery. There are four engines geared to two shafts through electro-magnetic couplings. When all four engines are running, the service speed is 16·75 knots, while one engine on each shaft is capable of maintaining a speed of 14·75 knots. The principal characteristics of the *Sicilia* are as follows:—

Length overall	383 feet
Length b.p.	347ft. 10in.
Breadth	52ft. 2in.
Depth	32ft. 2in.
Gross tonnage	5,250 tons
Deadweight tonnage	1,200 tons
Draught	17 feet
Cargo capacity	54,750 cu. ft.
Service speed (4 engines)...	16·75 knots
Service speed (2 engines)...	14·75 knots

The arrangement of the main propelling machinery is the first of its kind to be built in Italy. It consists of four Fiat-Ansaldo two-stroke single-acting Diesel engines coupled to two shafts by means of A.S.E.A. electro-magnetic couplings. Each engine develops about 2,300 h.p. and has six cylinders with a bore of 480 mm. and a stroke of 640 mm. The normal output is 7,200 h.p. at 280 r.p.m., reduced to 150 r.p.m. at the shafts. The manœuvring position for each pair of engines is on the inboard engine, so that one engineer is able to operate a pair. Manœuvring can be carried out by the use of the couplings while one engine of each pair is running ahead and the other astern. Any possible risk of error while manœuvring has been eliminated by the use of locking and safety devices. The engines are not subjected to so many blasts of cold starting air as usual when this method of manœuvring is employed, nor is there such a large demand for compressed air. Fresh water cooling has been employed in the cylinder jacket, while the pistons are oil cooled. It is understood that maintenance work will be carried out on the engines when the vessel is on the slower service.—*The Shipping World, Vol. 127, 19th November 1952; p. 403.*

Oil Additives Relieve Boiler Slagging

At the Research and Development Centre of The Babcock and Wilcox Company, Alliance, Ohio, U.S.A., troublesome superheater deposits were found to be characterized by low-melting constituents, and laboratory work was carried out on selected materials which, when intimately mixed with the oil ash, would raise its melting point so that upon deposition on heating surfaces it would be dry and thus more readily removable. The effect on fusing temperatures by intimately mixing synthetic oil ash with various additive compounds was studied, using A.S.T.M. coal-ash fusibility determinations. The first fusion cones were made from an actual oil ash obtained by burning a quantity of a typical troublesome oil until the ash was left as a residue. Since this was a tedious and expensive way to obtain oil ash, all subsequent samples were compounded by mixing the individual constituents in proper proportion to obtain the same chemical composition as present in the oil ash. Comparable fusing temperatures were obtained when using this synthetic ash. A number of materials were added individually to the synthetic ash and it was found that some mixtures had higher fusing temperatures than the oil ash alone. The next step in the investigation was to set up a pilot furnace for testing promising additives under simulated conditions. The equipment consisted essentially of a small furnace in which the fuel-additive mixture was burned and a series of three air-

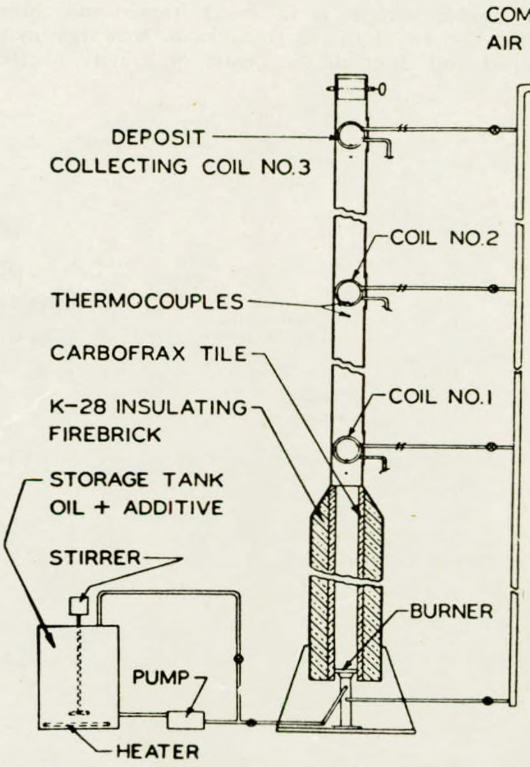


FIG. 2—*Schematic diagram of test equipment*

cooled coils upon which the resulting deposits from the products of combustion were collected (Fig. 2). The coils were placed at different levels in the stack to study the effect of various temperature zones on deposit formation on the cooled metal tubes. During the pilot tests the additive was mixed with the fuel prior to combustion, as it was believed that the effectiveness of the additive would be increased thereby owing to its more even dispersion through the fuel and its more intimate contact with the oil ash throughout the combustion process. For this reason a mixing device was incorporated in the test arrangement (Fig. 2). To accelerate the tests and increase the rate of deposit formation, Diesel fuel oil to which a synthetic ash was added was used to produce deposits similar to those obtained on commercial boiler units. Additives in the form of finely divided material suspended in the fuel and metallic organic compounds of the soap type (soluble or miscible with oil) were tried in the pilot test plant in various weight ratios to the ash in the oil. Alumina, magnesium oxide, and calcium oxide were found to be the most promising of the additives tested in producing a powdery deposit that was easily removable from the coils. This contrasted with the hard, glassy, fused material which was formed when these additives were not used.—*Marine Engineering and Shipping Review, Vol. 57, November 1952, pp. 59-62.*

Ship Model Correlation

The problem of ship model correlation is a real anxiety to the shipbuilder and to the experimenter alike. It is significant that the introduction of the fully turbulent model has coincided not only with the more widespread adoption of the smooth welded hull but also with the increasing preference for Diesel propulsion and its associated use of the low-pitch-ratio propeller. The smooth hull has destroyed the fortuitous validity of the Froude frictional coefficients; and the greater favourable efficiency scale effect of the low-pitch-ratio propeller, as compared with that of earlier normal and high-pitch-ratio screws, has caused ship propulsive coefficients now to be appreciably greater than those previously predicted from the model values. These two additive effects have recently produced some rather extraordinary errors in power prediction; and overestimates of the order of 30 per cent in the actual power found subsequently necessary in the ship are not unknown. Such errors have seriously disturbed British shipbuilders; and with some the model experiment has lost caste. This state of affairs is unfortunate; and since the model experimenter can hardly be blamed for welded ships and Diesel engines, all in the profession should appreciate the quite fortuitous circumstances which have necessitated the complete overhaul of model prediction methods preserved without fundamental change ever since Froude's introduction of the subject. Such overhaul requires the correct knowledge of ship and model resistance on the one hand; and the corresponding knowledge of ship and model shaft-horsepower on the other. The author submits what he believes to be a reasonably complete treatment of the methodology of ship-model correlation. This methodology works; and is unified in a way that no other existing treatment can similarly claim. It avoids quite a number of controversial issues and shows that their future solution need not delay the immediate evaluation of the correlation problem. For example, the true minimum turbulent line, whether specific resistance is finite or zero for zero relative viscosity are interesting issues, but they are not prerequisites to further correlation progress. Vital prerequisites are greater knowledge of edge effects on planks and their relation to radius effects on cylinders and ship models, since both lead to the demonstration of variable extrapolation. Studies of this by geoism research are being greatly hindered not only by cost but by tank wall-effect destroying complete similarity when the larger geoisms are tested. Numerous attempts have been made to solve this problem but a sufficiently reliable and comprehensive method is still awaited. The mastery of correlation will transform the model experiment. At present and in the past what the model has said goes. The shipbuilder has been led to believe that if model A is better than model B, then the corresponding ships will show the same comparison. This is, however, one of the grossest assumptions in the subject. It implies that models can first be tested accurately; and granted this, that they correlate to the ship in exactly the same way. The last few years have fortunately greatly changed professional conscience on model infallibility. The next few years should show a similar professional awakening on correlation implications. Only international sifting of every aspect of the problems involved will ensure the necessary perspective and energy being devoted to their solution.—*Paper by E. V. Telfer, read at a meeting of the Institution of Naval Architects in Genoa, 26th September 1952.*

Pneumatic Control Equipment

A range of measuring and controlling instruments operating on the pneumatic force balance principle has recently been introduced by Sunvic Controls, Ltd. The "null balance" principle of measurement, normally associated with electrical measuring circuits, is widely accepted to be capable of giving very accurate results, and this advantage is also a feature of the pneumatic "null balance" method of measurement. The method of operation of the "Nullmatic" instruments is briefly as follows: the variable to be measured is caused to exert a force on one side of a bellows unit and this force is automatically backed off by an equal and opposite force exerted by servo-air introduced to the opposite side of the bellows by means of a flap and nozzle form of control. This servo-air pressure is directly proportional to the value of the variable being measured and can be transmitted to an indicating instrument, which has the form of a simple pressure gauge but is appropriately calibrated, or it can be used for automatic regulation of the variable. Where continuous records of the variable are required, the servo-air pressure is connected also to a pressure type recording instrument. An outstanding unit in

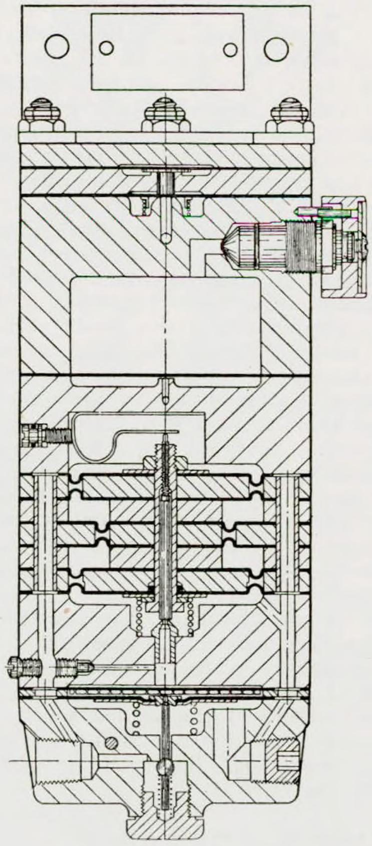


FIG. 1—*Sectional arrangement of "Nullmatic" control unit*

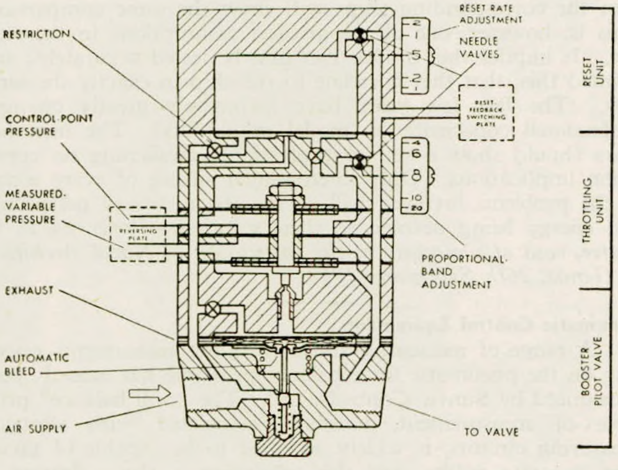


FIG. 2—*Arrangement of a "Nullmatic" control unit incorporating proportional plus reset features*

the new range is the "Nullmatic" controller. This instrument, also operating on the pneumatic force balance principle, is devoid of links, pivots, cams, etc., which, it is claimed, gives it a high quality performance. The controller is built up in the "stack" method of construction which affords easy access to all components and thereby simplifies maintenance. A sectional arrangement of a typical unit is outlined in Fig. 1. The only working parts of the controller are the nozzle, pilot and needle valves, all of which can be removed easily for cleaning. The controller can be pneumatically linked to any of the "Nullmatic" transmitters for the purpose of regulating flow, pressure, temperature, level and many other variables. The schematic arrangement of a controller incorporating proportional plus reset features is given in Fig. 2. The net effect of the forces within this controller is detected by the measuring nozzle in the booster pilot valve. A force-balance is established when the control point and the measured variable servo-pressures are equal, and when the valve, intermediate, reset and reset-reference pressures are all equal to one another, but not necessarily of the same value as the control point and measured-variable servo-air pressures; when the controller is in balance there is no flow of air, in fact, through either of the needle valves. When the proportional band adjusting needle valve is wide open, the controller is set to give the narrowest possible proportional-band, and the resulting action is practically equivalent to "on-off" control. Alternatively, if this needle valve is completely closed, a 200 per cent proportional-band is obtained, because the area of the control point diaphragm and that of the measured-variable diaphragm are in the ratio of 2:1. The purpose of the "reset" element of the controller is, of course, to bring the regulator (valve, etc.) to a fresh steady position as required by a change of load on the plant and which restores the measured variable to the desired value. In the "Nullmatic" controller, the rate of change of pressure of the air fed to the regulator is governed by the setting of the reset needle valve. The reset-reference pressure is reproduced in the reset-pressure chamber by means of the 1:1 ratio relay. Consequently, if the control circuit is not in balance, the reset-reference pressure changes at a rate dependent upon the setting of the reset needle valve. The air pressure in the intermediate chamber is thereby raised or lowered, thus producing the gradual and continued change in the pressure of the air fed to the regulator, and which is necessary to restore the control system to a state of balance.—*Engineering and Boiler House Review, Vol. 67, November 1952; pp. 328-330.*

Heat Transfer to Sprays of Water in Steam

An investigation is made into the mechanism of the flow of water, and the heat transfer from steam to the water, from the coarse spray nozzles used in commercial apparatus. The heat transfer can be regarded as taking place in two phases, the film phase and the drop phase, corresponding to the flow of water as a film and in drops. For a range of pressures up to about 1 atmosphere, the first phase is the more significant so that the use of a larger number of nozzles (to produce smaller drops) becomes of secondary importance in heat transfer, where the initial temperature difference and velocity are the important parameters. Methods of arriving at the values of the various dynamical stages are demonstrated. Heat-transfer coefficients in the film phase are of a much greater order than those used in conventional tube flow arrangements, although the physical relations are the same. The drop-phase coefficient has also been measured.—*Paper by S. Weinberg, submitted to the Institution of Mechanical Engineers for written discussion, 1952.*

Gear Manufacturing by Hot Rolling

In bar stock, the "fibres" are positioned in the direction of the rolling. Therefore, when machining the teeth of a gear by shaping or hobbing, these "fibres" are severed. This is disadvantageous to the tensile strength of the workpiece. The direction of the fibres is much more advantageous in die-forged blanks, however, the machining of the teeth will cut the "fibres" here as well. These handicaps have been completely eliminated by a process which has now been perfected at the "Matyas Rakosi Works" after two years of experiments. By this process the tooth profile is pressed from both sides into the blank, heated to red heat by the gradual pressure of a formed rolling tool and by its alternate rolling. The teeth thus produced are so accurate that they require no finishing. Only half a minute —approximately one twentieth of the entire shaping time—is needed for heating and rolling the small nickel-chromium steel gears manufactured as described above. The thin oxide coating developed within this short time does not prevent case-hardening; these gears have run for thousands of kilometers in gear boxes without becoming defective. The saving in material is about 22 per cent. The manufacturing cost of a rolling machine is considerably lower than that of any gear hobber or shaper and its operation is very simple. This method appears to be advantageous for the production of other parts also, such as splined shafts, racks which have teeth only in the middle, etc.—*Hungarian Technical Abstracts, Vol. 4, No. 3, 1952; p. 87.*

Calculation of Welding Stresses

This report by R. Spronck and J. J. L. van Maanen, issued by the Centre Belge de Recherches Navales, describes tests performed during the construction of a partially welded cargo vessel and a completely welded tanker. Measurements were made of deflexions and deformations in the neighbourhood of welded joints in the bottom plating after a complete transverse joint had been welded. The measurement of local deformations at a number of places on the surfaces of plates permits the stresses to be calculated in zones which are some distance from the welds, provided that the deformations remain elastic. Where the stresses in the immediate neighbourhood of the welded joint are required, it is necessary to determine them by measuring the deformations caused by drilling small holes. The theoretical considerations underlying the determination of these stresses are first considered, and the authors then describe the actual tests performed and tabulate the results. In the partially welded cargo vessel the stresses due to welding were found to have widely varying values. It can be concluded, however, that in the neighbourhood of the weld, longitudinal stresses of the order of 20 tons per sq. in. and transverse stresses of 8 tons per sq. in. will be encountered; these stresses will decrease with increasing distance from the joint. Riveting introduces stresses which, while appreciable, are considerably lower than those due to welding. In the case of the all-welded tanker, the results indicate that high elastic stresses were present, and that the stress distribution was far from uniform. The results obtained are analysed and discussed in some detail.—*Journal, The British Shipbuilding Research Association, Vol. 7, November 1952; Abstract No. 6,760.*

ENGINEERING ABSTRACTS

Section 3. SHIPBUILDING AND MARINE ENGINEERING

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New Turbine Tankers

The turbine tanker *Caltex Liverpool* is the first of a series of seven similar vessels of about 17,000 tons d.w. ordered by Overseas Tankship (U.K.), Ltd. This series was preceded by a series of four motor tankers of about 12,000 tons d.w. The *Caltex Liverpool* was built by R. and W. Hawthorn, Leslie and Co., Ltd., Hebburn-on-Tyne, who have three similar vessels also under construction or on order. A speed of more than 15 knots was obtained over the measured mile with the ship in loaded condition. The vessel showed excellent manœuvring qualities during turning trials and the turning circle appeared to be remarkably small for a vessel of this size and speed. The principal particulars of the *Caltex Liverpool* are as follows:—

Length overall	544ft. 4in.
Length b.p.	515 feet
Breadth moulded	70 feet
Depth moulded to upper deck	39ft. 9in.
Draught	30ft. 3⅞in.
Deadweight at above draught	17,460 tons
Gross tonnage	11,814 tons
Net tonnage	6,886 tons
Cargo capacity	824,116 cu. ft
Maximum horsepower	8,200 s.h.p.
Speed in service	14 knots

The cargo space is divided into nine oil tanks, each subdivided by two longitudinal bulkheads, making a total of 27 oil cargo compartments, having a total capacity of 824,116 cu. ft. The vessel is of the single deck type with poop, short bridge and forecastle decks, with raked stem and cruiser stern. An oil fuel bunker is arranged port and starboard at the forward end of the engine room, with a cofferdam between the bunker and the cargo tank. Oil fuel bunkers with cofferdam are arranged immediately forward of the cargo tanks, with a deep tank for oil fuel or water ballast, a case cargo space and pump room between the bunkers and the fore-peak. The double bottom in the engine room is arranged mainly for feed water and the fore and after peaks for water ballast. The main pumproom is immediately forward of the main engine room, arranged between the bunkers and adjoining the cargo tanks, with an oil settling tank over the after end. The pump room contains three horizontal pumps, connected to a 12-inch piping system for loading and discharging cargo, and three vertical duplex pumps are connected to an 8-inch diameter stripping system, all with control geared to the upper deck. Full flooding carbon-dioxide fire extinguishing is provided for engine and boiler spaces and for main cargo pump room. The steering gear is of the four-ram electro-hydraulic type, with two electrically driven pumping units. The main propelling machinery consists of a single-screw geared turbine installation developing a maximum power of 8,200 s.h.p. Steam is supplied by two Yarrow watertube boilers at a pressure of 450lb. per sq. in. and a temperature of 750 deg. F. The power unit consists of one h.p. turbine of the impulse type and one l.p. turbine of the impulse reaction type. The gearing is of the double-reduction articulated type, the drive from the turbines being transmitted to the primary gears through flexible couplings, and the drive from the primary gear wheels transmitted in turn to the secondary pinions through quill shafts and flexible couplings. The astern turbine of the impulse type is incorporated in the aft end of the l.p. ahead casing, and develops 60 per cent of the service ahead power. The two boilers are of the Yarrow marine three-drum type with superheaters, air heaters and forced draught fans capable of supplying a total of 90,000lb. of steam per hour. The MeLeSco superheaters have 44 elements in each boiler, of the multi-limbed single-pass type with forged return bends. A desuperheater of the tubular type is fitted to each boiler, capable of desuperheating 30,000lb. of steam per hour for various auxiliary services. Automatic combustion control is provided.—*The Shipping World, Vol. 77, 10th December 1952; pp. 461-465.*

Gas Turbine-driven M.T.Bs

The gas turbine-driven motor torpedo boat *Bold Pathfinder* was recently placed in service by the Admiralty and a similar ship, the *Bold Pioneer,* will soon follow. The machinery installation is on new lines, comprising Metrovick gas turbines of 4,500 b.h.p. in combination with Diesel engines of 2,500 b.h.p. The turbines are developed from the Gatric type installed in M.G.B. 5559 (ex 2009). The *Bold Pathfinder,* built by Vosper, Ltd., is of the round bilge type and is 122ft. 8in. overall with a length b.p. of 117 feet and a beam of 20ft. 5in. The *Bold Pioneer,* which was constructed by J. S. White and Co., Ltd., is a hard chine boat, 121 feet overall with a length b.p. of 116ft. 3in. and a beam of 25ft. 6in. The vessels carry four 21-inch torpedo-tubes and one 4·5-inch gun and they are so designed as to be interchangeable as M.T.Bs or M.G.Bs. A complement of two officers and sixteen ratings is carried.—*The Motor Ship, Vol. 33, December 1952; p. 354.*

Brittle Strength and Transition Temperature

This paper presents the results of a study of the available test data regarding brittle strength and transition temperature from the wide plate tests sponsored by the Ship Structure Committee and affiliated programmes. The study includes (1) brittle strength and (2) transition temperature range between brittle and ductile fracture. Both of these appear to be related primarily to a variable designated herein as p, the ratio of the radius at the root of the notch to the net cross-sectional area of the specimen. The brittle strength shows a general tendency to increase with a corresponding increase in p for a considerable range, which covers all values of the variable for which data could be obtained. On the other hand, the transition temperature tends to remain practically constant for all values of p less than some critical value, whereas above this critical value transition temperature appears to drop as p increases. Some of the discrepancies that have been observed in the strength and transition temperature of small specimens relative to large specimens may arise from the influence of the parameter. In general, for geometrically similar specimens, the value of p is numerically larger for the smaller specimens. This perhaps explains why small specimen tests are in general less severe than are tests of large specimens or tests of prototype structures.—*W. C. Hoeltje and N. M. Newmark, The Welding Journal, Vol. 31, November 1952; pp. 515-s - 521-s.*

Photographic Analysis of Sprays

The study of sprays and their properties has been hampered by difficulty in obtaining trustworthy measurements of the number, sizes, and velocities of drops at various locations within a spray. Many attempts to evaluate the character of a spray have been concerned only with the distribution of the flow and give no information about the drops. An example is the patternator, which operates with cups placed at various locations in the spray. After the spray fills the cups for some period of time, the contents of each cup can be measured to evaluate the flow rate at that location. Other methods of analysis attempt to measure the number and size of the drops of a spray. Most of these depend on physical sampling of the spray or on the scattering of light by the spray. Physical sampling is usually accomplished by having drops impinge on cups or microscope slides in the spray or by sucking them out of the spray through a tube. The captured sample can then be analyzed leisurely for number and size of drops. These procedures are untrustworthy because it is questionable if the sample analyzed is representative of the original spray. The photographic method of spray analysis described in this paper was developed to give spray measurements which are not biased by physical sampling and which are susceptible to direct interpretation in terms of number, sizes, and velocities of the drops. In the photographic technique of spray analysis, pictures are taken of a small volume in the spray without disturbing the flow pattern with any objects. The drop images on the film are measured and counted, giving the size distribution of drops in the small volume of the spray (the spatial distribution). For most applications, however, the information desired is the size distribution of drops passing a cross-sectional area in a unit time (the temporal distribution). Since the temporal distribution is the product of the spatial distribution and the average velocity of drops of each size, it is necessary to measure drop velocities to obtain the temporal distribution. Therefore double exposures are taken of regions in the spray with a small known interval between exposures. The resulting photographs show a pair of images for each drop, one image made at each exposure. The velocity of each drop can be calculated from the distance between images and the interval between exposures.—*J. L. York and H. E. Stubbs, Transactions A.S.M.E., Vol. 74, October 1952; pp. 1,157-1,162.*

Amyl Nitrate

The U.S. Navy is conducting tests on a compound designed to improve Diesel fuel. The new chemical compound is designed to raise the quality of Diesel fuel to Navy requirements without the refining now necessary. The compound is a result of four years of research by the Ethyl Corporation. It is a blend of several nitrates found to be the most economical and practical as a quality booster of Diesel fuel. Small amounts of the blend added to Diesel fuel will improve its quality, causing it to ignite more readily in the combustion chamber of an engine. The compound consists of a mixture of primary amyl nitrates which is blended into the finished fuel. As little as one-tenth of one per cent by volume is sufficient to bring many distillate heating oils within the cetane number range of commercial Diesel fuels. Moreover, it is effective in improving the cetane number of all types of Diesel fuels, regardless of the crude source, refining technique or sulphur content. Ethyl's ignition improver promises to be of great assistance to the oil industry in meeting a demand for Diesel fuel that has expanded four-fold since 1941 and is expected to double again by 1960.—*Diesel Progress, November 1952; p. 86.*

New Bearing Alloys

Work at the Fulmer Research Institute has led to the development of an important new series of aluminium/tin bearing alloys. Aluminium/tin alloys, when cast, form a continuous film round the grain; this renders them very brittle at the temperatures at which they are normally used—say, 120 to 140 deg. C. In the past, the tin content of aluminium alloys has therefore been limited to less than 10 per cent, although it was assumed that the anti-friction properties of alloys of this type would improve with the increase in tin content. It has been shown that by a process of cold working followed by recrystallization heat treatment, the tin can be redistributed in a dispersed form with great improvement in mechanical properties at the slightly elevated temperature at which bearings operate. Alloys containing 30 per cent of tin have been developed by this method, and bearing tests by the Tin Research Institute have shown that they have high fatigue resistance. Promising results have been reported on engine trials with the new materials.—*Gas and Oil Power, Vol. 47, November 1952; pp. 267-268.*

Hot Machining of High-alloy Materials

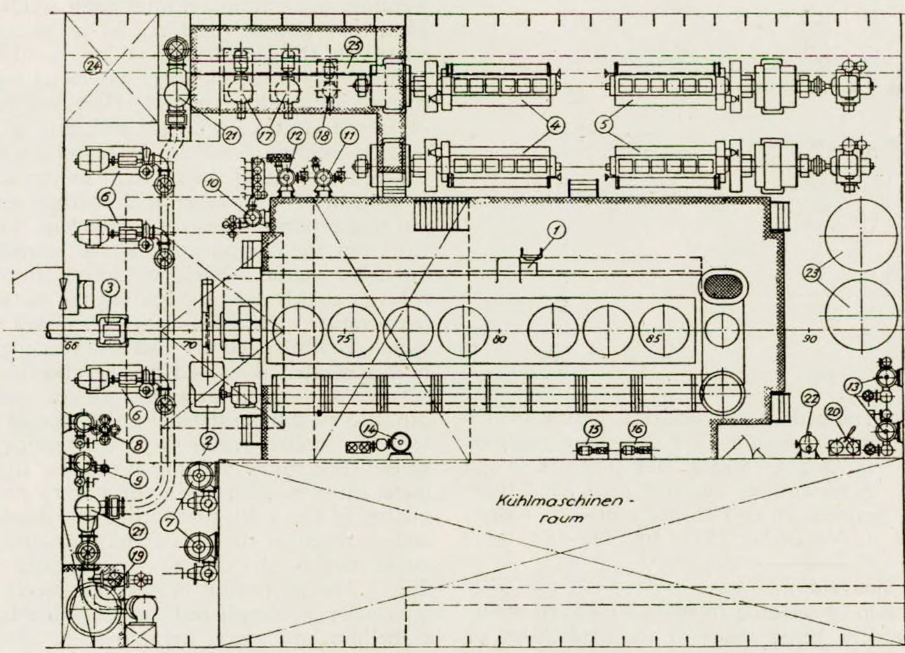
The article by A. A. Caminada published in the journal "Materials and Methods" draws attention to the fact that interest in hot-machining methods has been much stimulated in recent years by problems encountered in machining high-temperature alloys and other refractory materials. Several techniques have been developed, each of them having its own advantages and limitations. The features of the respective processes are critically discussed in this article, in mutual relation, and in relation to cold-machining procedure. Hot-machining procedures are basically of two types: in one the entire piece is heated, whereas in the other only the spot immediately in front of the tool is preheated, and this area is removed as quickly as possible. These two basic methods have been applied chiefly in the following ways: (1) by machining the work while it is still hot from other manufacturing processes; (2) by preheating the work and machining before it cools, meanwhile insulating the holding device to prevent excessive loss of heat; (3) by heating the workpiece on the machine immediately ahead of the tool by induction, with a flame, or with an electric arc; (4) by surface heating the workpiece in advance of the tool. The first procedure is used in steel mills, for example, for hot sawing of steel billets: the others have been used for machining of parts in the shop. Experience gained in the use of the respective methods is described by the author, and the discussion is illustrated by diagrams showing results obtainable by hot machining of stainless nickel-chromium steel and other materials. Although investigation of the potential usefulness of hot machining is not yet complete, it is already clear that the method has advantages in handling materials which give difficulty in cold machining, that a good finish is obtained on hot-machined work, and that for some materials the procedure is an economic proposition.—*The Nickel Bulletin, Vol. 25, October 1952; p. 242.*

Refrigerated Cargo Vessel

The refrigerated cargo vessel *Alstertor* is the third post-war vessel of this type built by the Deutsche Werft. The leading particulars are: Length o.a. 390ft. 11⅝in.; length b.p. 357 feet; breadth moulded 46ft. 11in.; summer draught 20ft. 0⅜in.; 3,150 tons d.w.; speed 16½ knots. The vessel is of the open shelter deck type. The hull is largely welded; as are also the rudder and the stern frame. The propulsion plant consists of a seven cylinder direct-reversing single-acting two-stroke M.A.N. Diesel engine of the crosshead type. The cylinder diameter is 700 mm. and stroke is 1,200 mm. At 125 r.p.m. the engine develops 4,620 h.p., the m.i.p. being 6·25 atm. Minimum speed approximates to 40 r.p.m. The refrigerating plant is so designed that temperatures of 10 to 12 deg. C. can be maintained for banana and − 12 deg. C. for meat cargo. There are three four-cylinder NH_3 compressors with a total capacity of 510,000 kcal. per hour, each compressor being driven at 280/350 r.p.m. by an electric motor of 82/110 h.p. having a speed of 1,100/1,450 r.p.m.—*Schiff und Hafen, Vol. 4, November 1952, pp. 423-429.*

Fluid Piston Type Bearings

Several novel types of piston pump bearings have been introduced of late, such as, for instance, the Gerard bearing, which has already been applied with success to grinding machines. In fact, application of these bearings is rapidly increasing owing to their inherent extraordinary advantages in certain cases. These bearings can be lubricated by the fluid being pumped, although it may have no lubricating qualities in the usual sense and may even be hot and corrosive. Moreover, materials may be used which would not be suitable for conventional bearings. Also, metal-to-metal contact for short periods can be tolerated and foreign particles which score the bearing will not necessarily affect its operation. In fact, the bearing surface does not need to be particularly smooth. In



Engine room of m.v. Alstertor

1. Main engine
2. Control board
3. Bearing
4. Two Diesel driven dynamos
5. Two Diesel driven dynamo-compressors
6. Cooling water pump (salt water)
7. Cooling water pump (fresh water)
8. Ballast pump
9. Bilge pump
10. Deck washing and fire pump
11. Cooling water pump (salt water) for use in port
12. Cooling water pump (fresh water) for use in port
13. Two lubricating oil pumps
14. Diesel fuel transfer pump
15. Sanitary pump
16. Fresh water pump
17. Two Diesel fuel centrifuges
18. Two lubricating oil centrifuges
19. Bilge oil-separator
20. Lubricating oil filter
21. Salt water strainer
22. Lubricating oil cooler
23. Starting air bottles
24. Diesel fuel day tank
25. Switchboard

the case of pumps which have to handle corrosive liquids or liquid metals, considerations of leak-proof design make it imperative that the bearing should operate in the liquid which is being pumped. This means that the bearing must be lubricated with fluids which may be low in viscosity, having very poor bearing qualities itself as it has to be selected exclusively from the aspect of corrosion resistance. Latest applications include radial bearings as well as double-acting thrust bearings. The radial bearing consists essentially of three or more pockets of pistons spaced around the circumference of the shaft. Usually, four such pistons are used, with each piston covering a little less than 90 degrees of arc, leaving some space between adjacent pistons. Each piston is a shallow rectangular box with an open end facing the shaft. Fluid under pressure is passed through an orifice into each box and exerts a thrust upon the shaft, holding it centrally in the bearing.—*The Engineers' Digest, Vol. 13, November 1952; pp. 370-371.*

Closed-circuit Hot Air Turbine

Hot air turbines of Escher Wyss design are now being developed in the U.S.A. by a company especially formed for this purpose. As will be seen from Fig. 7, in small plants of 3,000 h.p. the compressor and the air turbine are arranged on a common shaft. This design can be used for powers up to 6,000 h.p. The unit shown in Fig. 7 has an overall length of

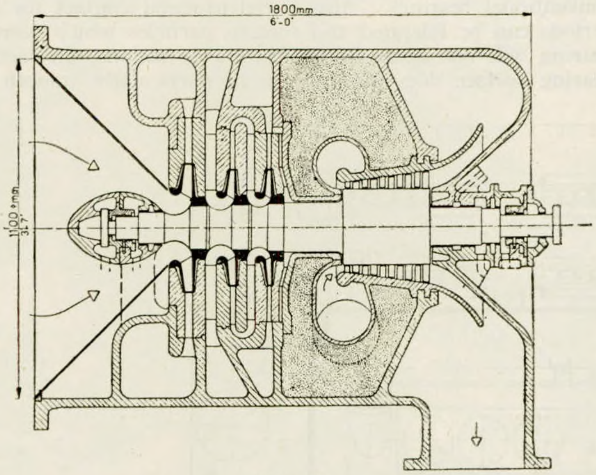


FIG. 7

only 1·1 metres and is arranged in a cylindrical casing of 1·8 metres diameter. This unit is designed for throttle conditions of 30 atm. and 650 to 700 deg. C. and a back pressure in the closed cycle of 6 atm. According to the fuel used, etc., thermal efficiency will vary between 26 and 28 per cent.—*C. Keller, Schiff und Hafen, Vol. 4, November 1952; pp. 459-463.*

Radiography in a Clyde Shipyard

With the introduction of welding to main structural members, the Clyde shipbuilders, being aware of the inadequacy of visual inspection of weld deposits, formed a committee to examine the possibilities of radiography as a means of routine weld inspection. Although quick to realize the potential value of this method, the shipbuilders were doubtful as to the suitability of existing appliances for site examinations. Such units as were available were heavy and bulky, and this necessitated the constant use of cranage and the erection of special staging and platforms for shell work. Improved units, manufactured with the requirements of the shipyards in mind, became available and about five years ago the Clyde Shipbuilders' Association embarked upon a scheme whereby radiographic inspection would be conducted on an organized regional basis employing three inspection units, namely: 1. A 90-kV. X-ray unit for the examination of plate thicknesses up to ½ inch. 2. A 140-kV. X-ray unit for the examination of plate thicknesses up to 1½ inches. 3. A 250-mg. radium source for the examination of heavier sections. Each unit is operated by an expert and is controlled from a parent shipyard, being available to members when and where required. The scheme, which works smoothly and well, is obviously of a temporary nature; already a few of the original members possess their own equipment and it is anticipated as the amount of welding increases that individual builders will acquire their own facilities. The firm with which the author is associated is responsible to the Association for the operation and control of the 90-kV. set and the author explains the function of this unit in the radiographic control of welding within that firm. Recent developments in the field of radiographic inspection due to the availability of artificially created radioactive isotopes are discussed and there is a comparison drawn between the results obtained using X-rays and gamma-rays with particular reference to plate thicknesses of ½ to 1½ inches.—*Paper by E. J. Duffy, read at a meeting of the Institution of Engineers and Shipbuilders in Scotland, 18th November 1952.*

Welding Processes

A knowledge of the inherent characteristics differentiating the various welding processes is important to the welding engineer so that the most suitable welding process can be selected. The various welding processes differ according to (1) Source of heating; (2) Type of shielding; (3) Application of pressure; (4) Addition of filler metal; (5) Completion of the joint; (6) Preparation for welding; and (7) Practical aspects of inspection. The source of heating characterizes many of the welding processes. Heat from electric power is created by resistance to flow of electric current. This is represented by spot-welding, seam welding, projection welding, flash welding, and upset welding. Heat may also be produced by an electric arc. This is represented in the shielded metal arc welding process, the atomic hydrogen process, the inert gas metal arc process, and the submerged arc welding process. Heat may be created by a discharge of stored electrical energy, and this is illustrated by the percussion welding process. A second source of heat may be from the combustion of gases, such as oxygen with acetylene, with hydrogen, or with other hydrocarbon gases. The oxyacetlyene welding process represents this group. Heat may also be generated from the combustion of coal, carbon, or other solid combustible material. Forge welding is an example where charcoal, coke, or coal is used for heating the metal parts up to a welding temperature. Heat may also be created by a chemical reaction, which is illustrated by thermit welding. In this process the heat generated is caused by the chemical reaction taking place between aluminium powder and iron oxide. Welding processes can also be differentiated by the type of shielding used to protect the molten metal from contamination with atmospheric gases. For many welding processes, shielding of the hot weld metal must be adequately effective to preserve the metallurgical quality of the weld, preventing any deleterious action of oxygen and nitrogen in the air combining with the weld metal. Weld metal may be shielded by gases, liquid flux, or solid granular flux. The protection of the hot metal against contamination is usually accomplished by gas shielding. Inert gases such as helium and argon are extremely effective in protecting the molten metal from oxygen and nitrogen in the atmosphere. Reducing gases such as hydrogen and carbon monoxide can also be effectively used to prevent oxidation and nitriding of the molten metal. A good example of reducing gases is the atomic hydrogen process, which is used for welding stainless steel, aluminium, and many other metals where protection against oxygen and nitrogen is extremely important. In the bare metal arc welding process, no special gas for shielding is used; and molten metal is exposed to the atmosphere. Another method of shielding is by liquid fluxes. In the oxyacetylene brazing process, the flux is applied as a solid material and is converted to a liquid by the heat of the welding process. This liquid on top of the molten metal shields the metal against contamination from the atmosphere. Still another method of shielding is by the use of solid granular flux material. The process which uses this type of shielding, and is widely used, is known as the submerged arc welding process. In this process the oxygen and nitrogen of the air are excluded by the granular flux surrounding the molten metal. In the arc zone the granular flux becomes a molten blanket covering the hot weld metal and protects it from the oxygen and nitrogen of the air. Welding processes also are characterized by the application of pressure. Processes differ according to the method of pressure application. In some processes pressure is an important factor in the welding operation, while in others pressure is not required. Flash welding is a process in which pressure is of prime importance. Processes which do not require pressure are, in general, the arc welding processes such as the shielded metal arc, inert gas metal arc, and submerged arc. All of the resistance welding processes require pressure. The success and quality of the weld will depend largely on the application of the correct amount of pressure at the proper time in the welding cycle. In some processes pressure is applied as a single blow, and in still others, the application is constant.—*J. J. Chyle, S.A.E. Journal, Vol. 60, October 1952; pp. 30-39.*

Embrittlement of Heat Resisting Steel

The phenomenon of embrittlement of steels due to heating at moderately high temperatures for extended periods of time is well known. The fact that the addition of stress can accelerate this condition is a more recent discovery. Data on this latter effect are not available in sufficient quantity to permit formulating exact conclusions on the effects of stress, temperature, and time on the embrittlement of alloy steels. The embrittlement of heat-resisting steels when subjected to long-time heating was first evidenced by the failure of boiler flange bolts. Although such bolts are generally designed with large factors of safety based on short time, high-temperature properties, premature failures are frequently encountered. These failures are normally at regions of stress concentration after operation at temperatures of 500 deg. C. for 3,000 to 12,000 hours. The retained properties of broken bolts at room temperatures can be determined best by a notch impact test. In most cases, the impact energy of the entire bolt is reduced considerably after only a few hours at high temperature. The steels most readily embrittled by temperature and stress are chromium-nickel-molybdenum and chromium-nickel steels. The Cr-Mo steels are relatively free from embrittlement effects. The impact properties of the Cr-Ni-Mo steels can be greatly increased by a reheat treatment. The major portion of the embrittlement, when it is large, results from heating alone. Furthermore, a steel not embrittled by heating is also not significantly affected by the addition of cyclic stresses. Creep damage for annealed Cr-Ni-Mo steel is large compared to the embrittlement due to the heating alone. The hardness and retained impact properties of stress-rupture specimens embrittled at 932 deg. F. were determined before and after reheat treatment for four steels. Results indicate that the hardness remains practically unchanged, except for the Cr-Ni-Mo steel, which exhibits a continuous decrease. The impact strength reveals a progressive embrittlement and then a recovery after sufficiently long times to rupture. The retained impact and tensile properties of stainless steel at room temperature after creep has been determined to illustrate the per cent reduction in impact strength and permanent damage as a function of creep time with creep stress as a parameter. The major conclusions that can be derived from these data are: (1) the onset of damage is shifted to longer times with decrease in the creep stress; and (2) permanent damage is developed only if a certain value of creep damage is exceeded.—*Paper by G. Sachs and W. F. Brown. Paper No. 74, The American Society for Testing Materials, 1952.*

Automatic Helmsman

The United Kingdom manufactures two automatic steersmen, either of which will steer a ship more accurately than the human steersman. Both makers manufacture a number of variations to suit the particular steering gear with which a ship may be fitted, and each type is designed to follow a gyro compass. The gyro compass itself, if it is to be accurate, must be arranged so as to have extremely little friction in its gimbal bearings, so it is usual for some kind of hunter or follow-up motor to be employed to drive the gimbal ring in its bearings, thus relieving the gyro itself of all except the very light load required to operate the follow-up mechanism. The Brown gyro compass uses an air jet to switch the follow-up motor, while the Sperry uses electric contacts to perform the same function. The Brown equipment now uses a thermionic amplifier to reduce the wear and tear of the electric contacts operated by the air vane. The Brown pilot is arranged so that the rudder always lies in a predetermined position to port or to starboard. The rudder angles are chosen to suit the ship and the prevailing weather, and are normally set so as to give

Fig. 5. CYCLE OF OPERATION OF BROWN AUTOMATIC HELMSMAN.

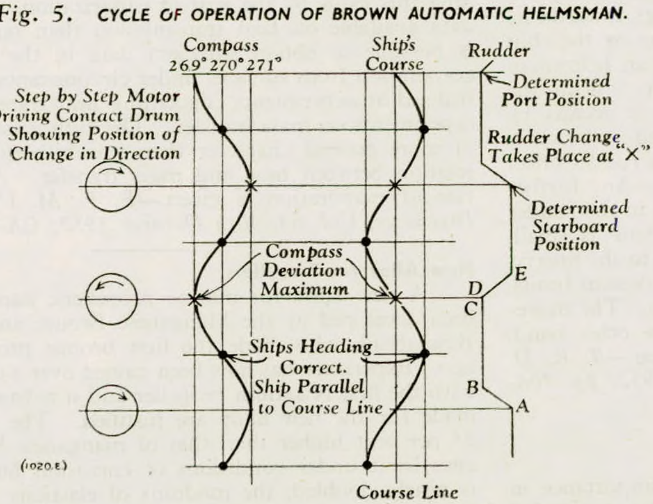
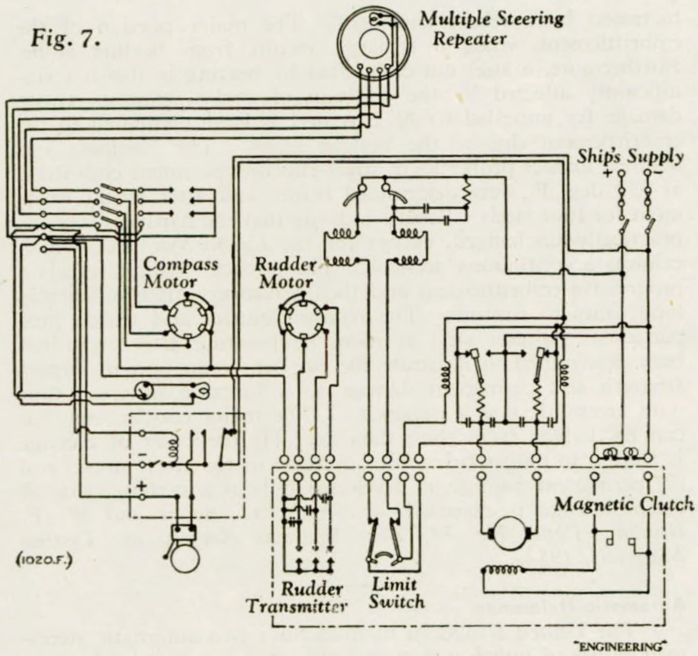


Fig. 6.

"ENGINEERING"

Fig. 7.



the least motion of the steering gear consistent with a reasonably steady course. When a ship is rolling or pitching the angular movement of the compass gimbals causes geometric errors, so that the lubber's lines appear to move to and fro. Clearly it is not desirable that, every time the ship rolls, the rudder should be reversed, so a dead motion is provided which will allow the ship (or the lubber's line) to yaw a certain amount before the rudder is applied. The Brown helmsman uses electrical contacts to control the rotation of an electric motor which is geared either to the ship's steering wheel or replaces the ship's telemotor system. The action of the Brown automatic helmsman is shown in Figs. 5 and 6, the wiring diagram in Fig. 7. Such a system will hunt continuously, but the determined rudder angle being fairly small, this is acceptable. The system may be made much more stable by means of a feedback from the actual rudder position, which is combined with the deviation from true course to operate the contact drum. This feedback has the effect of advancing the instant of release of the rudder as the ship swings towards its course and of hastening the instant of application as the ship yaws away from its course; it thus acts as a human helmsman should when he "meets" the swing of the ship. With the feedback added there are now two reset loops or means by which the contacts may be brought central, and it becomes possible for the contact to become central with the rudder offset at some angle less than the pre-determined limits. Any further yaw of the ship will be balanced by small changes in the rudder position. In this manner, therefore, when the yaw is small, the servo will behave as a continuous type, similar to the Sperry, but with larger yaws, where the rudder reaches its present limits, it will function as an on-off or "bang-bang" servo. The movement of the rudder in the Sperry system, on the other hand, is limited only by the limits of the steering gear.—*J. R. D. Walker, Engineering, Vol. 174, 28th November 1952; pp. 706-708.*

Instruments and Techniques in Vibration Research

Vibration testing has become of very great importance in all aspects of modern engineering. The method of test may fall into one of several categories, namely: (1) Excitation of a model of the final structure to determine its critical frequencies and resultant amplitudes. (2) Artificial excitation of the final structure to confirm that amplitudes without specified maxima are not produced. (3) Measurement of the vibration amplitudes over the structure under normal operating conditions to confirm that these amplitudes are under the specified maxima. A vibration test on a model is comparatively easy and has the advantage that it may be performed before the full-size structure is built. The application of scale factors and the translation of the results to apply to the full-scale structure may be difficult. On the other hand, the artificial excitation of a large structure requires proportionately large input powers and correspondingly expensive apparatus. For these reasons full-size testing has tended in the past to have been of limited application, except so far as measurements were taken over the normal range of operating conditions of the possible vibration-producing units, such as motors, gears, propellers, etc., installed in the structure. The aircraft industry has possibly been the most progressive in the development of comprehensive vibration testing, but the specification of vibration limits and tests is now common in many other industries. The paper discusses methods of producing and measuring vibrations and makes particular mention of equipment developed for a research project involving the vibration testing of ship models up to 20 feet in length, in air and in water.—*Paper by D. S. Gordon, read at a meeting of the Institution of Engineers and Shipbuilders in Scotland, 12th December 1952.*

Evaporation from Water Surface into Air Stream

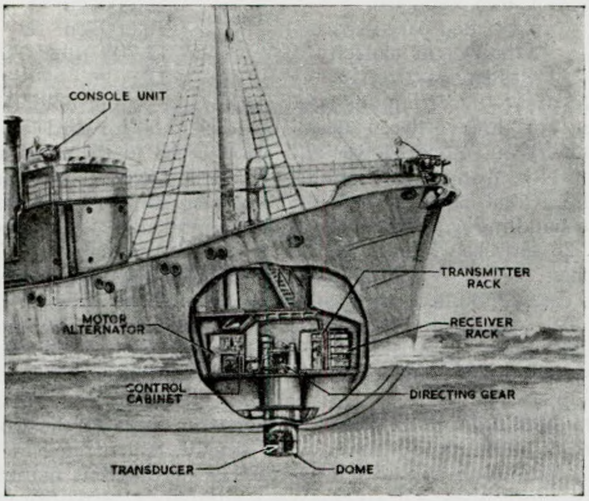
When a humid surface is brought into contact with air not fully saturated with water vapour, evaporation, or condensation will take place according to the temperature of the surface lying above or under the dewpoint. With evaporation heat is withdrawn from the surroundings and with condensation heat is released. If the heat necessary for the evaporation under stationary conditions is supplied mainly by the air, the surface, in the long run, will reach a temperature lying between the dewpoint of the air and the initial temperature of the air. In the case of an airstream rapidly flowing along a surface, the heat exchange with that surface is so intense that heat, supplied unintentionally in other ways, may be neglected. The difference in temperature between air and surface in this case is a measure for the humidity of the air. It has been possible to show experimentally that above a certain air velocity the self-adjusting temperature of the surface is independent of the air velocity. The operation of a good psychrometer is based on this fact. The air temperature (dry bulb temperature) and the barometer reading are measured. If the value of the heat transmission of the air to the surface is known, it is possible to calculate the quantity of evaporating water directly by dividing this value by the heat of vaporization. As there are more data available on heat transmission than on mass transfer, it is possible to obtain sufficient data in the way indicated on evaporation from surfaces under circumstances which have been realized at experiments concerning heat transfer but not yet at experiments on mass transfer. The author establishes a formula of more general character than was hitherto available for the relation between heat and mass transfer. A formula for the rate of evaporation is given.—*E. F. M. Van Der Held, De Ingenieur, Vol. 64, 10th October 1952; Ch. 89-94.*

New Alloy for Propellers

A new alloy for marine propellers, named Nikalium, has been developed by the Manganese Bronze and Brass Co., Ltd., Birkenhead, who made the first bronze propeller many years ago. Experience has now been gained over a considerable period with the first Nikalium propeller and it is found that the claims made for the new alloy are justified. The tensile strength is 25 per cent higher than that of manganese bronze, the endurance limit under conditions of corrosion fatigue in sea water is nearly doubled, the modulus of elasticity is higher and the specific gravity considerably lower. Variations in the mechanical properties throughout a large Nikalium propeller are much less pronounced than those of a manganese-bronze propeller, since the material is not nearly so sensitive to section effect. Experience over several years has shown that exposure to the corrosive influence of sea water has practically no effect upon Nikalium, and, as it contains no zinc it cannot suffer from dezincification. Repair by electric welding is easily carried out on Nikalium propellers, while the fusion burning process may also be satisfactorily employed. Nikalium is welded with Nikalium rods, so that similar mechanical properties and resistance to corrosion are achieved in the weld and parent metal.—*The Motor Ship, Vol. 33, January 1953; p. 442.*

Echo Finder for Whaling

An echo whale-finder which has been brought to the commercial stage after some years of practical trials, has recently been demonstrated. The apparatus is a type of echo-sounder, employing what is known as an electro-acoustic transducer mounted in a streamlined, sound-conducting dome, projecting below the keel of the whale-catching vessel. The dome can be retracted when not in use. The transducer is trained from a console on the bridge and a repeater from the ship's gyro-compass, incorporated in the training control unit, automatically stabilizes the transducer in azimuth to the last bearing set by the operator. Once the contact with the whale has been established, changes of course do not throw the beam off the target. The equipment has been tested under operational conditions in the South Atlantic during the past three whaling seasons in co-operation with Chr. Salvesen and Company, Leith. By using the new device, the gunner of the whale-catcher is provided with continuous information of the whale's position below the surface and he can close for a shot in the shortest possible time.—*The Motor Ship, Vol. 33, January 1953; p. 411.*



Kelvin Hughes echo whale-finder, shown ready for operation

Steering Ships

From any number of preselected steering stations, the "hand electric" system provides rudder control. Its operational advantage over other systems is the reduction of steering effort required, plus its accuracy and reliability. In essence the system consists of a remotely controlled servo-mechanism that positions the steering engine stroke, or control, shaft in accordance with helm order. The helm order is transmitted by a selsyn generator (to which the steering wheel or knob is geared) over a five-wire cable to the steering-gear room. Here the helm receiver—a "matching" control transformer—produces an electrical error signal proportional to helm order. This signal is magnetically amplified and controls a stroking motor that determines the amount and direction of rudder called for by the steering engine. As the motor strokes the steering engine, it also turns the rotor of the helm receiver so that the error signal is reduced to zero precisely when the correct amount of rudder is called for. Thereupon it stops running and holds that position until a new helm order is initiated. To go one step further an elementary course-holding feature, "basic auto-steering", is added to the system. This is done through use of a differential selsyn generator—one that produces a signal proportional to the difference between its own rotor position and that of an input generator. A gyrocompass provides a reference axis and the excitation for the differential generator, and it in turn signals the helm receiver for rudder proportional to course error. The helm receiver then controls the rudder position exactly as before. Only there is this difference: the gyro is controlling the rudder through differential generator—the human helmsman is not necessary. An emergency override switch permits instant return to manual control. The more elaborate "advanced automatic" system utilizes a modified helm transmitter such that the helm angle specified by it is proportional to heading error and rate of change of this error. Affording rudder "anticipation" is the function of the rate of change signal. It prevents overshooting the course or oscillation of the ship. Other features of the advanced system include "controlled manœuvring", which is obtained through use of adjustable rudder limits. Maximum rudder angle that the auto-pilot will call for is set by the helmsman.—*A. D. Hammes, General Electric Review, Vol. 55, November 1952; pp. 37-40.*

Turbine Installation for 600lb. per sq. in. and 950 deg. F.

The cargo liner *Nestor* is the first of three vessels building for the Ocean Steamship Company, Liverpool (managers, Alfred Holt and Company). Built for the Australian trade of her owners, the *Nestor* has the length of 452ft. 9in. between perpendiculars, a moulded breadth of 64ft. and a depth of 35ft. 3in. Her gross tonnage is 7,800 tons and her deadweight 9,565 tons. The machinery of the *Nestor* and her sisters departs in a number of respects from the conventional forms of steam turbine installations generally adopted at sea. The initial steam pressure and temperature are higher, a new type of boiler is introduced, steam is not bled from the turbines for feed heating, and atomizing pressure of the boiler fuel is much higher than that usually encountered in merchant service practice. The turbines are designed to operate at 600lb. sq. in. g., 950 deg. F. exhausting to 28½in. vacuum with cooling water at 75 deg. F. The maximum power is 8,000 s.h.p. at 112 r.p.m., with an economical power of 7,250 s.h.p. For astern operation the steam is reduced in temperature to 750 deg. F. maximum. The turbines are of the three cylinder type, the l.p. turbine housing the astern stages which will develop 3,500 s.h.p. The top half h.p. cylinder casing is of cast molybdenum vanadium steel, and contains a velocity 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. Cast carbon steel is used for the i.p. cylinder which 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 l.p. 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 is 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. Admission to the h.p. cylinder is 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. Blading is of stainless steel throughout, all blading and nozzles being fully machined except the diaphragm blades of the final l.p. stages. The condenser, which has a fabricated shell, has a cooling surface of 6,170 sq. ft. The gears which were also manufactured 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 fabricated. Incorporated in the design is an automatic turning gear which enables the turbines to be rotated by means of a turning motor. The turning gear rotates the propeller shaft once every ten minutes. It is fitted with an interlock system which prevents the engagement of the gear while the turbines are rotating, and conversely prevents the admission of steam to either the ahead or astern turbines while the turning gear is engaged. This feature is of particular value during prolonged standby. Steam is supplied by two Foster Wheeler boilers constructed by David Rowan and Company, Glasgow. The design is new, being a single furnace boiler with the superheat controlled by attemperator and reduced for astern operation by dampers. A total of 14 electrically operated Clyde soot blowers are fitted for use at sea. The closed feed system departs slightly from current practice in that no bled steam is taken from the main turbines for feed heating, sufficient steam being available from a back pressure turbo generator of W. H. Allen's design and manufacture to provide all auxiliary services and feed heating up to a final temperature of 245 deg. F. in the deaerating feed heater of Hick Hargreaves design and manufacture. The motor driven main feed pump is of Worthington Simpson design and manufacture. The pump and the turbo generator are also noteworthy for the Allen-Stoeckicht epicyclic gears with which they are equipped. For standby purposes, a Weir main feed pump with a capacity of 70,700lb. per hr. at 800lb. per sq. in. is provided.—*Shipbuilding and Shipping Record, Vol. 80, 20th November 1952; pp. 667-669.*

Propulsion Plant of Holt Liner

The two Foster Wheeler boilers fitted in the Holt liner *Nestor* operate at a pressure of 648lb. per sq. in. at 950 deg. F. The normal evaporation is 25,000lb. per hr. which can be raised to a maximum of 30,000lb. per hr. Three-stage superheating is carried out by two MeLeSco superheaters built by The Superheater Co., Ltd., in the first two stages, and a final stage superheater, which was built by the boiler contractors, David Rowan and Co., Ltd., Glasgow. In the first two stages the steam temperature is raised to 735 deg. F., and in the third stage it is raised to the operating temperature of 950 deg. F. The superheater is of the single-pass type with multilimbed elements and "Concen" metal-to-metal ball joints. Only two headers are employed, these being the inlet to the first section and outlet from the second section. Between the primary and secondary an arrangement which saves two headers is employed; this consists of a studded steel plate with cone seats on either side, which forms the connexion between the outlets of the primary and inlets of the secondary superheater elements. The elements in this instance are made from a prefabricated continuous tube, having the return bends made from the tube itself without welding, by the MeLeSco forged "U" bend process developed by The Superheater Co., Ltd. They are produced in such a way that the wall at the outsides of the bends, irrespective of the radius, is equivalent to the tube gauge, and at the same time the full area is maintained; in other words, there is no deformation of the tubes. —*The Shipping World, Vol. 127, 17th December 1952; p. 478.*

Mariner-class Steamships in Service

The *Keystone Mariner* called at Bordeaux last month on the return leg of her first transatlantic round voyage, having come from Bremerhaven via La Pallice, and the *Old Colony Mariner* visited the port of Liverpool. Five of this new type (C4-S-1A) of fast strategic cargo vessels, are being built by the seven leading American shipbuilding companies. They are being built for the United States Maritime Administration and leased to private companies for operation. The Bethlehem Steel Company acted as design agents for the Administration and prepared specifications, contracts and detailed working plans, etc., for the entire class, supplying these particulars to the other shipyards. The new design provides a type of vessel essential for national defence and useful in normal commercial service. The possibility of future conversion for use as naval auxiliaries has also been borne in mind, and the basic cargo vessel design can be adapted for passenger carrying. A recent report states that these vessels are to have reinforced deck sections for gun platforms, an area from which helicopters can be launched and recovered, and provision for the installation of torpedo launching equipment. Admiral Cochrane, chairman of the U.S.M.A., is stated to have said, following the successful trials of the *Keystone Mariner,* that "her speed without cargo was so far in excess of 20 knots as to ensure that she will be equal to this speed while fully loaded. She displayed remarkable fuel economy whilst steaming at 17,500 s.h.p.". The *Keystone Mariner* has been built by the Sun Shipbuilding and Dry Dock Company, Chester, Pas., and the *Old Colony Mariner* by the Quincy yard of the Bethlehem Steel Company. The *Mariners* have the following principal particulars:—

Length overall	563ft. 8in.
Length between perpendiculars	528ft. 0in.
Moulded breadth	76ft. 0in.
Moulded depth, to main deck	44ft. 6in.
Draught, fully loaded... ...	31ft. 0in.
Net tonnage	5,367 tons
Gross tonnage	9,214 tons
Cargo capacity...	767,000 cu. ft.
Insulated cargo space	30,000 cu. ft.
Deadweight capacity	13,700 tons
Passengers	12

There are two continuous steel decks and seven holds with upper and lower 'tween decks on the four forward of the machinery and upper, lower and orlop decks in the three after ones. Nos. 1 and 7 lower holds are in the form of deep tanks, having a total capacity of 1,040 tons. An impressive outfit of cargo-handling equipment is installed and the 14 kingposts carry 26 derricks which range from 5 to 60 tons in capacity. The winches, of the Westinghouse adjustable-voltage type, are, in most cases, mounted on masthouses clear of the decks. Topping lift winches for each derrick are fitted to the kingposts. The machinery consists of a single-screw set of two-casing cross-compound double-reduction geared turbines, designed to develop 17,500 s.h.p. at 102 r.p.m. The *Mariners* are thus the highest-powered single-screw merchant ships in the world. Steam at 600lb. per sq. in. and 865 deg. F. is generated in two watertube boilers supplied by a number of different manufacturers. Three-phase, 450 volts, 60 cycle alternating current supplies are maintained for most purposes on board.—*The Marine Engineer and Naval Architect, Vol. 75, December 1952; p. 553.*

New Tanker

The motor tanker *Ashtarak,* which was delivered by the Swedish shipyard of Kockums mek. Verkstads A/B, Malmo, to Les Petroles d'Outre-Mer, S.A., of Paris, is the first of a new type of 19,000-tons deadweight standard tanker which this yard has started to build, and of which another seven ships are on order for various owners. The principal dimensions of the *Ashtarak* are as follows:—

Length overall...	556ft. 4in.
Length b.p.	525 feet
Breadth moulded	72 feet
Depth moulded	40ft. 1in.
Draught fully loaded	30ft. 4in.
Deadweight capacity	19,610 tons
Tonnage, gross...	12,828 tons
Tonnage, net	7,549 tons

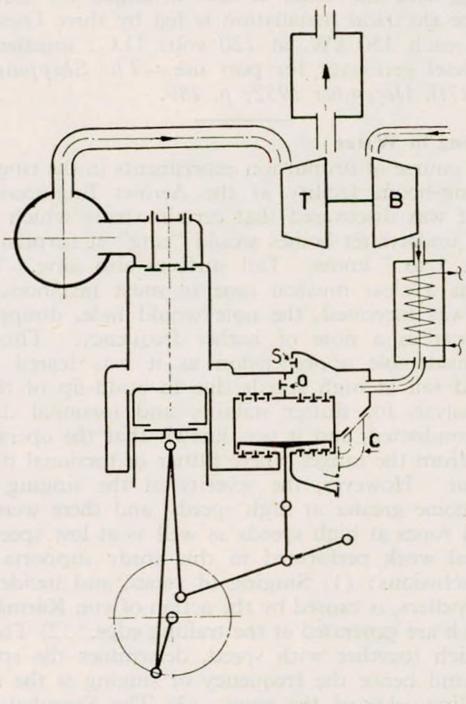
A feature of the ship is the electrical installation. In conformity with normal French tanker practice, this was specified by the owners for three-phase current at 380 volts and 50 cycles per second. The shipyard is now to standardize on A.C. electrical installations for all tankers, instead of the D.C. installations previously used. The main engine is an 8-cylinder two-stroke double-acting Kockum-M.A.N. Diesel engine, with an output of about 8,000 b.h.p. at 110 r.p.m. This gives the ship a speed of 15¼ knots.—*The Shipping World, Vol. 127, 31st December 1952; p. 513.*

First Atomic Ship

In America it is planned to build two atomic-powered submarines of 2,500 tons displacement, and the first of these, the *Nautilus,* was laid down at the yard of the Electric Boat Company last June, the anticipated launching date being 1954. A second similar vessel is stated to be in prospect, but whereas the *Nautilus* will have Westinghouse plant, the General Electric Company will be responsible for the second ship. The cost of the machinery in the *Nautilus* will, it is thought, be in the neighbourhood of 25 million dollars and the total expenditure 40 million dollars. The method employed for the propulsion of the *Nautilus* will involve the installation of an atomic pile, in which, through nuclear fission, heat will be supplied and converted into steam, the steam being then utilized to drive a steam turbine coupled to a generator which supplies the power to the propelling motors in the normal way. The atomic pile will be of moderate size, shielded by walls of lead several feet thick, so as to protect the engine room personnel. The reaction will be controlled by rods extending into the pile. When they are withdrawn from the pile they no longer absorb the flying neutrons and the chain reaction begins, producing large quantities of heat. The pile is not in itself a heat exchanger, and two vertical heat exchangers are indicated supplying steam to the turbine. A coolant will be circulated through the pile for this purpose and it may be that mercury will be employed. If this is so, no doubt as in land installations, the mercury vapour, after passing through a mercury turbine, will be condensed, steam raised and supplied to a steam turbine, the mercury and steam turbines being geared together to drive the generator.—*The Motor Ship, Vol. 33, January 1953; p. 395.*

Werkspoor-Lugt Supercharged Engine

Shop trials were recently run of the first Werskpoor-Lugt Diesel engine ordered and built as a pressure charged unit. This engine, shortly to be delivered to the shipyard of Jos. Boel and Sons, Tamise, Belgium, where it will be installed in the first of three similar cargo vessels of 5,700 tons on order for Leif Hoegh and Company, Oslo, has a cylinder bore of 600 mm. with a stroke of 900 mm. Of the "low-built" type, it is to run only on Diesel oil. Results from the trials have verified the initial claims of the builders, the power being increased from 1,800 b.h.p. for this class of engine when normally-aspirated, to 2,400 b.h.p. at 160 r.p.m., the b.m.e.p. being raised from 70-71lb. per sq. in. to 95lb. per sq. in. for the normal continuous output. It is to be 100lb. per sq. in. on sea trials. The mechanical efficiency of the new engine is 87 per cent. It is fitted with a Brown-Boveri exhaust turbo-blower, adapted to deliver the major part of the necessary quantity of air in series with all the top sections and two low-sides of the four engine-driven reciprocating, double-acting pumps attached to the sides of the engine. The remaining lower pump sections deliver normally-aspirated air in series with all the top sections and two lower sides of the engine. The remaining lower pump sections deliver normally-aspirated air in parallel with the turbo-blower at the full scavenging air pressure of 7lb. per sq. in. It is claimed that this arrangement gives a slightly better overall result than with all four double-acting scavenging pumps in series, although the results of the latter system could be further improved by adapting the blower characteristic to give a higher efficiency at an increased air delivery. The suction of the other two scavenging air pumps, drawing from atmosphere, can be easily connected in series with the exhaust turbo-blower. It is considered a reasonable precaution by the builders to maintain delivery from the reciprocating pumps at all loads and speeds, especially when manœuvring and during long periods of "dead-slow" running, when the pumps regulate the desired output automatically. They are designed so that, as stated, they may deliver the normal quantity of free air from the normally aspirated engine at a b.m.e.p. of 5 kg. per sq. cm. (71lb. per sq. in.), or for the b.m.e.p. of 6·65 kg. per sq. cm. (95lb. per sq. in.), with the pressure charged engine at its normal continuous output. When installed in the ship, the blower will draw from outside the engine room, with the possibility of an arrangement whereby the air suction can be taken from within the engine room in heavy weather. The exhaust gases enter the turbine at an equal pressure, all pressure pulsation being eliminated by adequate dimensioning of the main exhaust line, the intermediate receiver between the blower and reciprocating pumps, and scavenging air belt. While the impulse system for gases to the turbine may have an advantageous effect on the turbine efficiency in relation to the total air quantity to be delivered to the system where all reciprocating scavenge pumps are arranged in series with the turbo charger, it has been proved that there is sufficient energy in the gases at a steady pressure to give about 65 per cent of the amount of air needed for cylinder charging and uniflow scavenging at full charging pressure, the balance of air being produced by the free aspirating pumps. On this engine, the air pressure at the inlets of the pumps in series is practically the same as the blower delivery pressure, and the discharge is at the slightly higher pressure of the scavenging air receiver. An air cooler is interposed in the duct from the blower to the pumps. On trials the exhaust temperature at the valves at 2,400 b.h.p. was 320 deg. C., and before the turbine 395 deg. C., with a pressure of 7·3lb. per sq. in. The fuel consumption is 0·352lb. per b.h.p. hr.—*The Motor Ship, Vol. 33, January 1953; pp. 432-433.*



Schematic arrangement of the series/parallel pressure charging system

S.—Extra suction valves for low running speeds. O.—Overflow valves to operate in connexion with the turbo-blower delivering air at full charging pressure. C.—Connection pipes for series arrangement with the lower ends of the double-acting reciprocating pumps. T.—Turbine. B.—Blower.

P.L.A. Tugs with Kort Nozzles

The four twin-screw Diesel-engined tugs, each with machinery of 1,200 b.h.p., built at Hessle by Richard Dunston, Ltd., were specially designed for handling large ships in limited spaces, and great care was therefore taken to give the pilot the widest possible range of vision from all angles. The tugs are identical, except one, the *Plangent,* which is fitted with twin rudders and Kort nozzles, the others being equipped with normal propellers and single rudders. Provision has been made so that nozzles can be fitted to the non-nozzle tugs, or so that they can be removed from the *Plangent* if desired. All are 85 feet between perpendiculars with a moulded breadth of 24 feet and a moulded depth of 12ft. 6in. The normal draught is 12ft. 9in. aft and 7ft. 2in. forward. In each tug there are two Crossley Diesel engines, each developing 600 b.h.p. at 250 r.p.m. and arranged for fresh-water cooling with automatic temperature control. In cold weather the closed circuit can be heated to ensure easier starting. The *Plangent,* on trials, obtained a pull of 20 tons. An output of 1,850 s.h.p. would be required without nozzles to achieve this pull. This increase in horsepower could not be accommodated within the limits of hull dimensions. The three 1,200 h.p. tugs, without nozzles, each obtained static pulls of 15 tons, and the *Plangent* developed this pull with 780 b.h.p. The twin rudders fitted to the tug equipped with Kort nozzles ensure that, when a large vessel is being handled at low speed or when stationary (conditions requiring maximum pull) both engines can develop full power and, therefore, maintain maximum pull and full manœuvring control of the tug. With the twin-screw tugs having single rudders under similar towing conditions, it is often necessary either to close down or reduce the power of one engine in order to be able to position the tug as directed by the pilot, in relation to the vessel she is handling. The reduced power means that maximum pull is not available. Many tug specifications state a required free running speed without reference to the required static pull, or the horse power to be developed under this condition. This leaves the builder with no option but to design a "free running" propeller—despite the fact that the real duty of a dock tug is to produce maximum pull at low or zero speeds. Such propellers permit only about 75 per cent of the full power to be developed for static pull, while the free speed of dock tugs is unimportant. The Port of London Authority, on the contrary, specified that the propellers for all four tugs were to be designed to be capable of absorbing full power at the bollard, thus producing maximum static pull. Since the revolutions are governed, the propellers will, therefore, require less torque, and absorb less power for free running than when pulling at the bollard. An incidental advantage claimed for the nozzle is that this loss of power is not so great as would otherwise be the case. For a dock tug, however, this is not important. The *Plangent* is the only twin-screw Diesel-engined nozzle tug in the United Kingdom.—*The Motor Ship, Vol. 33, January 1953; p. 389.*

Automatic Control of Fuel Viscosity

Experience has shown that in the utilization of boiler oil in the Diesel engines of motor ships it is essential to maintain appropriate temperatures throughout the oil system, and particularly to ensure that the fuel oil is delivered to the valves at a viscosity which allows effective atomization in the fuel valves. Constant vigilance is therefore necessary on the part of the engineers, and it seems possible that their path might be eased by the adoption of a device which would automatically control the viscosity of the oil when it reaches the valves. Automatic viscosity control is now being used with the boilers in some American steamers where the same problem arises and it is necessary to maintain the correct viscosity at the oil burners. The equipment employed is known as a Viscorator and it would seem that it could be adapted for employment with Diesel machinery. The most suitable viscosity for Bunker C fuel for oil burners is found to be about 150 Saybolt Universal (130 sec. Redwood No. 1) at the oil burner. Tubing a quarter of an inch in diameter connects the Viscorator to the hot end of the oil heaters before delivery to the fuel valves in the case of a Diesel engine, or to the oil burners with oil-fired boilers. The hot oil circulates through the Viscorator and is returned to the oil system by ⅜-inch tubing to the settling tank in a cargo ship or the deep tanks in a tanker. When the viscosity differs as little as one second Saybolt Universal from the settling, which is usually about 150 SSU to meet the general requirements, the Viscorator immediately reacts on an air-controlled steam valve and the temperature is correspondingly raised or lowered in the heaters and the viscosity of the fuel increased or diminished. In addition, the unit continuously makes a recording of the temperature and viscosity of the oil for permanent record. Even when oils are blended, which happens frequently in the case of boiler oil for Diesel engines, it is claimed that a uniform viscosity is maintained. It is found that in the use of the Viscorator with oil-fired steamers, even with extremely heavy oils having a gravity of 5 to 6 API or about 1·033 specific gravity, the working of the Viscorator is satisfactory, also when manœuvring, during which period the viscosity is found to be constant within 5 to 6 plus or minus SSU.—*The Motor Ship, Vol. 33, January 1953; p. 389.*

Vessel for West African Trade

The motorship *Burutu Palm,* which has been completed by Short Bros., Ltd., Sunderland, for the Palm Line, Ltd., is intended for the West African trade of her owners. The *Burutu Palm* is a ship of 8,516 tons deadweight, and has a single deck and open shelter deck, with poop and forecastle. She has been built to the highest class of Lloyd's Register. The leading particulars of the ship are as follows: —

Length b.p.	425 feet
Breadth, extreme	57ft. 9in.
Depth moulded to shelter deck	37ft. 2in.
Summer draught	23ft. 9½in.
Tonnage, gross...	5,500 tons
Tonnage, deadweight	8,516 tons

The propelling machinery, which was supplied and installed by the North Eastern Marine Engineering Co. (1938), Ltd., consists of a four-cylinder N.E.M.-Doxford opposed piston airless injection oil engine with two lever-driven scavenging air pumps, direct-coupled to the shaft. The cylinders are 560 mm. in diameter and the combined stroke of the pistons is 2,160 mm. The engine is designed to develop 3,000 b.h.p. in service at 122 r.p.m. An interesting innovation in the engine controls is that the fuel pressure control is operated by means of a lever, instead of the more usual handwheel. There are two auxiliary boilers, one of Fleming and one of Cochran type. Both are oil burning, and the latter is also arranged for exhaust gas firing. The electrical installation is fed by three Diesel-driven generators, each 150 kW. at 220 volts D.C., together with a 25-kW. Diesel generator for port use.—*The Shipping World, Vol. 127, 17th December 1952; p. 480.*

Vanes Singing in Water

In the course of propulsion experiments in the ring-channel and rotating-boom facility at the Aerojet Engineering Corporation, it was discovered that certain struts which are used to support underwater bodies would "sing" at surprisingly low speeds, e.g. 5 to 7 knots. Tail surfaces also sang. The note emitted was a clear musical tone in most instances, and, as the speed was increased, the note would fade, disappear, and then reappear as a note of higher frequency. This singing caused considerable apprehension as it was feared that the strut would fail at high speeds due to build-up of the vibrations. Analysis for flutter stability and torsional divergence had been conducted, and it was known that the operation was well away from the ranges where flutter or torsional divergence might occur. However, the severity of the singing did not tend to become greater at high speeds, and there were usually quiet speed zones at high speeds as well as at low speeds. The experimental work performed in this study supports the following conclusions: (1) Singing of vanes, and incidentally of marine propellers, is caused by the action of von Kármán vortex streets which are generated at the trailing edge. (2) The critical length, which together with speed, determines the spacing of the street, and hence the frequency of singing is the thickness of the trailing edge of the vane. (3) The Strouhal number, Nb/U, can be made to apply to these streets and assume the customary range of values by increasing the length b by a fraction of the boundary-layer thickness. (4) Singing does not occur at speeds from 0 to 60 knots on vanes which have trailing edges thinner than 0·007 to 0·010in. per ft. of chord. This is suggested as a limit of trailing-edge thickness for marine propellers to suppress singing. (5) Singing will occur on a vane of the general type considered here whenever the plate can assume a mode of vibration such that the modified Strouhal number Nb/U is approximately 0·18. (6) One of the principal types of vibration of the plate is with the trailing edge vibrating in scallops. Each scallop shed a vortex street. (7) Vanes in which the trailing edge is not normal to the flow will sing and the corresponding vortex street is inclined to the flow. The component of velocity normal to the trailing edge controls in the Strouhal number for this case. (8) Thick vanes require a higher velocity for the inception of singing than thin ones, other things being equal. (9) The effect of angle of attack of the vane is to weaken the singing somewhat. This seems to be caused by the effect of finite span and the shedding of the lift vorticity which distorts and suppresses the vortex streets. (10) Leading edges and tips of vanes are not factors in the generation of the vortex streets and hence do not influence singing. (11) From consideration of the mechanism of the action, the singing note is radiated directly from the oscillating surfaces of the vane and not from the street or wake. (12) The vibration of the singing vane is generated by the oscillatory circulation produced about the vane as the oscillating vorticity is shed in the street. This results in oscillating transverse forces on the vane which drive the vibration. (13) In general, when the vane is not singing the vortex streets are not being generated.—*G. A. Gongwer, Journal of Applied Mechanics, Vol. 19, December 1952; pp. 432-438.*

French Diesel-electric Tug

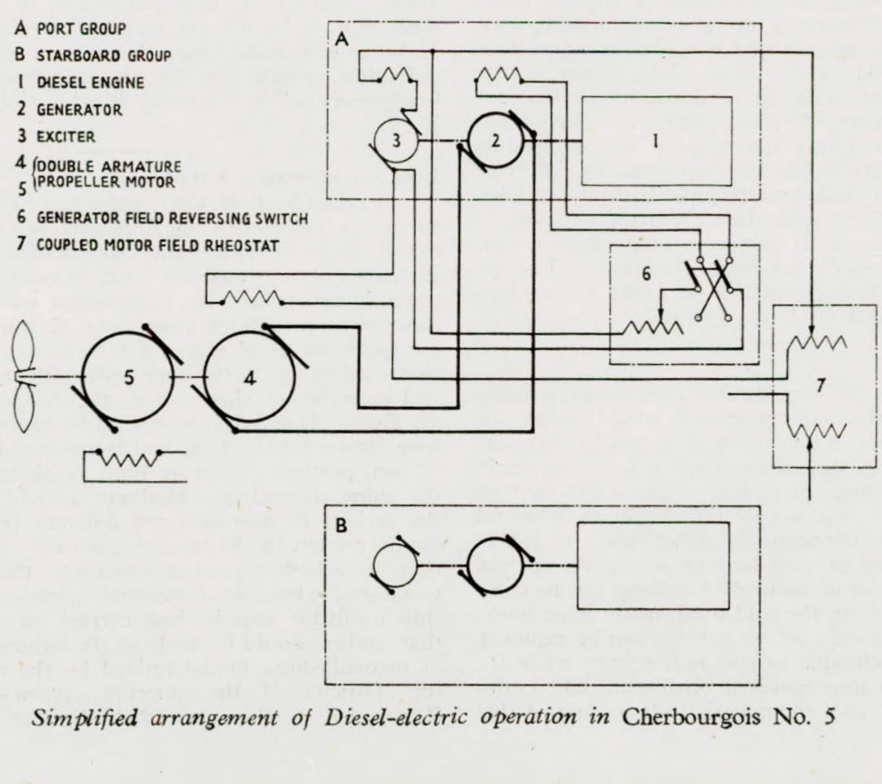
Sea trials were recently completed of the deep sea tug *Cherbourgois No. 5,* built by the Chantiers et Ateliers Augustin-Normand, Le Havre, to the order of the Société Cherbourgeoise de Remorquage et de Sauvetage. Propelled by Diesel-electric machinery, the vessel is one of the most interesting of the type to be built in France. Built to the requirements of Bureau Veritas, highest class, for deep sea towing, the vessel conforms to the requirements of the 1948 London Convention and to the Seattle Convention. Electric welding was largely used in the construction, but the shell plating was riveted. The propelling machinery consists of two Diesel driven generators supplying current to a double armature propulsion motor direct coupled to the propeller shaft. The generators are driven by M.A.N. 4-stroke, single-acting supercharged Diesel engines, type W.8.V.30/38, each developing 1,020 b.h.p. at 550 r.p.m., and built by the Société Générale de Constructions Mécanique, La Courneuve. The electrical installation is divided into two groups, each comprising:—

Diesel engine of 1,020 b.h.p. at 550 r.p.m.
Generator, 630 kW., 500 v.
Exciter, 55 kW., 220 v.
Propeller motor, 1,500 h.p. at 150 r.p.m.

Each propeller motor is controlled on the Ward-Leonard principle, that is, it receives current from a generator with variable excitation. Reversal is effected by altering the direction of the armature current, this in turn being produced by reversal of the generator field. The speed of the propeller motor can thus be varied between 0-120 r.p.m. by variation of the generator field, and from 120-150 r.p.m. by field weakening in the motor. Variation of the generator fields as well as their reversal is effected by means of a reversing switch that can be operated either from the bridge through a servo-motor, or directly from the control panel in the engine room. A simplified layout of the electrical system is shown in the accompanying sketch.—*Shipbuilding and Shipping Record, Vol. 80, 20th November 1952; pp. 679-680.*

Solid Inclusions in Bearing Oil Supply

Sleeve bearings operated under normal conditions are always subjected to some foreign solid particles in the oil supply. Internal-combustion engines produce solids in the combustion process; other particles are produced by wear; and others enter as dirt, rust, and scale from the air or piping. In order to ascertain the effect of solid inclusions in sleeve-bearing oil supply, an experimental test programme was arranged to obtain first a basis of comparison by using clean oil, then to proceed to soft small particles and increase the particle size, concentration, and hardness until some definite influence upon the operating characteristics of the bearing was obtained. A large number of test points were recorded, from which the following conclusions were obtained, based upon average operating conditions: (1) Soft particles of graphite and molybdenum sulphide in the oil supply to lead-base-babbitt sleeve bearings will produce no improvement in the bearing operation. (2) Graphite particle size has no effect upon the bearings' operating characteristics when used in small concentrations as in this programme. (3) Graphite particle concentration will produce a



Simplified arrangement of Diesel-electric operation in Cherbourgois No. 5

change in the bearing oil-flow characteristics. Increased concentration will increase the oil pressure required to maintain the same oil flow. (4) The coefficient of friction may be increased by increasing particle concentration. (5) Only the tests with small amounts of rouge caused friction values lower than clean oil. (6) Particles of hard substances must be larger than the minimum oil-film thickness to cause appreciable increases in wear or friction. (7) Wear may take place without a large increase in the bearing temperature. (8) Shaft wear may be much more rapid than bearing wear owing to the babbitt embedding the large sharp particles. (9) Oil may be discoloured by small concentrations of finely divided particles, but a discoloured oil does not indicate the friction to be expected in a bearing. Very small amounts of hard particles will cause rapid wear even though the oil looks clean. (10) Filtration that removes any-size particle will be helpful, but the removal of all particles larger than the minimum oil-film thickness is required to stop wear and increased friction.—*H. G. Rylander, Mechanical Engineering, Vol. 74, December 1952; pp. 963-966.*

Friction Between Aluminium and Other Metals

With one exception, no information has been published on the coefficient of friction of aluminium against other metals in the absence of lubrication. The exception is some results published by R. Aida, who determined the coefficient of friction by measuring the angle at which specimens commenced to slide down an inclined plane. The contact area of the specimens measured 20 × 20 mm. and the load ranged from 10 to 300 g. The authors briefly summarize the results obtained by Aida for aluminium against cast iron, steel and brass. In their own experiments, for which only preliminary results are reported, the authors were concerned with much higher pressures (100 – 3,000 kg./cm.^{2}) under which the coefficient of friction (static and dynamic) of aluminium and two Al-Cu-Mg alloys (different heat treatments) against steel and an Al-Cu-Mg alloy was determined. In the apparatus, which was mounted in a tensile testing machine, two small test blocks were pressed under a measured pressure against both sides of a test strip to which tension was applied by the machine. Static and dynamic coefficients of friction were calculated from the loads required to start and continue the pulling through of the test strip. The coefficients of friction are shown plotted against the pressure on the test blocks. The coefficients of friction varied differently with pressure depending on the materials being used. The coefficient of friction against steel was always higher than that against the Al-Cu-Mg alloy blocks sliding over a strip of the same material; the coefficient of dynamic friction was, however, higher than that of static friction. Particularly favourable conditions for power transmission are obtained in the case of heat treated Al-Cu-Mg alloy on steel, the difference between the coefficients of static and dynamic friction being less than with any of the other combinations, and the variation of the coefficients of friction with pressure being likewise only moderate.—*H. Kostron and F. Schalberg, Aluminium, Vol. 28, October 1952; pp. 341-346. Abstracted in Light Metals Bulletin, Vol. 14, 5th December 1952; pp. 961-962.*

Hot Radiography

When making welds of considerable depth, radiographic inspection for flaws as the weld progresses would ensure the discovery of faults in time and in many cases would eliminate the necessity of removing the entire joint weld. The main obstacle to such a procedure, of course, is the waste of time incurred in waiting for the weld to cool sufficiently, as industrial X-ray film is vulnerable to temperatures higher than 100 deg. F. Moreover, a long process of post-heating to relieve internal stresses may be required before controlled cooling can be commenced. After radiographing the cold weld, many more hours of preheating may be necessary before welding can be resumed. The development of a technique to take radiographs while the material is still quite hot now makes it possible to take radiographs without having to cool the material. It is claimed that in certain cases the new technique reduces by some 55 per cent the total time required to weld pipe joints in high-temperature steam piping. In order to make it possible to place the X-ray film in position while the metal is still hot, a water-cooled jacket is provided, which is placed between the metal and the film. In the case of a pipe weld, it consists of two hollow cylinder sectors hinged at one end and fitted with wing nut fasteners at the other. Piping connexions provide inlet and outlet for the cooling water. Hollow raised sides of the jacket help to maintain an even temperature distribution along the surface which comes into contact with the film cassette. Flexible cassettes containing the film fit snugly between the raised edges of the jacket and are held in place with heavy rubber retaining bands. A source of gamma radiation is placed at the centre of the pipe and penetrates the weld.—*The Engineers' Digest, Vol. 13, December 1952; p. 405.*

Pneumatic Breakwater

After the outer gate of the train ferry dock at Dover harbour was removed for overhaul in September it was observed that the inner gate was vulnerable to waves and swell when it was being raised or lowered. As a protective measure, a pneumatic breakwater was installed on the recommendation of the Civil Engineer (Southern Region), British Railways, acting in consultation with the Docks Engineer, Docks and Inland Waterways Executive. The installation is now in regular use, when conditions require that the gate should be protected. The breakwater consists of a number of perforated pipes laid under the water. Compressed air is pumped through these pipes, with the effect that the swell is greatly reduced. At Dover the pipes are laid on the sea bed due to the shallow depth of the water. The pipes are made up from lengths and flanged at the ends for assembly. Compressed air pumped through the pipes escapes through the perforations and forms a cushion of air which reduces the waves and swell. In the Dover installation the compressed air is supplied from a Palouste air compressor, supplied by the Blackburn and General Aircraft Co., Ltd. (Turbomeca Division). This compressor has an output of 500 to 1,800 cu. ft. of air at 50lb. per sq. in. with shaft speeds of 26,000 to 35,000 r.p.m. Running on kerosene, the Palouste compressor is started by a 24-volt battery. The installation has been stated to be giving every satisfaction in operation, a 4-foot swell having been reduced by as much as 50 per cent. This type of breakwater may well be installed in many small harbours and docks where there are economic and geographical difficulties which prevent the construction of conventional breakwaters.—*The Shipping World, Vol. 128, January 1953; p. 21.*

Long-period Waves in Harbours

A paper by J. H. Carr, published in the Proceedings of the American Society of Civil Engineers, reports results of field and model studies of sources and characteristics of long-period waves in harbours, and analyses basin resonance and water motions induced by waves. By long-period waves, the author means those with periods of from 1 to 12 min. or more, as distinguished from usual waves of 8 to 20-sec. periods. It is shown that models are particularly adaptable to this type of study, and examples of their use in the design of harbour facilities are given. Even when restrained by heavy tackle, moored ships have shown longitudinal and transverse drift amplitudes up to 10 feet, snapping mooring lines, breaking piles, and damaging the ships themselves. Harbours should be so designed that the periods of resonance are different from the periods commonly present in the incoming waves. There are certain locations in a harbour, corresponding to the loops in the vertical vibrations, where the horizontal vibration is a minimum, and ship outfitting can be best carried on. The author believes that studies should be made of the natural period of oscillation of moored ships, as determined by the mass of the ship and the elasticity of the mooring system.—*Applied Mechanics Review, Vol. 5, December 1952; Abstract No. 3,447.*

ENGINEERING ABSTRACTS

Section 3 Shipbuilding and Marine Engineering

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Step Welding of Steel

An article by T. Noren in "Stahl und Eisen" describes a new welding process, which is particularly suitable for alloyed tool steels that are prone to crack on welding. There are two modifications. In the first, the material to be welded is heated to its austenitizing temperature, held there, and then cooled to a temperature at which the austenite transforms only very sluggishly to the pearlitic or intermediate structure; it is welded at that temperature, with a welding rod, the hardening temperature of which coincides with the welding temperature. The whole is then allowed to cool naturally to room temperature, and annealed at low temperature. When the austenite transformation is not sufficiently sluggish, the second procedure is employed. Here the material is heated as before, but cooled to a temperature at which the austenitic state of the welding rod material is more stable, and welding carried out. In this case, the body of the material remains soft. In both cases the welds are free from cracking.—*Journal of The Iron and Steel Institute, Vol. 172, December 1952; p. 439.*

Long Beams Under Transverse Impact Loading

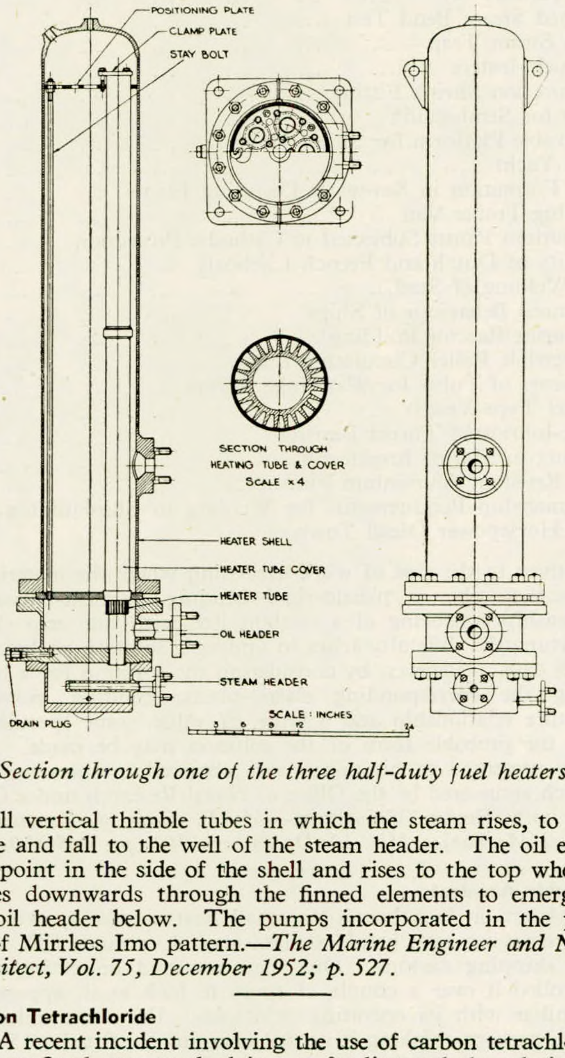
The object of this paper is to set forth the results of an investigation of the behaviour of long beams under transverse, constant-velocity impact loading, when a plastic-rigid type of analysis is adopted. It was expected that such an analysis would be satisfactory for problems involving large strains, and easier to evaluate than the corresponding elastic-plastic solution. Consideration is first given to the case of ideal plasticity. Elastic strains are neglected and the material of the beam is assumed to flow plastically at a constant yield limit. In this case expressions for the bending moment, shear force, curvature and deflexion distributions along the beam are obtained analytically for any given impact velocity. The manner in which the solution for a beam having an elastic-ideally plastic bending moment-curvature relationship converges to the plastic-rigid solution, as EI increases, is discussed. Consideration is next given to the case of work-hardening where the material is assumed to obey a plastic-rigid bending moment-curvature relationship consisting of a straight line with non-zero slope. Unfortunately, difficulty arises in finding a solution analytically in this case. However, by considering the solution for a beam having the corresponding elastic-plastic bending moment-curvature relationship and a large EI value, some speculation as to the probable form of the solution may be made. The results presented in his paper were obtained in the course of research sponsored by the Office of Naval Research under Contract with Brown University.—*M. F. Conroy, Journal of Applied Mechanics, Vol. 19, December 1952; pp. 465-470.*

Avoidable Accident

Aboard a dock boat of a small boat company, two men were removing new hand extinguishers, 10-pound, CO_2 type, from shipping cartons. One man removed an extinguisher and rolled it over a couple of times to look at it, apparently unfamiliar with its operating principles. For some unknown reason, perhaps thinking it needed charging, he fitted a 15-inch crescent wrench to the screw cap on the charging side of the cylinder. As he loosened the cap a hissing noise was heard. "It's leaking gas", shouted his helper, as he came over to look on. The screw cap was turned back and this man began reading a tag (instructions for filling). While he was reading the instructions, the helper removed the cap from the discharge line and started again to loosen the plug on the charging side. Suddenly, there was a loud "pop" and the interior of the dock boat became dense with CO_2 fumes. One man was killed instantly and the other critically injured. In commenting upon the case, the investigating officer concluded "it is evident that the plug on the charging side was loosened until the number of threads holding were not sufficient to withstand the pressure within the cylinder. When the threads let go, the escaping gas also fractured the rupture disc on the discharge side. This allowed the gas to pass out through the two openings, thus setting the cylinder in a whirling motion".—*Proceedings of the Merchant Marine Council, United States Coast Guard, Vol. 9, November 1952; p. 246.*

Oil Fuel Heaters

The oil fuel burning installation in the Alfred Holt liner *Nestor* has been supplied by Associated British Combustion, Ltd., of Meonstoke. There are four A.B.C. burners to each boiler and features of the equipment are the wide pressure range (up to 500lb. per sq. in.) and the use of three half-size vertical heaters, rather than the more usual duplicate full-size units. These are a new design of secondary surface heater, and a sectional drawing is shown here. Each heater contains a number



Section through one of the three half-duty fuel heaters

of tall vertical thimble tubes in which the steam rises, to condense and fall to the well of the steam header. The oil enters at a point in the side of the shell and rises to the top where it passes downwards through the finned elements to emerge at the oil header below. The pumps incorporated in the plant are of Mirrlees Imo pattern.—*The Marine Engineer and Naval Architect, Vol. 75, December 1952; p. 527.*

Carbon Tetrachloride

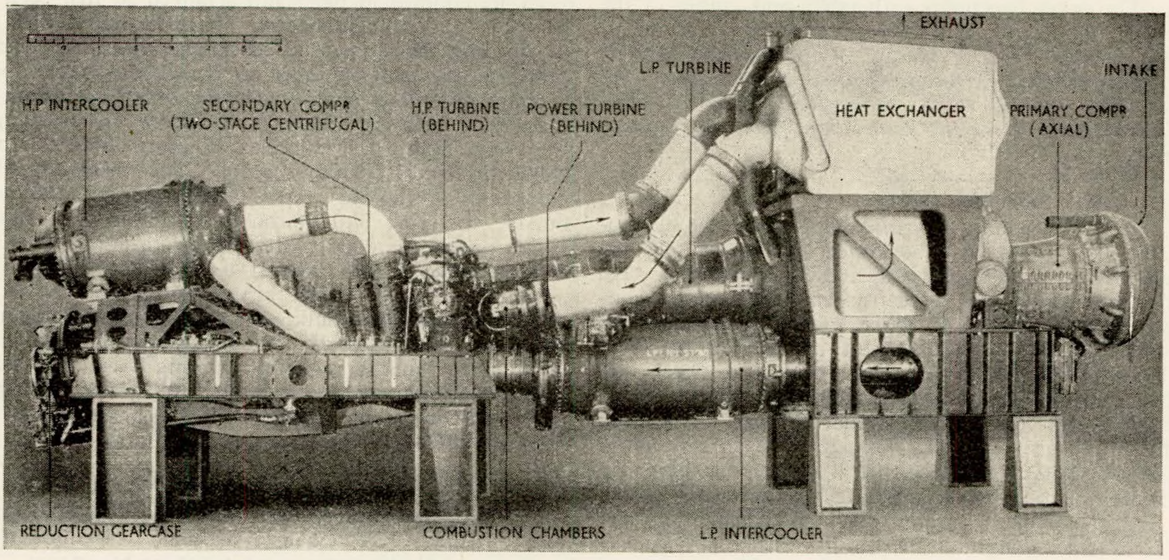
A recent incident involving the use of carbon tetrachloride in a confined area resulted in one fatality and the admittance to hospital of ten. Carbon tetrachloride was used to clean machine equipment in a confined space. The first symptoms resulting from exposure to this toxic solvent did not appear until one week later when one person died and ten were admitted to hospital within the following twenty-four hours. The inherent hazards of carbon tetrachloride have been stressed many times, whether used as a fire-extinguishing agent or for any other purpose. Unfortunately, some of the personnel directing operations involving the use of carbon tetrachloride are not always informed of these hazards or perhaps they do not take all the necessary precautions. Carbon tetrachloride is widely used and has been the cause of a considerable number of cases of poisoning. Prolonged, excessive, or repeated exposures to the compound are hazardous and may result in serious injury or death. It is easily absorbed by the mucous membranes, the lungs, and to some extent, by the skin. In the presence of water, especially at elevated temperatures, carbon tetrachloride is corrosive. In the presence of open flames or when exposed to hot objects, carbon tetrachloride decomposes to form phosgene, a deadly gas, and hydrochloric acid. It can cause injurious effects due to inhalation, ingestion or by prolonged or repeated contact with the skin. The toxic effects can be either acute or chronic. The former is brought about by a single exposure to a heavy concentration of carbon tetrachloride. The latter is the result of repeated exposure to lower concentrations. Symptoms may not appear until days after the initial exposure. One case was reported of a worker who had only a five minute exposure to a heavy concentration from a fire extinguisher. Nine days later he developed convulsions and other complications. Fortunately he recovered. The hazard of using a carbon tetrachloride extinguisher on a fire is greatest when the liquid is discharged on slow burning or smouldering electrical equipment, automobile motors or any metal object involved by fire. Since these fires are small and extinguished without any appreciable loss, personnel are prone to remain in the area and will be exposed to the deadly effects of phosgene. All personnel should be removed from the vicinity, and the area well ventilated. The chemical reaction of carbon tetrachloride on hot metals is rapid decomposition with the production of phosgene. Fire fighting personnel are enjoined to use the self-contained breathing apparatus when working in areas that have been exposed to phosgene gas.—*Proceedings of the Merchant Marine Council, United States Coast Guard, Vol. 9, October 1952; p. 227.*

The Royal Yacht

The Queen has approved a model showing the general arrangements of the small hospital ship to be used as a Royal yacht in peacetime. To be launched by Her Majesty in April next, the ship is expected to be complete at the end of 1953. The keel was laid by John Brown and Co., Ltd., in June and most of the structure below the tank top and many bulkheads and frames are already erected. Arrangements are such that the ship can be converted into a hospital ship with the minimum structural alteration. Should she be required as a hospital ship, the Royal and State Apartments could be converted into hospital wards, operating theatre, etc., for which air conditioning is being fitted. The after end of the shelter deck is strong enough to allow a helicopter to land on it with patients. The load displacement is about 4,000 tons and the maximum draught in any seagoing condition will not exceed 16 feet. The ship's overall length will be 413 feet; her length at the waterline 380 feet; her maximum beam 55 feet, and her moulded depth 32ft. 6in. Single-reduction geared steam turbines fed by two boilers will drive twin screws to give her a continuous cruising speed of 21 knots. She will have a modified cruiser stern and a raked bow. Three masts are to be fitted. The Royal Standard will be worn at the main, the Flag of the Lord High Admiral at the fore and the Union Jack at the mizzen. The ship will be fitted with stabilizers to reduce her roll in bad weather. There will be Decca Navigator and radar to assist her navigation. The ship will be fitted out for voyaging in both cold and tropical waters. The entire structure above the waterline is to be riveted.—*The Shipping World, Vol. 127, 26th November 1952; p. 424.*

Gas Turbines for Light Naval Craft

The advantages of applying the experience gained on aircraft gas turbines to marine propulsion have been fully appreciated by the Admiralty for some time, and after considerable investigation by Rolls-Royce, Ltd., of Derby, under the direction of the Engineer-in-Chief of the Fleet, a contract was placed in 1946 for the production of two gas turbine engines for propulsion of coastal craft. Design work on these engines, which were designated R.M.60, was started in December 1947, and by 26th June 1951 the first set was on test. During these first tests 90 per cent of the design power was attained and 220 hours of test running were completed before the engine was



The Rolls-Royce 6,000 h.p. marine gas turbine, showing principal components

stripped for examination. The installation of these engines in a former S.G.B. will be completed next year. Naval requirements demanded an economical low cruising power and the high compression ratio necessary has been achieved by multi-stage compression with intercooling. A heat exchanger is also included. The low pressure axial compressor delivers air through a sea water-cooled intercooler to a two-stage centrifugal compressor, intercooling again being employed between the two centrifugal stages. Air at maximum cycle pressure is then passed through a heat exchanger before being delivered to the combustion chambers. The resulting high-temperature gas is expanded in series through three mechanically-independent turbines. The high pressure turbine (i.e. the furthest aft) drives the high pressure compressors by means of a hollow shaft, the power turbine drives the Rotol three-bladed variable and reversible pitch propeller by a shaft passing through the hollow one- and two-stage reduction gears, and the low pressure turbine drives the low pressure compressor. Due to the mechanically-independent power turbine, this cycle has the advantage of low power economy and improved flexibility. Aero-engine practice has resulted in a light and compact power unit giving considerable increase in total power and a reduction of 50 per cent in total machinery weight as compared with the lightest steam machinery yet produced for Naval purposes. In addition a saving in machinery space has been made possible. The R.M.60 is designed as a medium life engine for development purposes only, and it is anticipated that the experience gained during its operation will materially assist in future development of marine gas turbines with long life between overhauls.—*The Marine Engineer and Naval Architect, Vol. 75, December 1952; pp. 521-522.*

Submarine Reactor Building

Completion of the saucer-shaped foundation for the 225-foot steel sphere that is to house a nuclear submarine power plant being built by the Atomic Energy Commission for the U.S. Navy has been announced, and work on assembling the hull of the land-based prototype submarine is under way, according to the Schenectady Operations Office of the A.E.C. and the General Electric Company. Known as the Submarine Intermediate Reactor (SIR), this project is under the direction of the Knolls Atomic Power Laboratory operated at Schenectady for the A.E.C. by the General Electric Company. The spherical design of the reactor building was adopted to give additional protection to operating personnel and to off-site areas during test operations beyond the many safety controls of the reactor itself. In the remote event that simultaneously all other controls failed, the resulting release of radioactive material would be contained in the sphere which will have a net "free" space of more than 5,400,000 cu. ft. The outer periphery of the building will be 706 feet. The sphere will rest on the concrete saucer just completed which is 179 feet in diameter and 42 feet deep. A ring of steel columns set on concrete outside the structure and reaching to the middle of the sphere will give further support to the building. Welded steel plates will make up the skin of the ball. Every weld in the structure must be X-rayed to assure that there are no leaks. To do this, on the bottom, a four-foot space is provided temporarily between the base of the sphere and the concrete saucer. After testing is completed, this space will be filled with concrete and aggregate. Inside, the concrete floor on which the reactor will rest will be slightly above ground level and the well of the saucer beneath the floor will be filled with a compacted mixture of aggregate and earth. The reactor building will be air-conditioned. The hull of the submarine will be assembled just outside the building and when the latter is completed and tested, the hull will be skidded into the huge ball through a special wall section and the sphere again sealed. The Submarine Intermediate Reactor is one of two nuclear-science approaches being made to the problem of utilizing atomic fuel for underwater ship propulsion. The other is incorporated in a project at the National Reactor Testing Station in Idaho. The reactor designed for the West Milton site will use liquid-sodium metal to take the heat out of the reactor core and into an exchanger or "boiler" where water will be converted to steam. The steam will then drive the turbines that propel the submarine.—*Mechanical Engineering, Vol. 75, January 1953; p. 29.*

Stability of Dutch and French Lifeboats

Models of five Dutch and two French types of lifeboats in wood and light alloy were tested for stability under various conditions of load and flooding. The authors give the basic analysis of the experiments and full particulars of the lifeboats and their models tested. According to the International Convention for the Safety of Life at Sea, the capacity of the air tanks in lifeboats should not be less than ten per cent of the volume of the boat up to the gunwale. This percentage was shown to be insufficient for one of the Dutch boats used. The other Dutch boats kept afloat, but with gunwale completely submerged. Apart from supplying reserve buoyancy, the air tanks improve the stability as well. If the occupants move their position, then the stability considerations are meaningless, so buoyancy only should be considered in the first place. It is proposed to increase the capacity of the air tanks to give more buoyancy. The paper concludes with tables and graphs of the results of the experiments, and there is a drawing of one of the lifeboats tested.—*H. E. Jaeger, J. W. Bonebakker and J. Pereboom, Netherlands Research Centre, Shipbuilding and Navigation. Report No. 9S, October 1952. Journal, The British Shipbuilding Research Association, Vol. 8, January 1953. Abstract No. 6,994.*

Couplings for Geared Propulsion Units

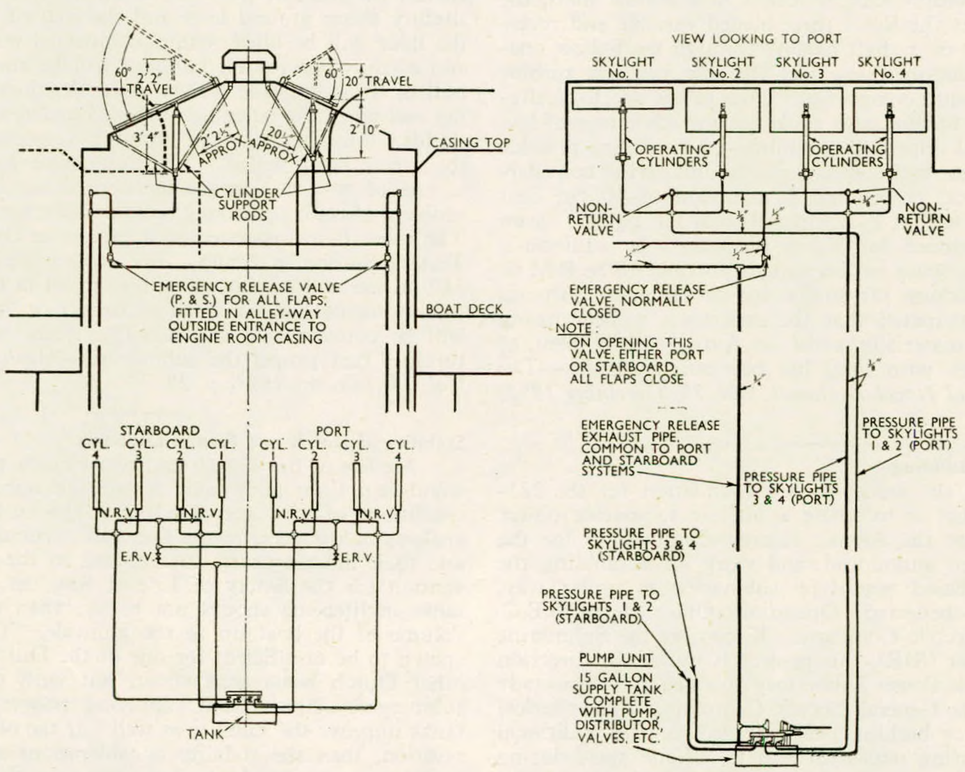
Four types of coupling are reviewed, namely the electro-magnetic slip coupling, the air-operated friction clutch, the oil-operated friction clutch, and the hydraulic coupling. No consideration has been given to electric drives and torque converters, as these are considered to be more in the nature of substitutes for gearing. The basis for comparison has been taken to be:—(a) Characteristics when used as a clutch; (b) Flexibility of transmission; (c) Acceptability of misalignment; (d) Efficiency of transmission. A tabular comparison is given enabling the advantages and disadvantages of each type to be assessed. A description of each type is given, together with details of some modifications which have been made for special purposes, and an attempt has been made to indicate the suitability of the various couplings for conditions likely to be met in service. Both the electric slip coupling, the oil-operated friction clutch, and the hydraulic coupling are well established for marine drives both in this country and abroad, but the air-operated or pneumatic clutch is not so well known, having come into prominence during the war, when it was used extensively under the arduous service conditions obtaining in invasion craft, both large and small. This type of coupling would appear to be worth serious consideration for propulsion drives in the smaller vessels, such as tugs and coasters.—*The British Shipbuilding Research Association, Report No. 68.*

Hydraulic System for Engine Room Skylights

A new system for the opening and closing of engine room skylights is fitted in the passenger-cargo liner *City of Port Elizabeth* and the three similar vessels of 12,500 gross tons for the Ellerman Lines. The eight opening lights in the engine room are coupled in pairs and arranged so that each pair can be opened independently to any desired extent. This is effected by opening the appropriate cock and operating a small hand pump at the control point on the engine room floor. This actuates the rams slung under the selected opening lights to open them with the minimum of effort. In an emergency—in the event of fire, for example—all open lights can be quickly closed by merely opening a valve. The emergency relief valves are placed in the alleyways of the engineers' accommodation, outside the port and starboard doors to the engine room. Furthermore, the open lights can be instantly closed from the control cocks to the exhaust or closed position. A further advantage claimed is that this system is much easier to install than an arrangement calling for mechanical shafting and bevels. With the hand-hydraulic system, installation involves only the running of small-bore copper piping from the control and emergency points to the skylights. It is stated that practically no maintenance is required.—*The Motor Ship, Vol. 33, February 1953; p. 479.*

Diesel Engine Indicators and Indicating

The Diesel engine user, unlike the development engineer, rarely requires more than a reasonably accurate measure of maximum cylinder pressure when tuning his engine. This form of the Farnborough indicator answers the purpose admirably. There are many other forms of maximum pressure indicator, usually based on the principle of trapping the peak pressure beyond some type of non-return valve. Most of these read low; some are affected by the rate of rise of pressure. There are others having spring-loaded pistons, the limit of travel of which is indicated either by mechanical or electrical device; they are extensively used for comparing maximum pressure. Recently the cathode ray optical indicator has come into general use for development work on high-speed engines. It is especially valuable for recording transient pressures such as are present in a modern injection system, also it is most useful for reproducing injector needle lift on the screen. In addition to the above, the cathode ray indicator has almost unlimited uses for measuring changes in clearance or gaps between moving and



Diagrammatic arrangement of hydraulic system for skylights stationary parts of any mechanical contrivance: for example, a change of clearance at any particular point in a bearing in which a shaft is rotating. It has been shown that the centres of some journals move in a peculiar pattern during each revolution and this can be studied by this versatile equipment. Another example is in the measurement of clearance between the tip of the rotor and the casing of a blower when running at various speeds and delivery pressures. The C.R.O. indicator consists basically of three main components, the pick-up unit, the amplifier, and the oscillograph; of these, the first is the most important and probably presents the most problems. Its function is to convert a mechanical quantity into a small electrical signal; various methods have been tried, all with certain advantages and disadvantages. Possibly the most favoured unit for oil engine work is the electro-magnetic type, on account of its ability to respond to frequencies down to zero, which implies that it can be calibrated. An advantage of this unit having low impedance is that it is fairly immune from damage due to oil, an invaluable asset when indicating the fuel system of a Diesel engine. Furthermore, it is adaptable to various applications and can be used with long cables to the recording apparatus, if necessary. The main disadvantage of the system is the sensitivity of the pick-up unit to temperature, but so long as this is taken into consideration when calibrating, it may be said that the advantages outweigh the disadvantages.—*G. B. Fox, Gas and Oil Power, Vol. 48, January 1953; pp. 7-10.*

Light Alloys in Admiralty Diesel Engine

The Admiralty A.S.R. II Diesel engine, designed and constructed by Davey, Paxman and Co., Ltd., is a striking example of aluminium construction. Basically the design is similar to the Paxman R.P.H. series which is of cast iron construction, and it is to be noted that in redesigning for aluminium there has been very little change and virtually no increase in the bulk of the various components. The engine is a V-type, four-stroke unit with a bore and stroke of 7 inches and 7¾ inches respectively, the range covering power requirements of 200 b.h.p. —the 12-hour rating of the 6-cylinder normally aspirated engine —to 800 b.h.p. with a 12-cylinder supercharged engine. The largest aluminium component is the crankcase, which is a one-piece casting in LM7-P. Cylinder blocks and cylinder heads are also castings. Among other aluminium castings are the cam box—the camshaft bearing directly in the aluminium— and the intake manifolds, each cast in one piece. The sump is in sheet aluminium alloy, fabricated by argon-arc welding; forgings in alloy H12 are used for the main bearing caps, rockers and other small parts. A somewhat unusual feature of the normally aspirated engine is an aluminium alloy exhaust manifold. This is fabricated by welding and comprises castings attached to an extruded tube which is encased in a sheet-aluminium water jacket. Both the fuel and lubricating oil pumps are in aluminium alloy, as also are the gear case covers, cylinder head covers, an air vessel for deaerating the fuel before its delivery to the fuel pumps, water piping and, of course, the pistons. Both propulsion and auxiliary purposes are fulfilled by the engine, and, where it is used for generator duty, another interesting application of aluminium is provided. The complete generator set is mounted on a fabricated underframe which is in two sections, each built up from extruded 12 inch by 4 inch channel section and 1 inch plate in aluminium alloy N5/6. The overall dimensions of the engine section of the underframe are 6ft. 6in. long by 5ft. 4in. wide by 2ft. 5in. high and that for the generator is 7ft. 8in. by 5ft. 4in. by 2ft. 5in. All the components of these frames are joined by Aircomatic Welding. —*The Shipping World, Vol. 128, 4th February 1953; p. 153.*

Burning of Boiler Oil in Diesel Engines

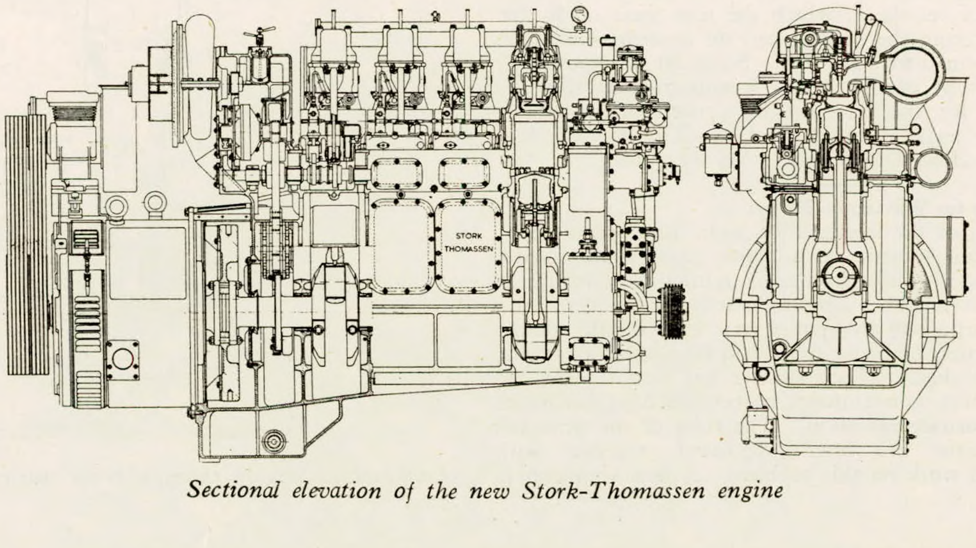
Experience of the burning of boiler oil in five different types of Diesel engines is described, explanations are given of the experiments made, and the economies effected are recorded. The development of the Archaouloff fuel injection system, as fitted to m.v. *Alcinous,* is described and the results obtained are illustrated. A short film made on the *Alcinous* when at sea shows the main engines working at various speeds, and a number of shots with the engine developing full power, showing perfect combustion, are included.—*Paper by A. G. Arnold, read at a meeting of the Institute of Marine Engineers on 13th January 1953.*

Diesel Maintenance Control by Spectrographic Means

A number of publications in recent years have described the excellent results obtained through spectroscopic analysis of Diesel lubricating oils in controlling Diesel-engine maintenance. Avoidance of serious engine failures, longer life of component parts, and more economical locomotive operation have resulted from the work of various investigators in a new application of this scientific tool. The American Locomotive Company has been collaborating with eighteen railroad companies in extending this new technique in a broad-scale evaluation programme. The ultimate results of the programme appear most promising. This paper outlines the overall evaluation programme, describes laboratory equipment and technique, collection and interpretation of field data, and typical test results. That there is a definite place for the spectrograph in the maintenance of Diesel locomotives is shown to be a justifiable conclusion on the basis of experience gained thus far.—*Paper by H. R. Sennstrom, 1952 A.S.M.E. Annual Meeting, Paper No. 52-A-61.*

New High Speed Diesel Engine

There has just been developed by Stork Bros. and Company, Hengelo, Holland, also the Thomassen Engine Works of De Steeg, in co-operation with Prof. J. J. Broeze and Prof. B. C. Kroon, a new high-speed two-stroke engine running at 600 r.p.m. and built in powers of from 400 b.h.p. up to 1,450



Sectional elevation of the new Stork-Thomassen engine b.h.p., including 10-cylinder engines, some of which are under construction for the Royal Netherlands Navy. The engine has a bore of 240 mm. and a piston stroke of 360 mm. Uniflow scavenging is employed and the mean effective pressure at full output is 6 kg. per sq. cm. (85lb. per sq. in.). The scavenging air is supplied from a centrifugal blower running at about 10 times crankshaft speed, and a Vulcan-Sinclair coupling is incorporated in the gear transmission. The engine is provided with oil-cooled pistons. The most interesting feature of the design is the special form of combustion chamber, a development of the basic principle originated by Ricardo. It is of smaller diameter than the cylinder, so that the rotational speed of the air increases when the piston approaches top dead centre. The injection nozzle is of simple design, having a single hole 0·9 mm. in diameter. The cylinder liners are chrome-hardened on the Van der Horst process. The diameter of the exhaust valve in the cylinder head is 120 mm. and the operating mechanism is light and rigid. The push-rods are lifted by cams acting on sleeves, no cam rollers being provided.—*The Motor Ship, No. 33, February 1953; p. 492.*

Measurement of Total Oil Content of Boiler Feed Water

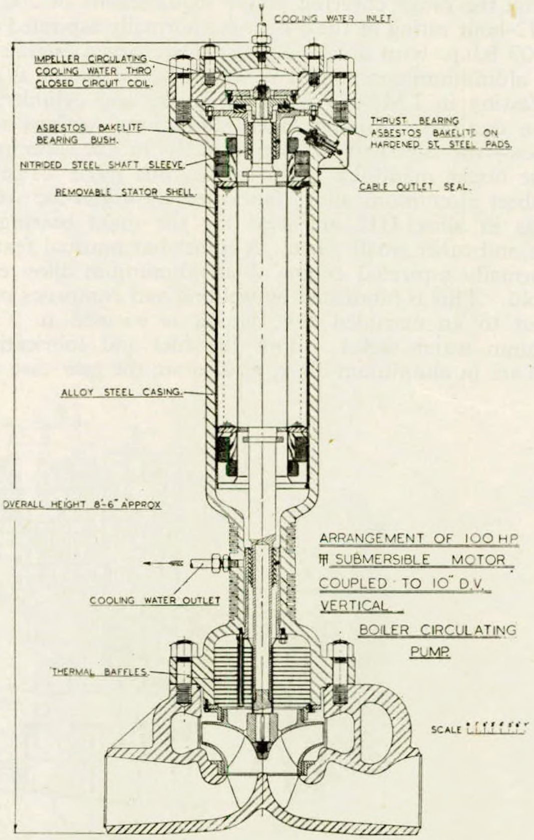
Details are given of the development of an instrument for determining the proportion of oil contained in the condensate from reciprocating steam engines, and the results of an investigation made on six vessels fitted with this type of machinery. The instrument developed at the Royal Technical College, Glasgow, makes use of the characteristic fluorescence of oil under ultra-violet light by comparing a solution of oil in ether, prepared from a sample of the feed water, with a previously prepared solution of known oil content under a mercury-vapour lamp. In the original form of the instrument visual comparison of the samples was made, and an improved apparatus of this type was used for assessing the oil content of all the samples obtained from ships in service. A later form of the fluorescence comparator employs photo-electric cells and a galvanometer for measurement of the degree of fluorescence, and hence the oil content, of the feed-water samples. It is considered that this instrument can be used with confidence; it eliminates the human element and has greater precision than the visual comparator. Samples were obtained from various positions in the feed systems of six ships, five of which were fitted with Scotch boilers and the sixth with watertube boilers. The water samples were taken from the air-pump discharge and the outlet side of the filter or filters, the intervals between sampling being determined by the filter-changing routine. Details are given of the filtering materials used, the frequency of cleaning, and the condition of the filtering media after use. The results of the tests made on the samples obtained indicate that the oil concentration in the condensate from triple-expansion engines of the orthodox type does not exceed 2 parts per million (p.p.m.). This low value was found to vary very little from one ship to another, whether superheated steam and cylinder lubrication were used or not. In the six vessels on which the tests were made, the feed water, on entering the boilers, had oil contents which, in general, ranged from 0 to 1·5 p.p.m. Some oil is removed by the conventional type of filters, but it appears that they are largely ineffective for the very low oil concentrations which, according to these tests, occur in practice.—*The British Shipbuilding Research Association, Report No. 69.*

Thickness of Tubes for Watertube Boilers

A comparison of the thickness of boiler tubes required by different classification societies both in this country and abroad, reveals that there is a wide difference of opinion concerning the optimum thickness of tubes for any particular diameter and pressure. The purpose of this paper is to examine the factors affecting the thickness of boiler tubes and to suggest a rational approach to the problem. Tube stresses and heat transfer are considered, including non-uniform heating such as occurs in tubes subject to furnace radiation. The rules of the principal classification societies are briefly reviewed, together with previous published work on this problem. A new approach to the problem is given. This defines the optimum thickness as that which will permit the maximum heat absorption rate, for a given working pressure, without exceeding a safe working stress. This idea is developed at some length in the paper, one of the most important conclusions being that the optimum thickness occurs when the pressure and thermal stresses are equal, i.e. the total stress is twice the pressure stress. A simple design formula based upon this conclusion is suggested. The work was carried out for the British Shipbuilding Research Association.—*Paper by D. W. Crancher, read at a meeting of the North-East Coast Institution of Engineers and Shipbuilders, 30th January 1953.*

Submersible Boiler Circulating Pump

The accompanying drawing shows a longitudinal section through a boiler water circulating pump which consists of an electric motor whose windings are immersed in the boiler water, and a pump driven by this motor. The combined motor and pump assembly is housed in a pressure casing designed to withstand the pressure of the system. The motor design is based on the Hayward Tyler electromersible "wet" motor, which has been in use for some time as the driving medium of the company's submersible pumping unit used in borehole and irrigation applications. The motor, a three-phase squirrel-cage induction type, is vertical. The windings are insulated with polyvinyl chloride of a special grade developed to withstand pressures up to 10,000lb. per sq. in. The rotor runs on journal tilted pad bearings, these pads having the same bearing properties as the Michell type thrust bearing at its top. This bearing takes the thrust and the weight of the moving parts and lubrication is provided by the boiler water. The three electric cables are led into the motor through patented cable entries which have also been designed for a working of 10,000lb. per sq. in. The rotor shaft extends from the motor through to the pump by way of a cooling neck, which incorporates several means of restricting the flow of heat from pump to motor. These include a reduction of shaft cross-section to reduce conduction, baffling



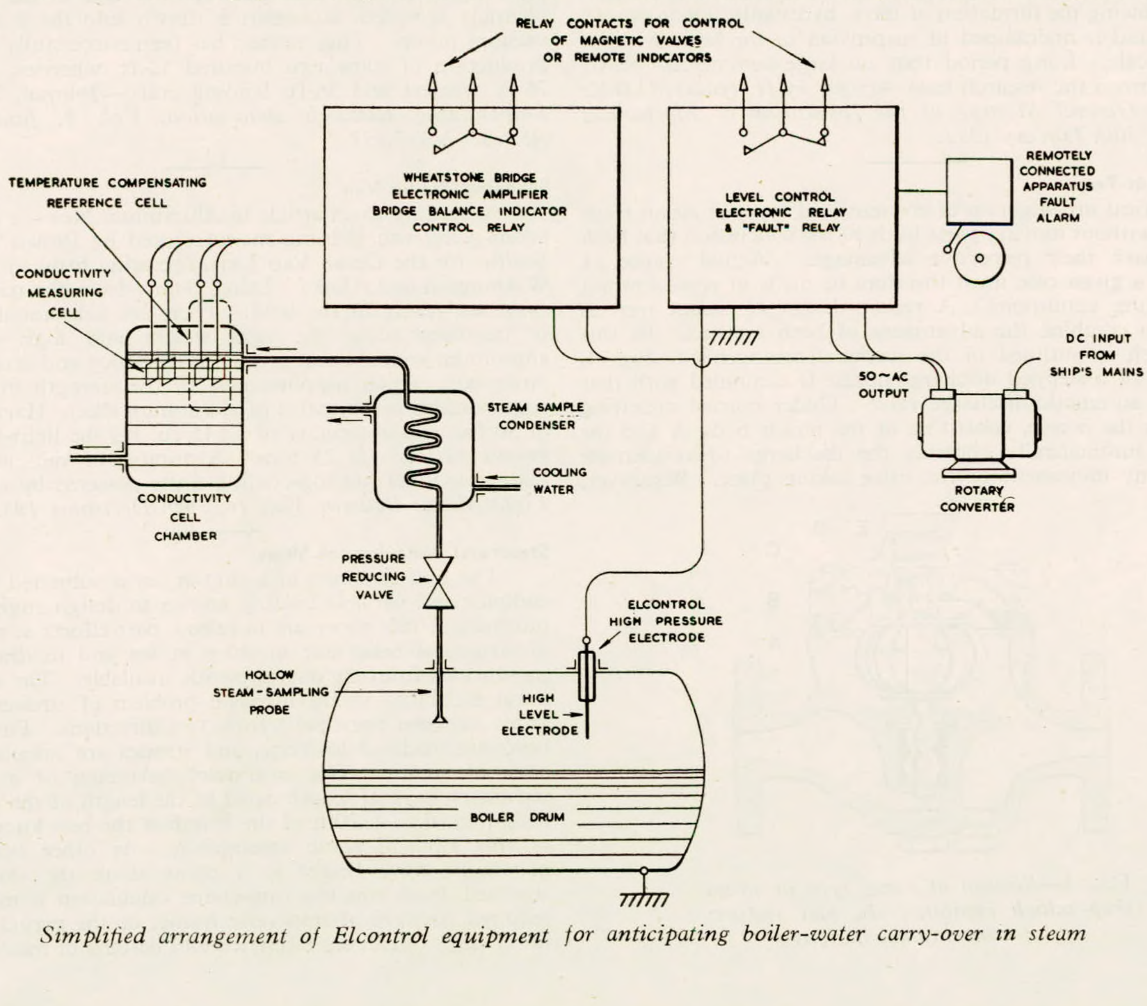
Longitudinal section through boiler water circulating pump to reduce convexion and a cooling jacket. In normal air motors the heat generated is removed by a fan, but in this case a small secondary impeller circulates the motor liquid through a high-pressure cooler, cooled by low-pressure water. As the motor is, in effect, in direct communication with the main boiler water system, the pressure in the motor is the same as that in the boiler system. Due to the restrictions previously mentioned, there is no flow of hot water from the pump to the motor. The motor cooling circuit has, therefore, to remove only the heat generated in the motor by copper and iron losses. A number of these pumps have already been fitted in marine boiler installations. The space taken up by a whole unit does not exceed 3 feet square by 8 feet high. The motor requires a three-phase electrical supply, and two or three gallons of cooling water at a pressure of approximately 15lb. per sq. in. At present the pumps are manufactured to suit temperatures up to 705 deg. F. and pressures up to 3,000lb. per sq. in. The maximum horsepower of the pumps so far produced is 125, and they weigh about 400lb. per h.p.—*The Shipping World, Vol. 128, 11th February 1953; p. 181.*

Control of Boiler Water Level and Solids Carry-over

The two most serious hazards which may be encountered in the operation of high-temperature steam plant are the sudden off-take of a large quantity of water with the steam, and the risk associated with the deposition of solids. In order to protect such plants from these risks, an entirely new type of electronic equipment has been designed by Elcontrol, Ltd., of London. A diagrammatic arrangement, showing the layout of the equipment is given in the accompanying drawing. An interesting example of this new equipment has been fitted in the Blue Funnel Line's *Nestor,* where it is designed to give increased protection of the superheater and turbines against the effects of solid carry-over. One set of equipment is provided for each boiler, and each set performs two functions; (a) continuous monitoring of the conductivity of the condensate; and (b) immediate warning of excessive foam or water-level in the drum. *Conductivity Alarm*—A sample of the steam, continuously drawn off from a convenient point in the boiler drum or steam-line, is passed through a small condenser, and the condensate is fed to an electronic conductivity controller. The equipment is calibrated, so that the actual conductivity of the condensate sample can be read at any time. Any undue increase in the solids carry-over will correspondingly increase the solids in the condensed-steam sample, thus raising the conductivity of the condensate. Variations will be immediately detected by the Elcontrol equipment, and the moment the conductivity rises to the pre-set permissible value, the relay in the controller will be operated. *Level Control*—A stainless steel probe is mounted in the drum by means of a specially designed high-pressure fitting. The length of the probe is arranged so that its tip is located at the maximum tolerated foam level in the boiler. The probe is connected, electrically, to an electronic relay unit, which is mounted in a convenient position. The probe forms part of a low-voltage A.C. circuit, which is completed (via earth) when the foam level makes contact with the probe tip. As soon as the circuit is completed, the electronic amplifier in the control unit is actuated and a relay is energized. The heavy-duty contacts of the relay are used to control an alarm bell, buzzer, or warning light. If desired, they may be used to control any other electrical circuit.—*The Shipbuilder and Marine Engine-Builder, Vol. 60, February 1953; pp. 118-119.*

Scale Formation in Sea-water Distilling Plants

The scales found in sea-water evaporators are formed of calcium sulphate, calcium carbonate, and magnesium hydroxide. Calcium sulphate scale can be avoided by using a sufficiently dilute brine concentration to maintain the sulphate in solution.



Simplified arrangement of Elcontrol equipment for anticipating boiler-water carry-over in steam

The conditions are indicated under which the formation of this scale can be avoided. Calcium carbonate and magnesium hydroxide scales are shown to be due to the carbonate alkalinity in the sea water, which is limited and reasonably constant. These scales are due to the break-up of the bicarbonate ions in the sea water as heating and boiling occur. Up to a well-defined range of operating temperatures, a scale that is predominantly of calcium carbonate is experienced. At higher temperatures, a scale predominately of magnesium hydroxide is formed; but both types of scale may occur in the region of the changeover zone. It is shown that the break-up of the bicarbonate ions produces carbonate ions, which give rise to the calcium carbonate scale and that, with further heating and boiling, break-up of the carbonate ions occurs with the formation of hydroxyl ions, which combine with magnesium ions to form the magnesium hydroxide scale. It is shown that some of the alkalinity leaves the evaporator with the blow-down; as calcium carbonate in solution, or calcium carbonate and/or magnesium hydroxide in suspension. The amount of scale formed is found to be proportional to the amount of sea water used, and no benefit is obtained by operating at low brine densities. It is shown that the rate of scale formation is greater for low temperatures than high temperatures, and increases with the temperature difference across the heating surface. Tests show that the use of organic dispersive compounds gives rise to a weaker scale structure, which allows some of the scale to be shed by cracking. The calcium carbonate and magnesium hydroxide scales can be completely prevented if the appropriate quantity of hydrogen ions is supplied by the injection of acids, such as hydrochloric acid or sulphuric acid, or an acid salt such as sodium bisulphate. These scales can also be prevented by the injection of ferric chloride, which provides a supply of ferric ions. The ferric ions combine preferentially with the hydroxyl ions formed by the break-up of the bicarbonate and carbonate ions —thus allowing the formation of ferric hydroxide, which is very insoluble and is maintained in suspension in the brine without forming scale. Long period tests on large commercial plants have confirmed the research tests.—*Paper by H. Hillier, O.B.E., read at a General Meeting of the Institution of Mechanical Engineers, 30th January 1953.*

Novel Steam Trap

A critical investigation of conventional types of steam traps with and without moving parts leads to the conclusion that both systems have their respective advantages. Actual choice of system in a given case must therefore be made in consideration of prevailing conditions. A recent design of steam trap is claimed to combine the advantages of both systems. In this trap, which is outlined in the sectional arrangement, Fig. 4, the action of a stepped discharge nozzle is combined with that of a fully automatic discharge valve. Under normal operating conditions the nozzle, consisting of the nozzle body A and the valve B, automatically controls the discharge of condensate without any movement of the valve taking place. Whenever, however, the amount of condensate to be discharged exceeds the discharge capacity of the nozzle, the discharge area is automatically increased. The operating principle in this case is based upon the fact that excessive accumulation of condensate will have to be dealt with when resuming operation after service interruptions, that is to say, when the condensate has cooled down to temperatures considerably below those encountered under normal service conditions. Temperature responsiveness of the trap is achieved by having the valve stem carried by a system of curved bimetal strips C which lift the valve as the water temperature decreases. Undue deformation of the bimetal strips is prevented by the provision of a leaf spring D which keeps the bimetal strips from lifting the valve further when it has reached the full-open position.—*Brennstoff, Wärme, Kraft, Vol. 4, December 1952; p. 432. Engineering and Boiler House Review, Vol. 68, February 1953; pp. 58-59.*

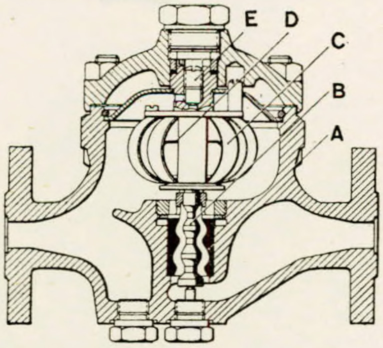


FIG. 4—*Section of novel type of steam trap which combines the best features of conventional designs*

Four Ways of Building Plastic Boats

An article by J. B. Alfers in a recent issue of Modern Plastics refers to the fact that the United States Defence Department has developed methods of making small boats out of plastics. Four methods of moulding hulls are described. Two involve a single mould: the hand lay-up and bag methods. Two involve two matching moulds: the injection and matched-die methods. In the hand lay-up method, glass, wetted by a resin, is laid layer by layer in a mould (male or female) and pressure is applied by rolling and squeegeeing. In the bag method, the plastic is compressed by a flexible pressure member, usually a bladder, operated by compressed air. The matched-die high-pressure method has been proposed but not yet used for boat hulls. It is a repetitive method, using a pair of matched moulds operated by a press. The injection (Marco) method makes use of a pair of matched moulds of rigid, air-tight construction. The glass fibre is laid in the mould, the assembly is sealed, and resin is drawn into the assembly by a vacuum pump. This method has been successfully used in the production of some two hundred 12-ft. wherries, and several 26-ft. whalers and 36-ft. landing craft.—*Journal, The British Shipbuilding Research Association, Vol. 8, January 1953; Abstract No. 7,033.*

Seagoing Trailer Van

According to an article in Aluminium News, a new type of ocean-going van is being manufactured by Brown Trailers, of Seattle, for the Ocean Van Line, operating between the state of Washington and Alaska. Lifted bodily from the trailer chassis, they nest neatly in the hold. There are substantial reductions in handling costs, the light weight and high strength of aluminium contributing greatly to efficiency and economy. The outer skin, which supplies most of the strength in the monocoque design, is fabricated of aluminium sheet. Having a length of 30 feet and a capacity of 1,545 cu. ft., the light-weight vans permit payloads of 25 tons. A number of vans are equipped with individual heating-cooling units powered by electricity.—*Light Metals Bulletin, Vol. 14, 19th December 1952; p. 1,018.*

Structural Behaviour of Ships

The hull structure of a ship at sea is subjected to the most complex and variable loading known to design engineers. The purposes of this paper are to review past efforts at observations of structural behaviour in ships at sea and to draw tentative conclusions from the data presently available. The approach of naval architects to the intricate problem of stresses in a ship at sea has been principally from two directions. First, assumptions are made of loadings, and stresses are calculated on the basis of statics. The customary balancing of a ship on a stationary wave of length equal to the length of the ship and of height equal to 1/20th of the length is the best known example of this kind of static assumption. As other examples, the maximum wave height at a point along the ship's side is assumed, from which a ring-frame calculation is made for the required strength of transverse frame; or the period and maximum angle of roll are estimated and designs of masts and other structures are based on calculated inertia forces and gravity components. The second direction of approach has been the evaluation of service experience. Throughout the years, structural weaknesses which have developed in ships at sea have been studied, corrective measures have been taken, and design formulas modified accordingly. The design method, based on static assumptions and on service experience, has produced generally effective hull structures believed to be reasonably well engineered to carry out their functions. However, the method is not entirely satisfying. Possibly some hull structures are still heavier than need be, and represent dead weight that could be more profitably carried in cargo or armament. Perhaps some of the usual static assumptions for the hull girder and its components do not truly represent actual structural loading patterns in a ship at sea. Consequently, even when structural weaknesses do appear, a rational analysis is not always possible because the very nature of the loading may be incompletely defined. For the sake of realism, and in appreciation of the continuing need for improvements in hull design, the naval architect looks at the structural behaviour in ships at sea with intense and objective curiosity to learn more about the actual loadings to which his structures are subjected. Recognizing the paucity of observations of structural behaviour at sea and the hazards of drawing broad conclusions from limited data, the authors have examined the service observations from the designer's point of view, and have concluded that these are the following trends: (a) Maximum hull stresses occur in waves whose length is approximately equal to the length of the ship. (b) Observed stresses are generally less than calculated stresses based on an L/20 static wave. (c) Stresses are greater when a ship is in a wave hollow than when a ship is on a wave crest. (d) Stresses due to wave impact are strongly influenced by the relative velocity of ship and wave, increasing as the relative velocity increases. (e) In combined pitching and rolling, as when crossing the seas obliquely, stresses are very much less than those in pitching only. (f) Waves in the order of 400 to 500 feet long may have wave height to length ratios as high as 1/14, compared to a ratio of 1/20 used in the usual longitudinal strength calculation. (g) Maximum stresses amidships in ships ranging between 300 to 500 feet in length in severe weather have been in the order of 4 to 6 tons per sq. in. (h) Strain and motion data from ships at sea are difficult to analyse, and the statistical approach appears to be the only practicable one.—*E. A. Wright and J. Vaster, Journal of the American Society of Naval Engineers, Vol. 64, November 1952; pp. 693-706.*

Hull Design of Welded Ships

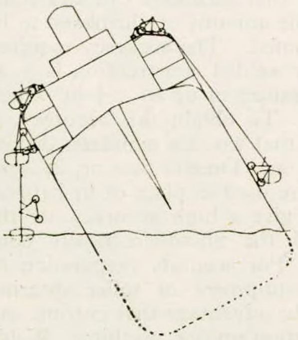
The following is an abstract of a paper by A. Svennerud, read at the Plenary Session of the International Institute of Welding, Gothenburg, 1952. The author points out that the extension of the use of welding in shipbuilding in recent years has increased productivity and reduced the building time. Nevertheless, the advantages welding presents in the field of ship construction have not as yet been fully realized. Although welding simplifies the construction of the majority of components, it also unfortunately results in a reduction of the longitudinal stiffness of the vessel, since it reduces the resistance to axial bending of the plating of a transversely-framed vessel. On the other hand, welding facilitates the use of longitudinal framing, and it appears desirable to use this system on major vessels of all types. For the larger tankers and similar vessels, in which longitudinal framing will be used in any case, it appears desirable to increase the area of the deck girders so that the thickness of the deck plating can be reduced; this is of advantage because the thinner plating is likely to be less notch-sensitive, and also because the openings found necessary in the deck plating will not have an unfavourable effect on the strength. A number of detail designs are discussed and criticized from the viewpoints of both theoretical strength and production in practice. The suitability of various sections for welded construction is considered. Some points to be borne in mind when designing for prefabrication are noted, and the author briefly discusses the extent to which riveting should be incorporated in a welded ship. The paper closes with brief details of two recent Götaverken designs for welded ships, the one a tanker of 17,500-tons d.w., and the other a dry-cargo vessel of 9,500-tons d.w.—*Journal, The British Shipbuilding Research Association, Vol. 8, January 1953; Abstract No. 7,007.*

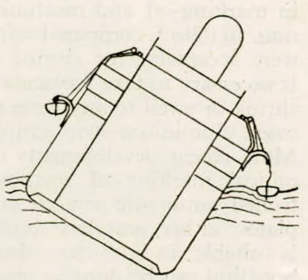
Crack Arresters

The following is an abstract of a paper by B. Persson, read at the Plenary Session of the International Institute of Welding, Gothenburg, 1952. Despite the progress made in recent years in the design of welded ships and in the execution of the welds themselves, and despite the improved quality of the steel which is now available, riveted crack arresters are still frequently employed. These crack arresters are simple and effective, but when a shipbuilding yard is concentrating on the production of welded ships, even the small amount of riveting required may prove an inconvenience. The author draws attention to a report on an investigation of fractured steel plates removed from welded ships, which was performed under the auspices of the Ship Structure Committee. The impact values given in this report are summarized, and it is shown that the higher the impact value the less tendency there is for a crack to be propagated. From the results, the author develops a curve which shows that a steel with an impact value of 20 to 25ft. lb. would prevent further propagation of a crack, and would therefore act as a crack arrester. Steels of a better quality than the ordinary ship steel supplied by the different steel manufacturers vary considerably both in quality and price, but it is possible to obtain a steel which meets the required specification, and which is offered at a price that enables it to compete with the combination of ordinary steel and crack arrester. A standardization of the requirements of the various classification societies would go far towards easing difficulties of supply. The author concludes with brief details of the design of crack arresters adopted by the firm of Eriksbergs Mek. Verkstads.—*Journal, The British Shipbuilding Research Association, Vol. 8, January 1953; Abstract No. 7,008.*

List-proof Davits

A new type of davit, the invention of a Dutch merchant service master, Captain Vreughendil, has been in successful service for some time in the Royal Rotterdam Lloyd motorships *Weltevreden* and *Kota Inten*. There are actually three alternative types of these davits, two intended for larger ships and the third for coasters and other vessels with lower freeboard. All three types are designed to allow boats to be lowered against a list of the parent ship, and also to cushion the boat should it strike the side of the ship due to the latter rolling. The first two types of davits are illustrated below. In one ("Listproof I") the boat is carried in a four-wheeled transporter, which is normally stowed on inclined trackways. The transporter consists of a pair of triangular frames, connected by a bar, each carrying two 3-ft. wheels. The boat is slung beneath the bar by means of pulleys, and the falls, which are of wire, are rove through these pulleys, the standing end being secured on deck and the running end being coiled in a single layer on the boat winch. This is provided with a hand brake, a cen-





"Listproof I" (left) and "Listproof II" (right) patent davits

trifugal brake and an emergency brake. The brakes can be controlled from within the boat if necessary. The method of lowering can be seen from the diagram. The transporter moves downwards with the boat, either protecting it from striking the side of the ship or running down the shell plating, until it is automatically arrested when the boat is a few yards from the water. The boat is then lowered to the water by operating the handbrake on the winch, and slipped by disengaging gear. On rehoisting, the boat is raised until it meets the transporter, which it carries up with it. The length of the davit exceeds that of the boat by some 2ft. The "Listproof II" davit works on the same general principle, but the transporter running on wheels is replaced by two connected skids, which slide down the ship's side. This gives a davit which is shorter than the boat. In the "Listproof III", intended for vessels of lower freeboard, the boat is carried between a pair of rolling quadrants. As the boat is lowered, these rock outwards, and, when at the end of their travel, hold the boat well out from the ship's side while it is lowered.—*The Shipping World, Vol. 128, 21st January 1953; p. 119.*

4,800 Horsepower Diesel Towboat

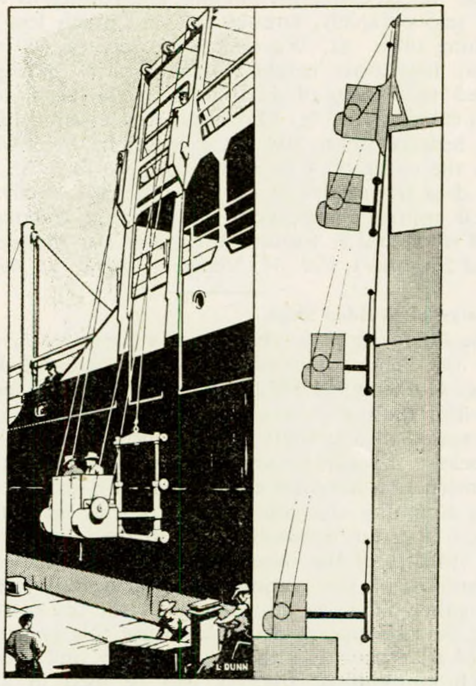
The 4,800-horsepower triple-screw Diesel towboat *Aetna-Louisville,* most powerful towboat on the Ohio-Mississippi oil-hauling route, is engaged in towing oil from terminals near New Orleans up the Mississippi and Ohio Rivers to the 45,000 barrel-per-day refinery near Ashland. A round trip of 3,274 miles takes about twenty-one days. The *Aetna-Louisville* has a length of 150 feet, a beam of 50 feet, and a draft of 10ft. 6in. Transverse framing is used with six transverse bulkheads and twenty-three watertight compartments. A V-bottom is a feature of the design, due to excellent results obtained on similar construction of the balance of the Ashland Oil and Refining Company's fleet. Welded construction was used throughout the boat. The engine room is insulated with Johns-Manville rock wool with perforated marine veneer. Other construction provides for 2-in. fibreglass insulation covered with masonite. Bottom and side plating is $\frac{3}{8}$-inch. Bilge strakes and tunnel plates are $\frac{1}{2}$-inch. Deck plating is $\frac{5}{16}$-inch and $\frac{3}{8}$-inch. The boat is unusually strong throughout and exceeds usual specifications. The vessel is powered by three Model 16-278A General Motors Cleveland Diesels, developing 1,600 h.p. each at 750 r.p.m. For short periods these three engines develop a total of 5,400 h.p. at 800 r.p.m. Three Baldwin bronze propellers 96 inches in diameter and 88 inches pitch are mounted inside Kort nozzles for peak power.—*Marine Engineering and Shipping Review, Vol. 57, December 1952; p. 96.*

Workmanship Requirements for Welding in Shipbuilding

This is an abstract of a paper by H. C. Steffensen, read at the Plenary Session of the International Institute of Welding, Gothenburg, 1952. The author briefly recapitulates the advantages of welding over riveting in shipbuilding. The outstanding problem today is how to weld rather than whether to weld or rivet. The residual stresses due to welding may cause serious distortions which require a considerable amount of work to put right; to prevent this distortion it is necessary to plan the sequence of welding in such a way that shrinkage can take place without restraint. Examples of the amount of shrinkage to be expected in particular cases are quoted. The accuracy required in marking-off and machining for welded construction is ± $\frac{1}{2}$-mm. (0·02in.), compared with tolerances of up to ± $\frac{1}{8}$-in. which were acceptable for riveted ships. To obtain this accuracy, it is necessary to take particular care that wooden templates do not shrink or swell to any large extent, and Oregon pine or, in some cases, aluminium-strip templates are used in place of fir battens. More recent developments which give a high accuracy are the optical marking-off method, and the photo-electrically controlled automatic oxygen cutter. For accurate preparation of plates, either acetylene burning equipment or roller shearing is suitable; in both cases there is the advantage that cutting and bevelling can be done as one operation on one machine. Welding skids or surfaces are essential for prefabrication. They must be perfectly flat; they must be stiff enough to avoid being pulled out of shape by the welding of the sections; and they should be made so that the sections can be held firmly by simple clips. A number of types of skid which have proved suitable are illustrated. The author then illustrates, with the aid of a plan, the routes traversed by a deck plate and a deck beam from the storage areas to the erecting berth. Finally, the necessity for employing steels of better quality than those suitable for riveted construction is menioned.—*Journal, The British Shipbuilding Research Association, Vol. 8, January 1953; Abstract No. 7,009.*

New Tankers

The new Shell 18,000 tons deadweight general purpose tanker *Helix* and her sister ships embody in their construction a number of interesting new features, developed by the Company's Marine Research Department. These include unusually complete safety arrangements against fires or explosions, a new layout of cargo pipelines, and a novel disposition of the cargo pumps. Perhaps the most original feature is the electrically-operated overside lift, which is to replace the normal gangway in these ships. This lift, which is the subject of patents, has been developed by the Group's Marine Research Department, and is constructed by Clarke, Chapman and Co., Ltd. It is capable of carrying six persons or 10 cwt. of stores at a time, is not affected by the list or trim of the vessel and does not depend upon guide rails. It will ease considerably the difficulty of getting on or off an oil tanker, which through its widely varying draughts, often necessitates the gangway being tilted to a very steep angle. As such, it should reduce the number of gangway accidents. It is also expected to prevent rats entering or leaving the ship by this means. The construction of the lift is shown in the accompanying illustration, from



New Shell overside lift

which it will be seen that the action of lowering it causes rams to push the body of the lift away from the side of the ship, ensuring that it comes down safely on the jetty. The principal aim of the new safety features in the ship is to prevent the escape of gases from the cargo tanks. This is achieved by special equipment which makes it unnecessary to uncover any opening in the tanks, either for loading and discharging or for sounding purposes. If, however, an accident occurs and dangerous gases do escape, they are prevented from entering those parts of the ship where naked lights are in use, such as the galley, the boiler room and the smoke room. It is claimed that there will be no greater risk of fire in these tankers than in any other type of ship. The principal particulars of the *Helix* are as follows:—

Length b.p.	530ft. 0in.
Breadth, moulded	69ft. 3in.
Depth, moulded	39ft. 0in.
Load draught ...	about 29ft. 7⅛in.
Load deadweight ...	about 18,000 tons
Gross tonnage ...	about 12,000 tons
Capacity of oil cargo tanks at 50 cu. ft. per ton, 98 per cent full	about 17,600 tons
Service speed	14½ knots

The single-screw turbo-electric propelling machinery for the *Helix*, together with all control gear, has been constructed by the British Thomson-Houston Co., Ltd., Rugby. The boilers and other contingent parts are being constructed by The Wallsend Slipway and Engineering Co., Ltd. The two main alternators each consist of an impulse turbine direct-coupled to a two-pole alternator, supplying three-phase alternating current to an electric motor directly connected to the propeller shaft. The propeller motor has a maximum output of 8,300 s.h.p. at 103 r.p.m. and a normal service output of 7,500 s.h.p. at 100 r.p.m. Two main condensers of the Weir regenerative type are to be fitted, one for each main alternator turbine, underslung from the turbines. The condensers are to be capable of maintaining a vacuum of 28½in. Hg., with sea water at 75 deg. F. when the machinery is developing full service power. The propeller is of four-bladed type, of solid manganese bronze.—*The Shipping World, Vol. 128, 28th January 1953; pp. 132-134.*

Removable Platform for Ships' Holds

A method of sub-dividing ships' holds by removable platforms is shown in Fig. 3. Three compartments (4, 5, 6) are obtained by the provision of two longitudinal bulkheads (2, 3). The panels (7) are pivoted at the side (8) of the hold, hinges

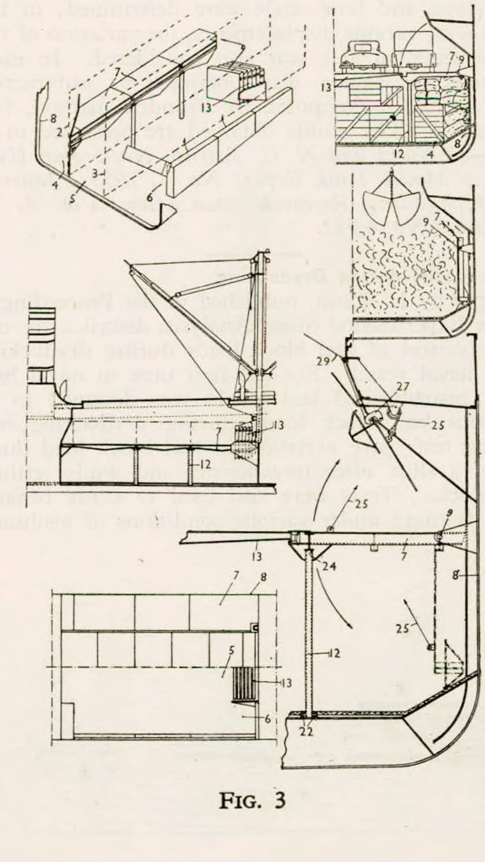


FIG. 3

(9) being provided. The stanchions (12) are fitted in sockets (22, 24) and are removable. The panels (7) are dropped when the hold is to be laden in bulk with loose cargo and to return them to their horizontal position, a cable (25) is provided. The operation of raising the panels may be used to advantage in unloading the cargo. The central section (5) of the hold is closed by panels (13), which are of the rolling type and are stowed together in a vertical position. For latching the pivoted panels (7) in their raised position, a hook (27) is provided. After the panel has been raised, the cables (25, 29) are released and the panel bears by gravity on the hook (27).—*British Patent No. 684,455, issued to H. Kummerman. The Motor Ship, Vol. 33, February 1953; p. 490.*

Moment Distribution Method of Analysing Tanker Framing

When the design of an oil-carrying vessel is under consideration, the arrangement of the main transverse members and the arrangement of the main members supporting the bulkheads, both longitudinal and transverse, may be of several forms. The final choice depends to some extent upon the individual preference of the designer but some consideration of the loading and stressing of the members may indicate certain fundamental factors which affect the choice of design. Even if no consideration is given to dynamic loading—which in a tanker must be of considerable magnitude, due to the fluid nature of the cargo—the stress system in the transverse members is extremely complicated and only broad conclusions can justifiably be drawn from any method of calculation at present available. While, therefore, for purposes of comparison it may be necessary to quote figures of bending moments obtained from mathematical calculations, any stresses derived from these may be wide of the mark if it is desired to determine the actual stress figures for a particular member. It is, however, claimed that the calculations will enable the form of distribution of bending moment to be determined and will show up the points where peak stresses occur and the relative magnitude of these peak stresses. The investigations of the author were carried out with the employment of the Hardy Cross method, which has become well known in recent years. This method of analysis is applied by the author to certain tanker frame sections including (1) a tanker with no strut in the wing tank; (2) a tanker with diagonal stiffening; and (3) a tanker with staggered wing and centre-tank bulkheads. The value of the strut in the wing tank is analysed, the effect of altering the position of the strut is investigated, and the comparative value of two struts or one strut is examined. Some general conclusions are drawn and attention is directed to the faults in the assumption that ship members may be considered to have fixed ends and may be based on an arbitrary length.—*Paper by H. J. Adams, read at a meeting of the North-East Coast Institution of Engineers and Shipbuilders on 9th January 1953.*

Accuracy of Calculated Stability Curves

In a paper read at the Second International Congress of Naval Architects at Ostende, J. W. Bonebakker explains that several methods are in common use for calculating ship stability curves, which, although formerly considered to be exact, now exhibit fairly large uncertainties. The author discusses the reasons for these inaccuracies and refers to the work of Prohaska, Råhola, and others who have analysed the problem. There are many seagoing vessels whose stability characteristics will always be safe provided their design conforms to standard practice; for example, tankers and dry-cargo ships of the open shelterdeck type, excluding sizes below about 2,000-tons gross in both instances; but the smaller the ship, the more difficult it is to determine the safety margin, and the more important it is to do so. This is illustrated by a numerical example which compares the stability of a small shelterdeck vessel of 1,200-tons deadweight and a 9,000-ton deadweight cargo liner. This representative example illustrates the need for determining really exact stability curves for small vessels. In the second section of his paper, the author briefly describes an apparatus made at Delft University for ascertaining accurately by means of model experiments the position of the centre of buoyancy for any displacement or angle of heel, taking into account the influence of camber, sheer, hatchways, and effective superstructures. This apparatus sets a standard for judging the accuracy of both approximate and the so-called "exact" methods for calculating stability curves.—*Journal, The British Shipbuilding Research Association, Vol. 8, January 1953; Abstract No. 6,993.*

Tunnel Type Vessels

The paper describes the general principles of the tunnel vessel. The upper part of the tunnel is considerably above the waterline, thus enabling the diameter of the propeller to be greater than the draught of water. When the vessel is at rest the water level inside the tunnel is naturally the same as it is on the outside; but when the propeller begins to revolve, the air which is enclosed in the upper part of the tunnel is forced out and replaced by solid water. It will be seen that by this means a large propeller, capable of utilizing considerable power, can be used in combination with a shallow draught. The tunnel is sealed on all sides, this being necessary because when once the air is forced out it must never be allowed to pass in or the propeller would not be working in solid water. The author traces the advances made in the design of such vessels over the past fifty years and illustrates the large variety of shallow-draught craft that have been fitted with tunnels and are operating successfully on inland waterways throughout the world. Tank tests, trial results, towing, steering and varying forms of tunnel, with or without flaps, are also dealt with.—*Paper by H. R. Mitchell, read at a meeting of The Institution of Engineers and Shipbuilders in Scotland, 13th January 1953.*

Influence of Draught on Propulsive Qualities

The paper gives an account of tests carried out at the Swedish State Shipbuilding Experimental Tank in Göteborg. The experiments were conducted in order to study the influence of the draught of a ship on the propulsive qualities of the propeller. They were not systematic, but were intended to provide a comparison between one practical case and the relatively rich material in literature. The original tests were carried out for Helsingborgs Varfs A. B., Halsingborg, with a paraffin wax model of a cargo ship with the displacement $V = 3{,}788$ m.^{3} and with the calculated trial trip speed $V = 12$ knots. Afterwards the tank extended the tests with the same model. Both resistance tests and self-propulsion tests were carried out at the draughts with even keel while only self-propulsion tests were run in the trimmed condition. All the self-propulsion experiments were carried out according to the so-called "Continental method" (Gebers) with the skin-friction correction applied as a towing force. The results have been converted to full scale in the conventional way. No turbulence stimulator was used during the tests.—*H. Edstrand, Publications of the Swedish State Shipbuilding Experimental Tank, No. 21, 1952.*

Italian Liner Enters Service

Italy's largest and fastest post-war passenger ship, the 29,083 tons gross *Andrea Doria* takes the name of a distinguished Genoese admiral and is to be followed by a similar vessel, not yet launched, to be named *Christoforo Colombo.* The appearance of the new ship follows closely that of the CRDA-built motorships, with a beaked stem, bulbous forefoot, samson posts before, and a single mast stepped above the bridge, massive funnel and superstructure decks tapering off at the stern. The hull form has resulted from self-propelled tank experiments at the Nautical Academy in Rome, prior experience having been obtained with smaller models at Genoa. Certain parts of the vessel have been prefabricated by electric welding and most of the upper deck erections are of aluminium alloy; as also are the two motor lifeboats and fourteen Fleming-powered boats. The *Andrea Doria* has been built to the highest class of the Registro Italiano, Lloyd's Register of Shipping, and the American Bureau of Shipping. She has the following principal particulars:—

Length overall...	697ft. 0in.
Length b.p.	626ft. 8in.
Moulded breadth	89ft. 11in.
Moulded depth to main deck	49ft. 10in.
Gross tonnage	29,083 tons
Service speed (35,000 s.h.p.) ...	23 knots
Contract trial speed at light draught (50,000 s.h.p.) ...	25·3 knots

The twin-screw three-casing Ansaldo-Parsons geared turbines are designed to develop a total service power of 35,000 s.h.p. at 143 r.p.m. and trial rating of 50,000 s.h.p. at 164 r.p.m.—*The Marine Engineer and Naval Architect, Vol. 76, January 1953; pp. 3-4.*

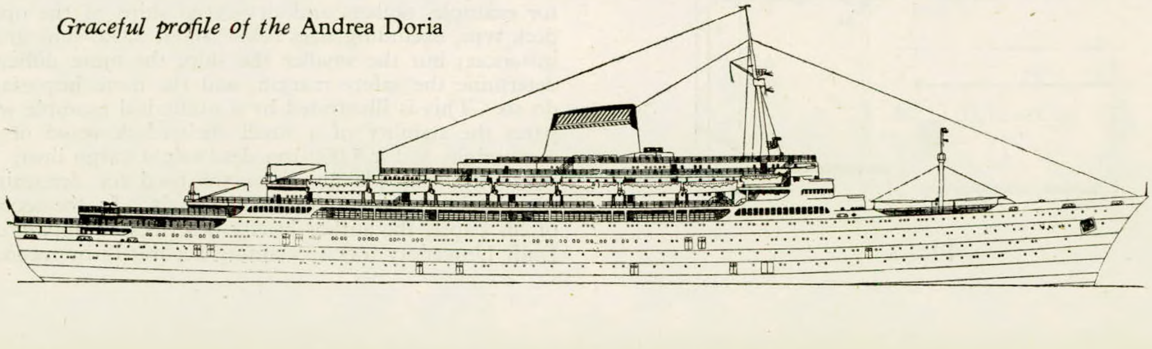
Model Tests on Fishing Vessels

During investigations into the resistance characteristics of a series of fishing vessels, it was found that one of the hulls tested showed very favourable results. Further tests have now been performed on this hull with the object of determining the effect on the resistance of variations in the displacement and in the beam-draught ratio. The models were of wax, and were made ⅛th full size. Turbulence was ensured by means of a trip wire in the usual position. The tests covered ship speeds from 6 to 12 knots. Six models were made for investigating the effects of displacement, all with the beam-draught ratio of the original model; and five models, all with the same displacement as the original model, were built with various beam-draught ratios. In each series of tests, the effects of variations in beam, depth, wetted surface, and bow angle were determined; in the tests performed with various displacements, the variation of the midships cross-sectional area was also considered. In each case, the residuary resistance was obtained by subtracting the frictional resistance, computed by Froude's method, from the total resistance. The results obtained are presented in graphical form.—*E. Sund and N. C. Astrup, Norwegian University Tech., Ship Model Tank Report No. 7, 1951. Journal, The British Shipbuilding Research Association, Vol. 8, January 1953; Abstract No. 6,982.*

Keel Block Loads during Drydocking

A paper by E. Wenn, published in the Proceedings of the Society for Experimental Stress Analysis, describes the measurement and control of keel block loads during drydocking tests of a U.S. naval vessel. For the first time in naval history, a system of instrumented keel blocks was designed to measure instantaneous keel block loads during drydocking of ships. Drydocking tests gave variation in keel block load during the landing of a ship, after unwatering, and while settling of a ship on blocks. Tests were also used to study behaviour of transverse framing under various conditions of loading and to

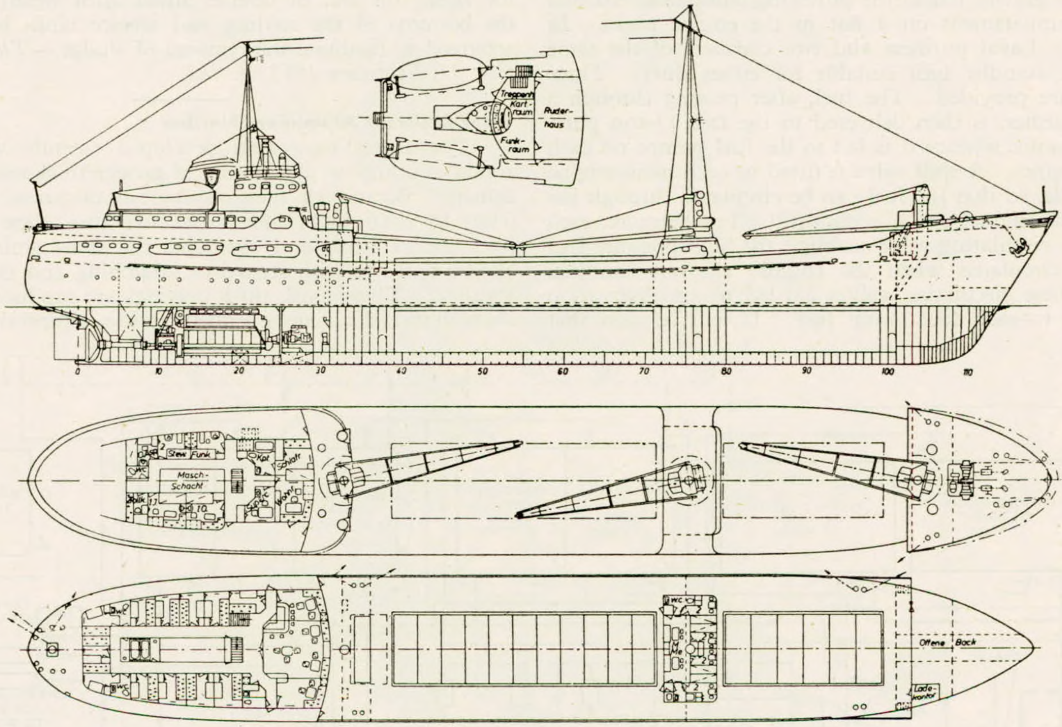
Graceful profile of the Andrea Doria



design, thereby, the size of the framing structure to ensure more uniform stress distribution under maximum load.—*Applied Mechanics Review, Vol. 6, January 1953; pp. 47–48.*

German Cargo Motorship

The cargo motorship *Ciandra* is the first of two vessels built by F. Lürssen Werft, Bremen-Vegesack for Transatlanta Reederei, Hamburg. The leading particulars are 76·8 m. length o.a., 68·2 m.b.p., 11·0 m. breadth moulded, 1,550 tons d.w., normal speed 12 knots. The vessel is designed with a small enough draft to permit navigation of the Great Lakes. The hatches are fitted with MacGregor covers and one 5 ton and two 3 ton electrically equipped deck cranes are provided which should prove especially useful in small ports where no cranes are installed. Propulsion plant is arranged aft and consists of an eight cylinder four-stroke Diesel engine of M.A.N. make with B.B.C. exhaust turbo charger for 40 per cent supercharging. The engine is rated at 1,200 b.h.p. at 260 r.p.m., cylinder diameter being 385 mm. and stroke 580 mm. There is a 30 kW. generator driven from the main shaft and a 20 kW. auxiliary generator. Two Diesel driven generators of 75 kW. each are provided for supplying electric current to the deck machinery.—*Schiff und Hafen, No. 12, 1952; pp. 537-542.*



Cargo motorship Ciandra

Diesel Lubricant Performance

The electron microscope, the paper states, can and should be used to control all received shipments of additive-type oils as to the maintaining of a normal additive pattern dispersity. Any unusual variation should result in a re-evaluation of the oil so as to prevent engine failures. It was found that minute residual amounts of such impurities as fatty acids, sodium, magnesium, and oxidizing agents change additive structure and completely revert a good engine operation to a very poor operation with excessive wear, deposits, or corrosion conditions. It was discovered as with the spectrograph that a scientific instrument used largely for research can be used practically and will give basic knowledge of engine conditions. Previous conceptions as to what a satisfactory Diesel-engine lubricating oil should be is changing, as knowledge and studies are increased in this field of colloidal and physical chemistry. Then, with the background of "letting the engine tell what is occurring", many of the apparently misunderstood happenings with lubricating oils will be solved, the paper indicates. The mixing of various types of additive oils may result in very serious problems, and a much greater understanding of basic fundamentals is needed, towards which the use of the electron microscope, in its infancy, is pointing the way.—*Paper by R. McBrian, 1952 A.S.M.E. Annual Meeting; Paper No. 52-A-40.*

Water-lubricated Thrust Bearings

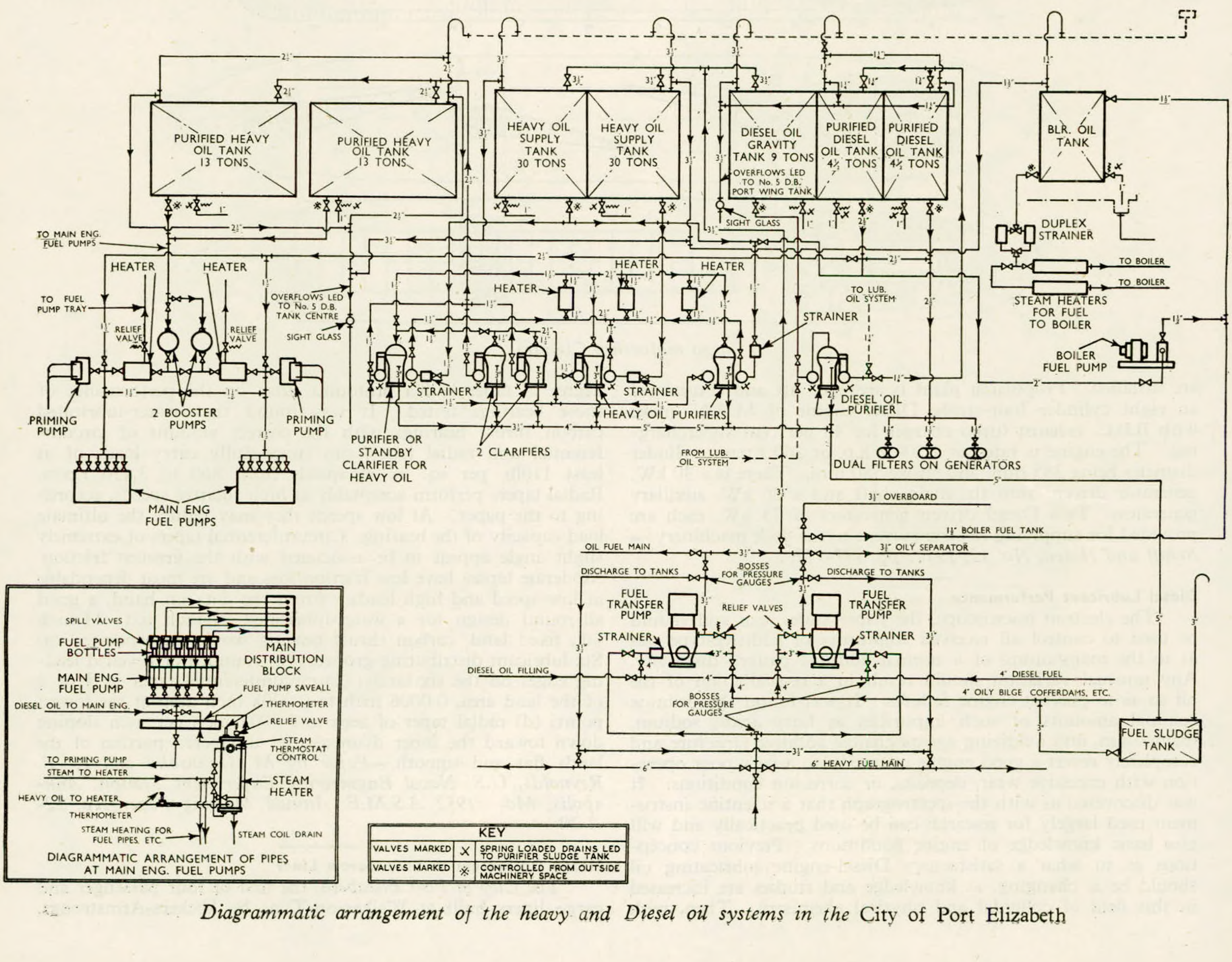
Experiments were made to determine the load-carrying ability of water-lubricated, tapered-land, carbon thrust bearings. Nine different bearings having various combinations of radial and circumferential tapers were tested. As was expected, the degree of taper had a profound effect on the performance of those bearings tested. It was found that water-lubricated carbon thrust bearings with the correct amount of circumferential and radial taper, can successfully carry loads of at least 110lb. per sq. in. at speeds from 860 to 3,450 r.p.m. Radial tapers perform acceptably at high rotative speeds, according to the paper. At low speeds they may reduce the ultimate load capacity of the bearing. Circumferential tapers of extremely slight angle appear to be associated with the greatest friction. Moderate tapers have low friction loss and are most dependable at low speed and high loads. From the data on hand, a good all-round design for a water-lubricated $2\frac{3}{4}$-inch o.d., $1\frac{3}{4}$-inch i.d., fixed land, carbon thrust bearing would incorporate: (a) Six lubricant distributing grooves; (b) rounded or bevelled leading edges on the six lands; (c) circumferential taper in $\frac{2}{3}$ or $\frac{3}{4}$ of the land area, 0·0006 inch to 0·0008 inch deep at the lowest point; (d) radial taper of zero to 0·0003 inch per inch sloping down toward the inner diameter; (e) untapered portion of the lands flat and smooth.—*Paper by M. Levinsohn and N. E. Reynolds, U.S. Naval Engineering Experiment Station, Annapolis, Md. 1952 A.S.M.E. Annual Meeting Paper No. 52-A-29.*

Heavy Oil System in Twin Screw Liner

The *City of Port Elizabeth*, the first of four passenger and cargo liners built at Walker-on-Tyne by Vickers-Armstrongs, Ltd., for Ellerman Lines, Ltd., is a ship of 12,500 gross tons, propelled by twin-screw Hawthorn-Doxford machinery capable of developing 12,650 b.h.p. at 115 r.p.m. Each engine has six cylinders, having a bore of 670 mm. and a piston stroke of 2,320 mm. and is designed to run on heavy fuel with a maximum viscosity of 3,500 seconds Redwood I at 100 deg. F. As the diagram shows, there are two heavy oil supply tanks, each with a capacity of 30 tons and filled by means of the Worthington-Simpson fuel transfer pumps which, it may be seen, can also be used for pumping the Diesel oil for the three 400-kW. Diesel generators. Each pump is capable of an hourly delivery of 42 tons of oil having a viscosity equal to that of water against an 80-ft. head, or 25 tons of 3,500-seconds fuel against a total head of 80ft. From the heavy oil service, or settling tanks, the oil is gravity fed to the purifying equipment situated in a special compartment on a flat in the engine room. In this are two De Laval purifiers and two clarifiers of the same make, also one standby unit suitable for either duty. Three steam heaters are provided. The fuel, after passing through a purifier and clarifier, is then delivered to the two 13-ton purified heavy oil tanks, whence it is fed to the fuel pumps on each of the main engines. A spill valve is fitted to each main engine fuel pump bottle, so that the fuel can be circulated through the pumps when the engines are at a standstill. Furthermore, each fuel valve has a circulating valve enabling the high-pressure fuel system to be circulated when the engines are not running. Drains from these circulating valves are led to an observation tank and then to each fuel pump tray. It will be seen that immediately before each main engine fuel pump there is a steam-operated fuel heater, having a heating surface of 20 sq. ft.—sufficient to maintain the heavy fuel at a temperature of 180 deg. F. A thermostatically-operated steam control valve is fitted to enable the fuel to be maintained at the requisite temperature. Diesel fuel is also led to the main engine fuel pumps, a changeover cock being fitted on each so that the engines may readily be switched to the alternate fuel. Steam tracer pipes are laid alongside and within the insulation surrounding the main engine high-pressure fuel piping between the fuel pumps and the valve blocks on the camshaft platforms. Each suction and discharge line of the two double-plunger, high-pressure, fuel-priming pumps is also lagged and steam-heated. The double-bottom, settling and daily service tanks for heavy oil are, of course, fitted with steam heating coils, the bottoms of the settling and service tanks being especially arranged to facilitate the removal of sludge.—*The Motor Ship, Vol. 33, February 1953; p. 483.*

Wear Resistant Aluminium Finishes

Procedures have been developed recently which produce anodic coatings on aluminium of greater thickness and increased density. Because of their substantial thickness, amounting to 0·001 to 0·005 inch, and denser structure, these coatings support greater loads and overcome the shortcomings of thinner anodic coatings with regard to scratching and indentation-type abrasion. These hard, thick coatings are produced in acid-type electrolytes which are operated at low temperatures and high



Diagrammatic arrangement of the heavy and Diesel oil systems in the City of Port Elizabeth current densities. Operating techniques are particularly important insofar as consideration must be given to positive contact during the complete anodic oxidation period. Exceptional agitation and circulation of the electrolyte must be maintained in order to decrease the possibility of burning in areas of high current density. Hard coatings for aluminium alloys are naturally of considerable interest to the aircraft industry, where a light metal with a hard wear-resistant finish offers a valuable replacement for parts produced from heavier metals of higher inherent hardness. Aluminium alloys with hard oxide coatings can be used for gears, pinions, bearing races, slides, various pistons, helicopter blade edges, and numerous other applications. These coatings offer also promising possibilities in other fields, as for instance in applications where aluminium alloys are exposed to the erosive influences of hot gases or liquids. Thus, they can be applied to advantage to high velocity pumps and blower impellers to minimize erosion. Hard chromium plating based on the zinc immersion process can be used to increase the surface hardness and wear resistance of aluminium alloys. Many applications involving wear or rubbing require the deposition of softer metals than chromium and, for such purposes, tin coatings are being used, which are applied by simple chemical immersion processes.—*The Engineers' Digest, Vol. 14, January 1953; p. 3.*

Pump for Semi-solids

In many industries the problem arises of pumping materials of a semi-solid consistency, such as clays, pulps, muds, slurries, silt, cement, washery fines and industrial wastes. Materials of this character are amenable to pumping once they have entered the pump, but to induce them to enter is a most difficult problem. The consistency we are considering is too thick to admit of the material being pumped by centrifugal means or of passing through the valves of a normal reciprocating pump. In the Meader pump, for which the sole manufacturing rights for the United Kingdom have been acquired by Henry Balfour and Co., Ltd., Leven, Fife, the problem has been neatly resolved by placing the pump cylinder round the material instead of introducing the material into the cylinder. Figs. 1-3 show diagrammatically the construction of the pump and the three stages in the operating cycle. The material to be handled is dropped into a hopper *a* at one end of which is a discharge pipe *b* leading to the delivery pipe, and at the other end is a hydraulic cylinder *c* containing a cutting cylinder *d* and a piston *e*. When the hopper has been charged, water under pressure is introduced at *f*, driving the cutting cylinder *d* through the material until it seals itself against a seating *g*, as shown in Fig. 2. By this action the cutting cylinder has enclosed a core of material, and in the next stage water is introduced through *h*, driving piston *e* across to the position shown in Fig. 3 and forcing the core of material through the discharge piece *b* into the delivery pipe. Finally, water is introduced through *i* to return the assembly to the position shown in Fig. 1, upon which the material in the hopper slides down into the vacant space—a movement facilitated by the shape of the hopper as shown in Fig. 4. Hydraulic power is provided by a multi-stage centrifugal pump driven by electric motor or any other convenient means. Alternatively the pump can be designed to operate with steam or compressed air. An automatic valve gear is normally supplied to control the sequence of operations, and in such cases an extension rod, *j* in Fig. 5, is provided to operate the valve gear in accordance with the movement of piston *e*. With the pump as described, the pumping stroke occupies one-third of the operating time; if, therefore, a continuous delivery is required, a triple-cylinder unit can be supplied. The Meader pump should greatly extend the scope of pumping semi-solid materials, for it is claimed that any material capable of being extruded at normal temperatures can be handled. It is claimed that maintenance should be light because the pump operates comparatively slowly and there are no valves in contact with the material being pumped. Unless a self-setting material such as cement is being handled, the delivery line can be left charged when the pump is shut down. The range of pump sizes now under consideration is from 5 to 100 tons per hour, but units of larger capacities could be designed, if required.—*Colliery Engineering, Vol. 28, No. 334; pp. 520-521.*

Initiation of Brittle Fracture

Experimental techniques have been evolved to initiate brittle fracture in structural steels from a sharp cleavage crack which synthesizes the fabrication cracks developed in welded structures. The cleavage crack is developed by the cracking of a small bead-on-plate of brittle hard-surfacing weld metal. Loading of the test plates containing such a crack-starter weld is accomplished by the use of explosives or a drop weight. Tests conducted over a range of service temperatures have demonstrated that structural steels of the ship plate type develop brittle fracture in the temperature range of ship fractures. Detailed correlation with ship fracture data indicated that the conditions of crack initiation, propagation and stoppage are reproduced by the tests. The significance of Charpy V and Keyhole transition criteria for the prediction of conditions of brittle fracture were investigated for various types of structural steels. It is

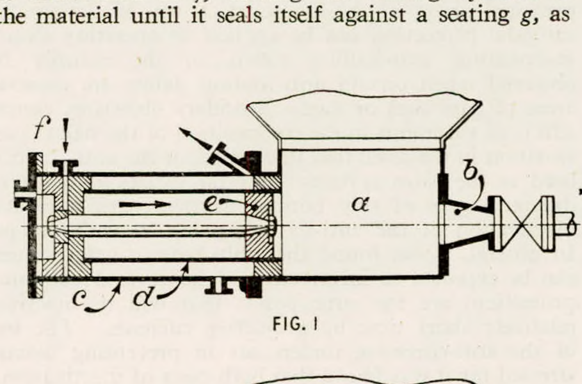


FIG. I

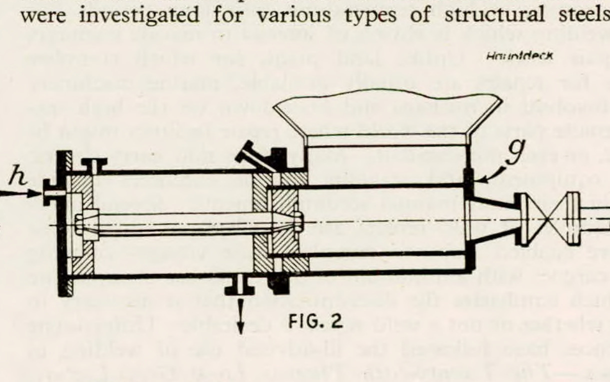


FIG. 2

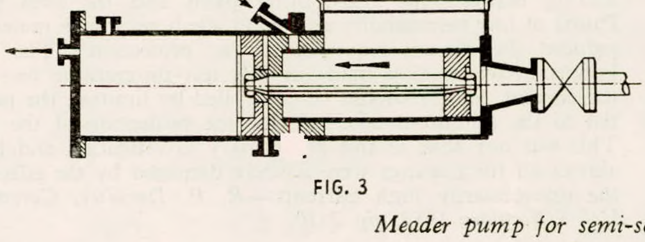


FIG. 3

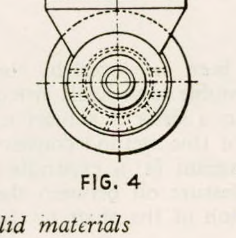


FIG. 4

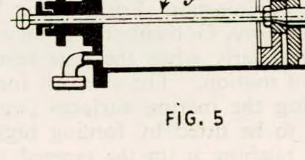


FIG. 5

Meader pump for semi-solid materials

demonstrated that only criteria associated with nearly completely brittle fractures of these specimens are significant to the problem.—*P. P. Puzak, E. W. Eschbacher and W. S. Pellini, The Welding Journal, Vol. 31, December 1952; pp. 561-s-581-s.*

Notched Steam Bend Test

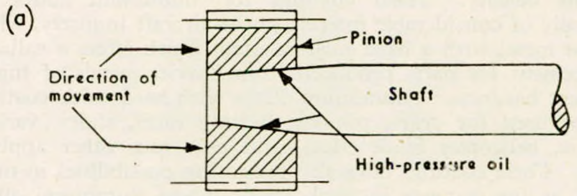
The test under consideration is the notched slow-bend test, used by the authors in their investigations of brittle fracture. A description of test performance, criteria, and phenomena encountered is given in a paper that was presented and discussed at the recent Symposium on Notch-Bar Testing. In this paper, several views could be put forward only as statements without proof, and so part of the present paper is concerned with the full experimental evidence for these statements. The other part deals with the results of new experiments, and, finally, the notched slow-bend test and the Charpy V-notch impact test are compared, with special regard to their merits as acceptance tests from a statistical point of view.—*J. E. de Graaf and J. H. van der Veen. Journal of the Iron and Steel Institute, Vol. 173, January 1953; pp. 19-30.*

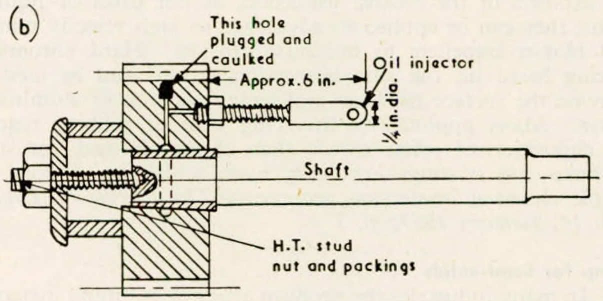
Welding in Marine Engineering

This lecture comprises a survey and commentary on the use of welding for the construction and repair of marine machinery for main propelling and ancillary purposes. The history of welding in relation to marine engineering is traced from the hammer welding of boiler seams in 1870 up to the present day, when welds made by modern processes are accepted for highly stressed components. Descriptions of welding processes are omitted since the lecture deals with welding applications rather than welding technique. Numerous examples are given and illustrated, and many of them relate to design details which are typical of good practice and are accepted by ship classification authorities. The lecture is divided under two main heads, namely, "New Construction" and "Repairs". A further subdivision is made under "New Construction", so that boilers, pipe-work, machinery components, turbines, gearing, electrical propulsion, refrigerating plant, and dredging craft are dealt with in that order. Also included in this section are some remarks of general interest on the subject of residual stresses and their relief by thermal treatment. Problems concerning the welding of alloy steels for gas turbines are discussed. There is need for research into the weldability of these steels, bearing in mind that not only the welds should be free from micro-cracks to start with, but they should also be equivalent to the parent metal in resistance to metallurgical and physical deterioration whilst operating at high temperatures over long periods. A field of welding which is always of interest to marine engineers is in repair work. Unlike land plant, for which complete resources for repairs are usually available, marine machinery may be involved in mishaps and breakdown on the high seas and in remote parts of the world where repair facilities might be primitive, or even non-existent. Many ships now carry electric welding equipment, and seagoing marine engineers include welding in their many manual accomplishments. Several interesting examples of weld repairs are described, which in some cases have enabled ships to complete their voyages carrying valuable cargoes with a minimum of delay. Other examples are given which emphasize the discrimination that is necessary in deciding whether or not a weld repair is desirable. Unfortunate consequences have followed the ill-advised use of welding in some cases.—*The Twenty-fifth Thomas Lowe Grey Lecture, read by H. N. Pemberton, 9th January 1953, before the Institution of Mechanical Engineers.*

Oil Injection Shrink Fitting

The oil injection shrink fitting has been developed by the SKF Company, Gothenburg, to avoid trouble with badly fitted keys, particularly when the part keyed to a shaft is subject to reversal of motion. The simplest form of this method consists in tapering the mating surfaces (see diagram (a)), expanding the part to be fitted by forcing high-pressure oil between the surfaces, pushing it up the tapered portion of the shaft to the correct position, and then releasing the pressure. In practice it is usually inconvenient to use a tapered shaft, and therefore a sleeve is introduced between a parallel shaft and the part to be fitted (see diagram (b)). The stud shown on the end of the





shaft, with a packing between it and the pinion that is to be fitted, is used in mounting the part accurately; also, it is essential for removal to restrain the pinion, which is liable to fly off violently when the oil pressure is applied. The advantages of the sleeve arrangement are that the part may be positioned precisely on the shaft without prejudice to the degree of interference, and that the sleeve may be used as a gauge when boring the part to be fitted. The key to success is the accurate machining of all parts to the tolerances laid down, and any shortcomings in this respect may lead to difficulties.—*J. A. Abrahams, Journal of The Iron and Steel Institute, Vol. 173, February 1953; p. 118.*

Shipbottom Paints Subjected to Cathodic Protection

This paper discusses the results of a test programme designed at a meeting of interested parties held at the Sea Horse Institute, Harbour Island, N.C. on 8th June 1950. The principal object of the investigation was to determine whether cathodic protection can be applied to operating ships without inactivating anti-fouling paints in the manner frequently observed when certain anti-fouling paints are associated with areas of bare steel or zinc. Scondary objectives concerned the effects of variations in the composition of the paints used and in variation in the dried film thicknesses of the anti-corrosive paints used in the paint systems. Several variations of two basically different types of ship bottom paint systems were studied for inactivation of the anti-fouling paint by cathodic protection. In general, it was found that ship bottom paint systems which can be expected to inactivate, and hence foul without cathodic protection, are the same paints that will be inactivated in a relatively short time by protective currents. The importance of the anti-corrosive undercoats in preventing inactivation is stressed for it was found that both cases of inactivation observed in this test were substantially eliminated by a better "barrier" coating between the anti-fouling paint and the steel plate. Paints of low permeability and good alkali resistance materially reduced the current requirements for protection. The most important observation made in this test programme was that the current density should be controlled by limiting the potential to the minimum necessary for the protection of the steel. This was not done in this preliminary investigation and hence almost all the coatings were severely damaged by the effects of the unnecessarily high currents.—*R. P. Devoluy, Corrosion, Vol. 9, January 1953; pp. 2-10.*

ENGINEERING ABSTRACTS

Section 3. SHIPBUILDING AND MARINE ENGINEERING

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Failures and Defects in Welded Ships

The recent failures of the T2 Tankers, *Pendleton* and *Fort Mercer*, have again focused attention on the necessity of eliminating these failures in service. Much has been done and is being done both to find the underlying causes of these failures and to prevent their occurrence. Structural failures are not, however, confined to ships in service. Unfortunately, they also occur during construction. Although these construction failures do not represent the potential hazard in the loss of life or property that service failures do, they should not and cannot be lightly dismissed. Every construction failure or defect of any kind, if undetected or unrepaired, may become the start of a service failure. The basic causes for these structural failures during construction may be classed in the same three groups as were the causes of service failures, namely, workmanship, material and design. A failure due to faulty workmanship is shown in Fig. 5. The crack in the vertical keel butt shown

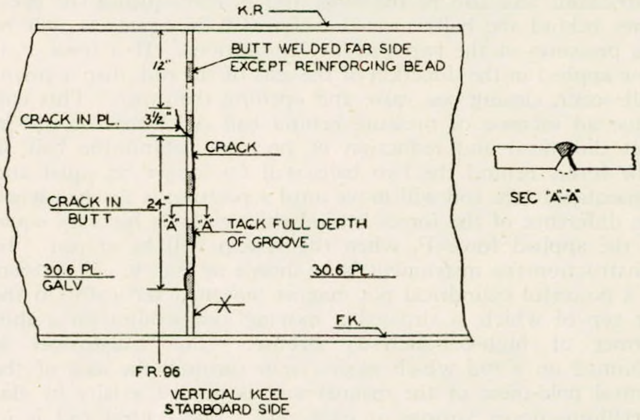


FIG. 5—*Cracked butt weld*

in this figure was caused by leaving overnight the partially completed butt, welded on one side and only partially block welded on the second side, with a sudden drop in the temperature, in addition to having an excessive restraint caused by improperly placed, oversized tacks to adjacent structure. After chipping out the crack in the plate and in the weld and removing the tacks, the cracks in the plate were first welded and then the butt rewelded in the normal manner. Failure due to incorrect design is illustrated in Fig. 32. This is the

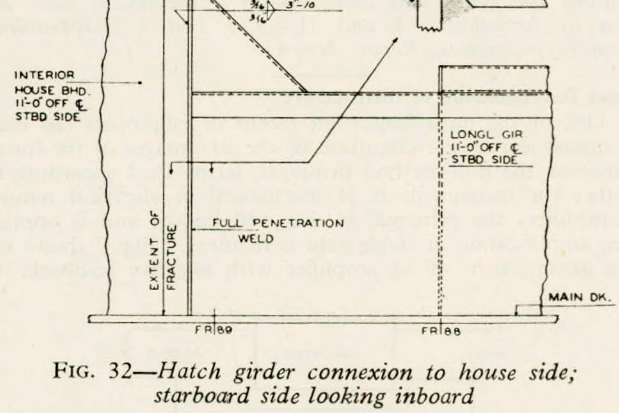


FIG. 32—*Hatch girder connexion to house side; starboard side looking inboard*

design of a hatch girder in way of the house front in a ship presently under construction. The hatch girder with full penetration welds is backed up inside the house by a light bulkhead with a continuous weld one side and an intermittent weld on the other. These fillet welds cracked following flame straightening of this interior bulkhead. Similar designs to this on previous ships have cracked in the same locations both in launching and in service. Good design must provide for a continuity of strength or a gradual tapering off of section rather than an abrupt discontinuity as shown here. Although these welds could be chipped out and repaired, a better repair would be to add a heavier insert in the interior bulkhead to allow for carrying the strength through the bulkhead. Failures from design are every bit as much trouble and expense as those from bad workmanship. Every improvement and simplified detail in design that can help eliminate a failure during construction is worth while.—*R. D. Bradway, The Welding Journal, Vol. 31, December 1952; pp. 1111-1121.*

Modern Monotower Crane

Crane design generally has kept pace with the latest engineering developments to meet the problems and demands of modern mechanical handling. The use of light alloys, the welding of steel and the latest developments in electrical control and hydraulics have all been incorporated in the most recent productions of the crane industry. It is the purpose of this paper to describe generally and show the method of approach to a few of the problems raised in the design and manufacture of a monotower crane, which is now being used in shipbuilding. This type of crane consists of a vertical mast, to which is hinged on the front a luffing jib and on the rear a fixed type of ballast box. The machinery house and operator's cabin are also fitted to the mast above the tower top. The above parts all revolve about the crane centre, the lower section of the mast being located within the crane tower. The structure of the crane is fabricated from rolled steel sections, with the exception of the jib, which is of aluminium alloy construction. The main structural members are all riveted, welding being used for a number of secondary steel members and the bogies and compensating girders of the travel gear. The crane lifts a maximum load of 35 tons at a radius of 110ft. from its centre.—*Paper by J. Patrick, read at a meeting of the Institution of Engineers and Shipbuilders in Scotland, 10th February 1953.*

Test of 40-ton Derrick

Measurements of mast deflexion and stresses were made at the works of William Gray and Co., Ltd., West Hartlepool, during a test carried out on a 40-ton derrick with unstayed mast. The experimental results are given in detail, together with particulars of mast and derrick arrangements. A comparison of these results has been made with the predicted values and quite good agreement obtained in the case of mast stresses, although some discrepancies are apparent in the comparison of mast deflexions, where in one case the theoretical value was about six times that recorded in the experiment. Typical calculations for stress and deflexion on a theoretical basis are given in Appendices I and II.—*The British Shipbuilding Research Association, Report No. 81.*

Recent Developments in Instruments

One of the most important recent developments has been the almost universal recognition of the advantages of the force-balance or the null-method principle, terms used according to whether the instrument is of mechanical or electrical nature. In amplifiers the principle is also well known and is applied when amplification of stable gain is required. Fig. 7 shows the basic arrangement of an amplifier with negative feedback, an

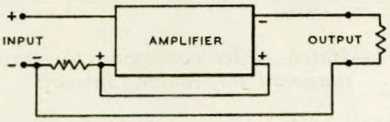


FIG. 7—*Feedback D.C. amplifier*

arrangement in which the output signal is compared with the input signal and any deviation from the desired ratio changes the working level to maintain the desired conditions. In precision laboratory measurements the use of null methods as such has long since been accepted, examples being potentiometric and bridge methods of measurement. Calibration errors and non-linearity of deflexion instruments have no influence on the accuracy of the required measurement as in both cases the indicating instrument is adjusted to give zero deflexion before measurements are taken. The human operator completes the closed sequence control system of the apparatus; he notices any deviation of the detecting device and manually restores the deviation to zero. The accuracy with which this can be done depends on the sensitivity of the galvanometer, and for perfect accuracy to be obtained the galvanometer would have to possess infinite sensitivity. The same applies to any automatic system and consequently, as infinite gain is not attainable, a finite deviation must occur to initiate the correcting action. The extent of the deviation needed for a given corrective action will depend on the gain of the system. If the amplifier gain is large enough, the deviation can be of negligible value and the output signal will be substantially independent of quite large variations which may occur in the value of the gain. The application of the force-balance principle is illustrated by reference to an electropneumatic converter. The purpose of this instrument is to convert an electrical direct-current signal into a proportional pneumatic pressure. Such an instrument serves an important purpose when it is desired to apply electrical measurements to pneumatic controllers which require pneumatic pressure inputs proportional to the measurement. Fig. 8 shows the operating principle in the simplest form. The central rod member,

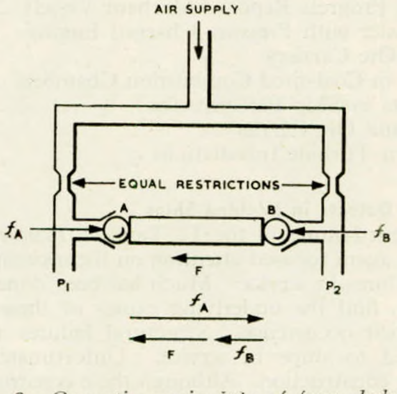


FIG. 8—*Operating principle of force-balance electropneumatic converter*

although guided axially, is not elastically restrained. At each end of the rod a ball-and-cone valve is arranged, each connected through similar restrictors to a common source of pneumatic pressure. Should the adjustments of the two valves be identical, i.e. with the rod in the mid-position, the air leak will be the same from each valve. The pressure-drops across the two restrictions will also be the same and in consequence the pressures behind the balls in each valve will be equal, as will be the pressures at the two pressure connexions. If a force F is now applied in the direction of the axis of the rod, displacement will occur, closing one valve and opening the other. This will cause an increase of pressure behind ball A (if the force is in that direction) and reduction of pressure behind the ball B. The forces behind the two balls will no longer be equal and consequently the rod will move until a position is reached when the difference of the forces behind the two balls becomes equal to the applied force F, when the system will be at rest. In construction the instrument is as shown in Fig. 9. It consists of a powerful cylindrical pot magnet mounted vertically, in the air gap of which is situated a moving coil wound on a thin former of high-conductivity bronze. The coil-former is mounted on a rod which passes right through the axis of the central pole-piece of the magnet and is guided axially by flat beryllium-copper springs at each end. The central rod is in two parts, insulated from each other, so that the disturbing

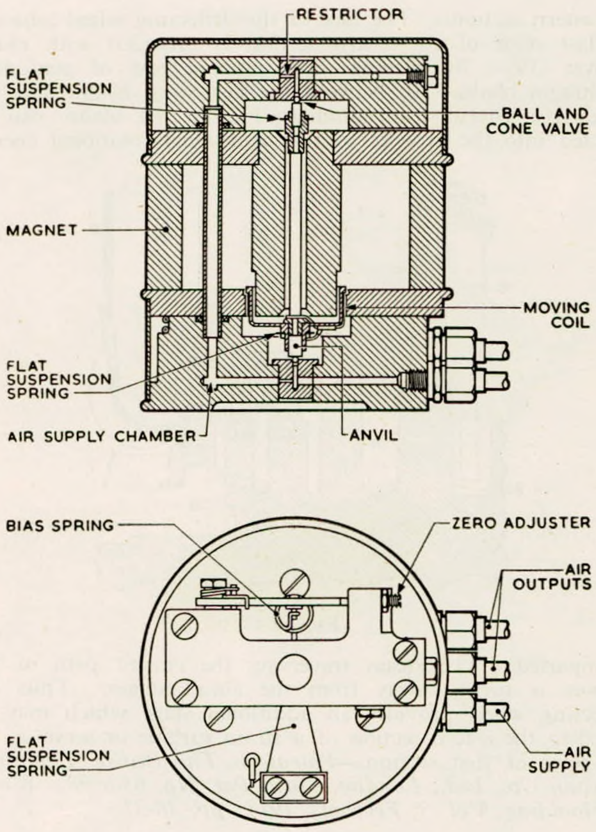


FIG. 9—*Constructional details of electropneumatic converter*

effect of ligaments may be avoided by using the springs to carry current to the moving coil. The ends of the rod carry glass anvils against which bear the $\frac{1}{16}$ inch diameter balls of two ball-valves, the thrusts of the valves acting in opposition.—*E. C. Klepp, Journal of the Institute of Fuel, Vol. 25, January 1953; pp. 295-303.*

Steam Condensate Corrosion Inhibitors

A laboratory method for studying corrosion in steam condensate systems is described. In brief, a synthetic condensate is produced in a glass tower by aerating heated distilled water with a mixture of carbon dioxide and air. This condensate and a solution of the treatment are proportioned into a test container by gravity feed. A number of steel test coupons are suspended in the latter and the liquid is mildly agitated with a paddle stirrer. At periodic intervals a specimen is removed from the bath and the weight loss determined. The temperature, free carbon dioxide and dissolved oxygen of the synthetic condensate, and the treatment concentration are controlled throughout the test. Corrosion test data are presented for synthetic condensate systems in which sodium hydroxide, sodium polyphosphate and neutralizing and film forming amines are used. The effect of precorrosion of the test specimen and of contamination of the condensate with boiler water on the functioning of several types of inhibitors are described. The merits and limitations of the test method and possible interpretations of certain of the results are discussed.—*H. Patzelt, Corrosion, Vol. 9, January 1953; pp. 19-24.*

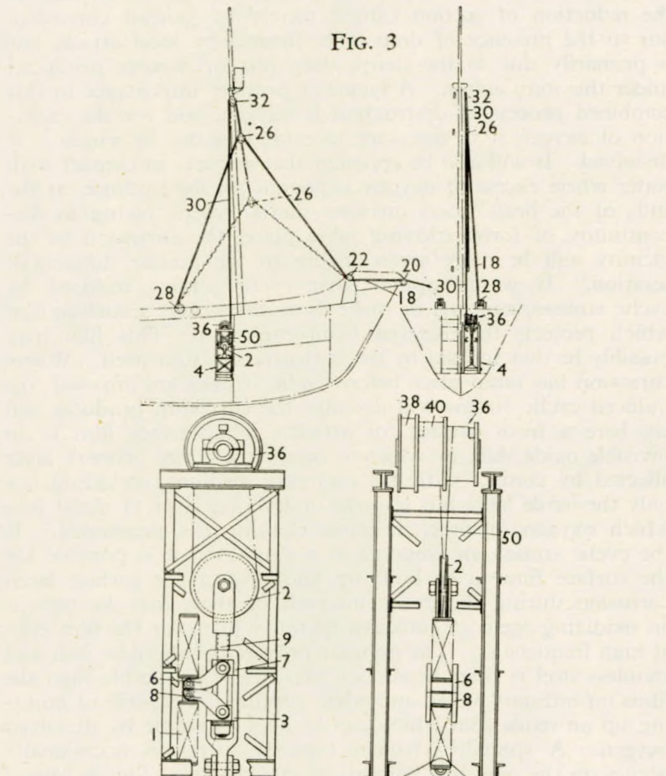
Some Observations on Corrosion in Engineering

It is the purpose of this paper to present a brief survey of the subject and focus attention on the more important corrosion problems which have come within the author's day-to-day experience. While this experience naturally has a marine bias, many parallel cases will be found in other branches of engineering. It is intended to deal briefly with the theoretical aspect of the subject, and to stress the practical side by referring to a number of cases of corrosion and the circumstances under which they have occurred. The most serious defects encountered with propeller shafts are associated with corrosion-fatigue. The methods adopted to exclude seawater from access to the shaft are not always completely successful, and once the attack is commenced and a corrosion groove formed in the periphery of the shaft, a condition for failure is established. The effect on the strength of metals through the combination of corrosion and cyclic stress has been shown by a number of investigators to produce much greater accelerated damage than would be the case with stressless corrosion, corrosion with static stress, or failure under cyclic stress applied in a corrosion-free environment. The corrosion-fatigue effect is not due to the reduction of section caused merely by general corrosion, nor to the presence of deep holes formed by local attack, but is primarily due to the sharp, deep pits or fissures produced under this joint action. A factor of primary importance in this combined process of destruction is oxygen, and for the exclusion of oxygen it is necessary to exclude water in which it is dissolved. It will also be apparent that at parts in contact with water where excess of oxygen is present as, for instance, at the ends of the brass liners on screwshafts, where, owing to discontinuity of form, eddying takes place, the corrosion in the vicinity will be more severe owing to the greater differential aeration. It would appear that cyclic strains induced by cyclic stresses, rupture, or make more permeable, a surface film which protects the material from corrosion. This film may possibly be that formed by the corrosive medium itself. Where corrosion has taken place before cyclic stresses are imposed, the induced cyclic strains will dislodge the corrosion products and lay bare a fresh surface for attack. The surface film is an invisible oxide skin, by which is meant the entire primary layer affected by contact with the corrosive medium, including not only the oxide layer but also the underlying layer of metal into which oxygen or other negative element has penetrated. If the cyclic strains are imposed at a slow rate, it is possible for the surface film to be built up and protect the surface from corrosion during the cyclic interval, or there may be present an oxidizing agent in sufficient quantity to repair the film even at high frequencies. The peculiar property of stainless iron and stainless steel is that the surface film is less permeable than the films on ordinary steels, and when ruptured is capable of building up an oxide film which can be kept in repair by dissolved oxygen. A specially virulent type of corrosion occasionally occurs on the pins and journals of crankshafts. This is caused by contaminated lubricating oil. It is found that the oil deteriorates after some length of service, due to atmospheric pollution, seawater leakage, and the leakage of combustion products from the cylinders of oil engines. The corrosive deterioration of the oil is cumulative, and depends on a number of factors, such as length of service, fuel with which it has been associated, and the moisture content of fuel, air, etc., also the presence of catalysts, such as copper components. The engineer can, however, do quite a lot to prevent deterioration of lubricating oil by providing thorough filtration, by separating piston lubrication from crankshaft lubrication, and by paying proper attention to methods of storing and replenishing lubricating and fuel oils. Unfortunately, the formation of acid in the cylinders of oil engines cannot be entirely prevented, and leakage into crankcases through worn piston-rings and cylinder-liners will occur to an extent, depending upon the condition of these parts. New oils on test are usually non-corrosive, but used oils taken from lubricating systems in motor ships are invariably found to contain water with acid reactions. Corrosion tests on used oils have shown that the addition of water causes the corrosive action of oil to be slightly more severe than when no water is present, that caustic treatment of a non-corrosive oil causes it to become corrosive, and that the corrosion effects of small amounts of seawater in the oil are of different character from those produced by oil contaminated by the products of combustion. Corrosion of crankshafts by contaminated oil may be less serious and in practice certainly less frequent than the corrosion of screwshafts by seawater, but in both cases the final effect is to reduce the fatigue endurance of the shaft.—*S. F. Dorey, C.B.E., Journal of The Institute of Petroleum, Vol. 38, November 1952; pp. 885-918.*

Absorbing Shock of Whaler Lines

A shock-absorbing device for whaler lines is illustrated in Fig. 3. The harpoon is secured to the line (18) and is shot from a gun in the usual manner. The line passes over and under pulleys (20, 22) and upwards over a pulley (26), and thence down to a winch (28). The pulley (26) is secured to a cable (30), which passes over a pulley (32) on the mast and down to the differential drum of a winch (36). The cable (30)

FIG. 3



is wound round the drum (38), to which it is secured. Another cable (50) is fixed to the drum (40) on which it is wound and passes over a pulley (2) attached to a shock absorber (4). This comprises two levers (6, 8), the latter being pivoted to a sliding piece (3) and the former similarly pivoted to a frame (7) through which the rod (9) slides. When a load is applied, the levers compress the springs (1).—*Brit. Pat. No. 685,204. Monarch Controller Co., Ltd., and R. Stephens, London. The Motor Ship, March 1953; p. 532.*

Steam Turbine Deflecting Wheel

It is customary to interpose between the ahead and astern sections of marine turbines a deflecting wheel which serves no purpose other than to deflect the steam traversing the ahead or astern flow paths to the exhaust hood. The object of this invention is to provide a steam turbine containing an astern section with a deflecting wheel which may be utilized to extract energy from the steam, in a manner similar to a radial turbine. In Fig. 9 a rotor (7) has an ahead section (8) and an astern section (9). The last stage rows (10, 11) of the two sections face each other, and are arranged to discharge steam to an exhaust hood area (17). In order to prevent steam discharged from either section from impinging on the discharge side of the other section and thus causing overheating and loss of power, a deflecting wheel (18) is interposed between the ahead and astern sections. The face of the deflecting wheel adjacent the last stage of the astern section is provided with radial grooves (19). By placing an additional row of stationary diaphragm blades (20) between the last stage blades and the wheel, the astern steam discharged from the blades can be directed into the grooves so that additional rotational energy

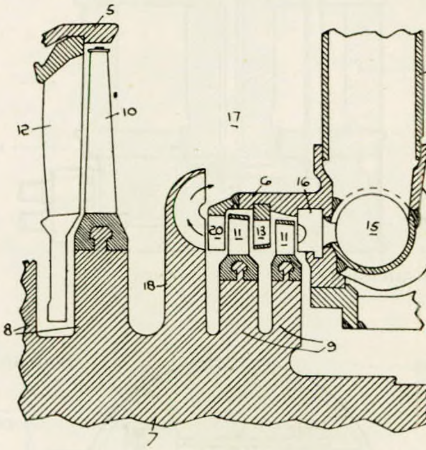


FIG. 9.

is imparted. The steam traversing the curved path of the grooves is turned away from the ahead stages. Thus the deflecting wheel provides an additional stage which may be added to the astern section of a steam turbine or serve as the last stage of that section.—*Patentees: The British Thomson-Houston Co., Ltd., London. Brit. Pat. No. 680,598. World Shipbuilding, Vol. 3, February 1953; pp. 36-37.*

Single Screw Kort Nozzle for Tug

Ordered from the Fairmile Construction Co., Ltd., of Cobham, Surrey, by the Aden Port Trust, the single-screw Kort-nozzle motor tug *Maamal* has recently completed successful trials. Specially designed for general harbour services, the *Maamal* has been built to comply with the requirements of Lloyd's Register of Shipping, for their class A.1, for towing purposes. A feature of this motor tug is that a Kort nozzle is fitted round the 63-inch propeller, the stern of the craft having been specially designed to obtain the best possible results with this device. As the *Maamal* must be able to withstand sudden crosspulls, much care has been devoted to the question of stability, and the design has resulted in a GM in service condition of 1·9 feet. In order to provide the maximum draught aft for the successful working of the propeller the keel is raked. Subdivision is effected by means of three transverse steel watertight bulkheads, which extend to the upper deck, forming four separate compartments, viz., fore peak, cabin, engine-room and store, and after peak. Entrance to the fore cabin is gained by an enclosed hatchway. The engine casing is of modern construction. Hinged flap skylights and sidelights, arranged in the casing sides, provide the natural lighting and ventilation of the engine-room, to which entry is gained through a hatchway on the casing top. An emergency escape hatchway is also provided. The main propelling machinery consists of a Blackstone Diesel engine, which is capable of developing 270 h.p. at 600 r.p.m. and has a continuous rating of 243 h.p. The S.L.M. oil-operated 2 : 1 reverse-reduction gear has been designed and manufactured by Modern Wheel Drive, Ltd. The engine is fitted with a hydraulic damper at the forward end, and there is a flexible pin-type coupling to isolate it from the reverse gear. This arrangement reduces the criticals to a minimum.—*The Shipbuilder and Marine Engine-Builder, Vol. 60, January 1953; pp. 36-39.*

Cargo Winches for Mariner Class Vessels

The Mariner class of vessel is the latest version of the high speed, dry cargo ship being built by the Maritime Administration. Gone are the days when the auxiliary power systems of ships are of small capacity and of little importance. The capacity of these systems has continued to grow so that whereas twenty years ago the average cargo ship had perhaps 200 kW. of installed generator capacity, today the average tanker or cargo vessel has from 800 kW. to 1,800 kW. and a recently completed passenger ship has over 10,000 kW. of generated capacity. The early use of electricity on ships involved direct current because direct current facilitated the obtaining of desirable speed-load characteristics on important auxiliary motors, and most particularly on the cargo winches. However, the clear superiority of the A.C. motor in regard to weight, space, cost and maintenance have brought about the almost universal adoption of A.C. power. For practically all shipboard applications, except cargo winches, satisfactory driving and control arrangements can be effected through the use of A.C. motors. For cargo winches, the D.C. motor is still required in order to obtain the proper performance, and some form of A.C. to D.C. conversion is required. Conversion equipment having special characteristics has been developed for this service and is now beginning to find extensive use. Extremely favourable overall winch characteristics, even more favourable than the older standard D.C. system equipment, can now be provided for use on ships employing an A.C. ships' service system. Such cargo winch equipment is being supplied by the General Electric Company for the C4-S-1a Mariner class of vessel. These C4's are the first large cargo ships equipped with an A.C. auxiliary power supply. Another recent trend with respect to auxiliary power system equipment is the use of packaged units. Turbine generator sets are being arranged as preassembled units complete with condenser and auxiliary equipment; cargo winch motor generator sets are arranged in self-contained units with the necessary generator and motor control equipment mounted as part of the M-G set assembly; and auxiliary under deck motor starters are now commonly arranged in multiple prefabricated assemblies known as "Group Control". Such packaged installations which are preassembled, prewired, and pre-engineered certainly offer many advantages to both the shipbuilder and the ship operator. The wiring between electrical units of a packaged assembly is done at the electrical manufacturer's factory and the whole assembly can be put in the ship and handled as a single piece of equipment.—*Paper by C. L. Eigelbach and F. A. Shean, abstracted in S.N.A.M.E. Bulletin, Vol. 8, January 1953; pp. 32-33.*

Determination of Natural Resonances in Mechanical Systems

In this paper some of the methods and conceptions used at Pametrada in the experimental exploration of turbine and gear vibrations are described and briefly discussed on the basis of examples relating to turbine and gear rotors, blading, and turbine casings. Particular emphasis is laid on the anomalous behaviour of the wave-speed curve in frequency regions in which coupled resonators have their resonant frequencies. The author presents an outline of practical experimental methods of investigation with which he has become familiar and which seem not to be generally known. All of these methods are in use at Pametrada. In selecting examples, however, the author has sometimes drawn on very early experience collected before the foundation of Pametrada; for the initial path-finding experiments have often been extended over a wider range of conditions than the later practical routine applications, and they offer, therefore, a better opportunity of explaining the general nature of phenomena even if, or rather because, they relate to structures the design of which is now obsolete.—*Paper by Dr. O. P. T. Kantorowicz, read at a Conference on Steam Turbine Research and Development held at the Institution of Mechanical Engineers, 6th March 1953.*

Back-to-back Testing of Marine Reduction Gears

In this paper is discussed the principle of testing gears by the back-to-back or power-circulation method. Results obtained from tests by this method on full-size marine gearboxes are described. The coefficient of friction for the teeth has been found to be in the region of 0·047 for hobbed pinions of 3½ per cent nickel steel meshing with forged-steel hobbed wheels, and 0·03 with a case-hardened and ground mesh. The bearing losses in a gearbox, which amounts to 65-85 per cent of the total losses, depending on the particular design, can be analysed with tolerable accuracy from simple theory. The oil flow in the bearings can also be analysed, and is found to consist of two components which can be separated from one another, one dependent upon inlet pressure and the other upon speed. The second component (the more important of the two) can be calculated. A considerable amount of work on the pitting of gear teeth is described. In a test-to-destruction of a gear, the crack which caused ultimate tooth failure started along a line of pitting near to the pitchline, and not nearer the root of the tooth where the maximum bending stress would normally be expected. The information which can be obtained from a normal shop test of a gearbox—a light run—is discussed, and finally the Pametrada torque loader, by means of which loading is maintained on gearing under test, is described.—*Paper by Dr. A. Cameron and A. D. Newman, read at a Conference on Steam Turbine Research and Development held at the Institution of Mechanical Engineers, 6th March 1953.*

Plant with Auxiliary Generators

Electric power requirements on board ship are particularly high in port. Auxiliary electric power plant is generally provided for this purpose on board ship; this, however, is uneconomical in view of the expense involved in providing such plant and the short periods for which it is used. According to this invention a marine propelling installation comprises one or more multi-crankshaft engines. Clutches are provided at one

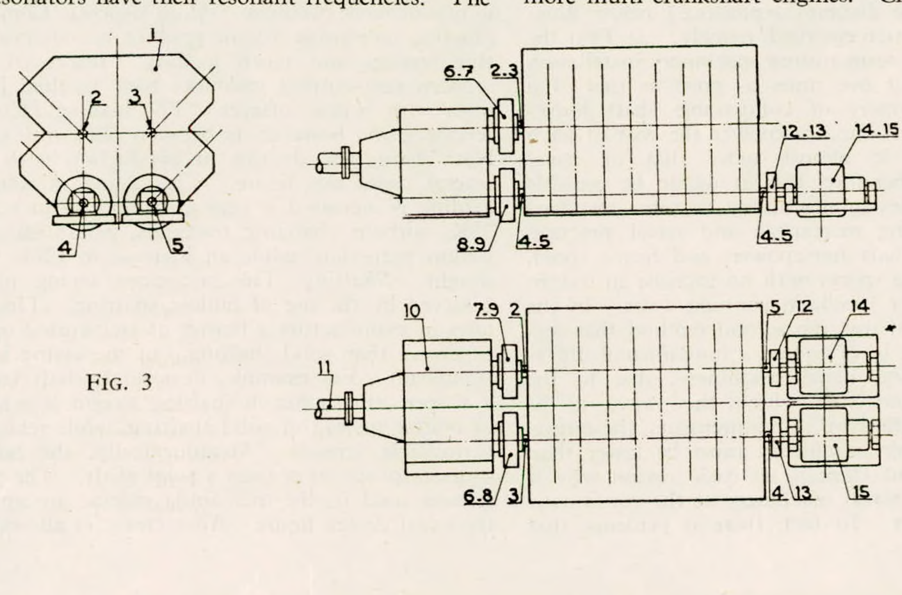


FIG. 3

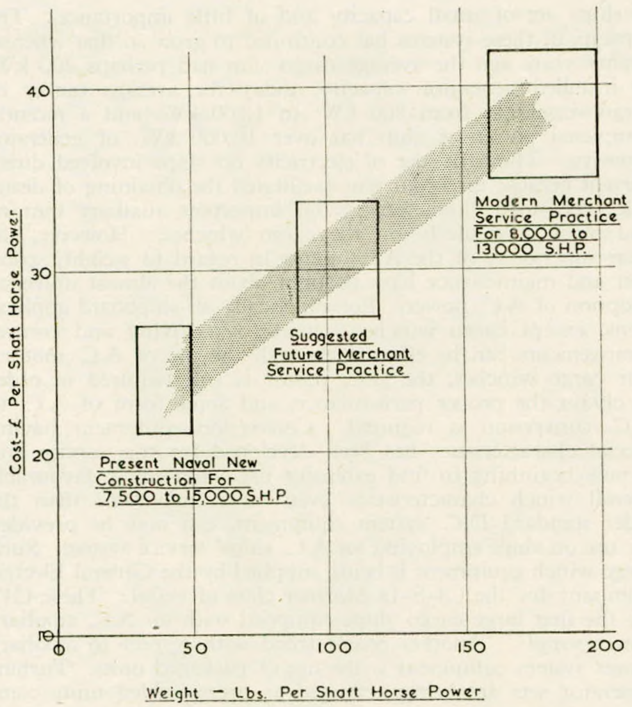
end of the crankshafts of the engine or engines for coupling them to a common load, preferably through reduction gearing. In Fig. 3 the multi-crankshaft engine (1) has four crankshafts (2, 3, 4 and 5). The upper crankshafts (2 and 3) are driven by twin banks of cylinder and piston units in V-arrangements, while the lower crankshafts (4 and 5) are driven by single banks of such units. The crankshafts are coupled at one end through clutches (6, 7, 8 and 9) to reduction gearing (10) driving a propeller shaft (11). The two lower crankshafts (4 and 5) are provided at their forward ends with clutches (12 and 13) through which they are coupled to electric generators (14 and 15). When it is required to produce electric power, one or both of the after clutches (8 and 9) can be disengaged, and the corresponding forward clutch (12 or 13) engaged, so that one or two generators can be put into operation.—*Patentees: Société d'Etudes de Machines Thermiques. British Patent No. 682,881. World Shipbuilding, Vol. 3, February 1953; p. 35.*

Thermal Distortion of Turbine Casings

It is axiomatic that, in order to obtain maximum efficiency, turbine machinery should be able to run with minimum clearance between rotating and stationary parts. An important problem in turbine design is that of making the machine able to maintain such clearances under different and transient conditions of load and temperature. This becomes particularly important in marine machinery, which is subject to rapid manœuvring, and the problem affects the design of both steam and gas turbines. It is obvious that casing distortion must be kept to a minimum. The case may even occur where a simpler turbine may prove more efficient than a more complicated design of theoretically higher performance, simply because the casing of the former is less liable to distortion and the machine is consequently able to run with smaller clearances. In this paper an attempt is made to analyse some of the ways in which the casings of steam and gas turbines distort under heating and cooling conditions. The derivation is included of a criterion for the susceptibility of materials to thermal distortion. Some practical cases of distortion observed by the author are quoted and discussed.—*Paper by B. J. Terrell, read at a Conference on Steam Turbine Research and Development held at the Institution of Mechanical Engineers, 6th March 1953.*

Lighter Steam Turbine Installations

Following a speech made in Liverpool by the Engineer-in-Chief of the Fleet, Admiral Sir Denis Maxwell, in which he put forward suggestions for lighter and cheaper steam turbine machinery, details have been made available by the Admiralty of these proposals. The suggestions arose out of detailed investigation sponsored by the Admiralty into a specific machinery installation for a large tanker. This drew attention to some fairly well-known facts, whose real significance is perhaps not so widely recognized. The diagram reproduced below illustrates the two main facts which emerged, namely: (a) That the specific weight of a modern steam turbine machinery installation in a merchant ship is about five times as great as that of a modern naval steam machinery of comparable shaft horsepower; and (b) that for the same horsepower the cost of such merchant ship machinery is almost twice that of naval machinery. It appeared, therefore, that it would be possible to close this gap, and to design machinery to some standard intermediate between existing mercantile and naval practice, at which either the same shaft horsepower, and hence speed, could be installed in existing spaces with no increase in weight or capital cost, or, speed for speed, in running costs. In the particular case referred to, it was the second method that was investigated in detail. There is, of course, a fundamental difference between mercantile and naval machinery, due to the different operating conditions for which they have to be designed. Because of the difference in requirements, the rating, or loading, of merchant ship machinery must be lower than that of naval machinery, but there is no basic reason why it should be less than that of naval machinery at the continuous cruising rating of the latter. In fact, there is evidence that



Cost/weight per shaft horsepower diagram

the specific weight of merchant ship machinery could be reduced to half its present figure, and yet remain within limits which are entirely acceptable to Lloyd's Register. The weight of nearly all the components in a machinery installation can be reduced by increasing the maximum allowable stresses or loading of these components. Weight can also be reduced by improved design, lightweight materials and perhaps by employing fabrication in place of casting. The following suggestions for reducing the weight of the main components have been put forward for consideration as being typical of the advances which could be made. It will be noted that they concern the steam-raising and other ancillary equipment rather than the turbines themselves. *Boilers:* The heat liberation or release rate per cubic foot of furnace volume is of the order of 75,000 B.Th.U. per hour for merchant ship practice, and six or more times this figure in the latest naval practice. By raising the merchant service figure to 120,000 or even 150,000 B.Th.U. per hour, a considerable saving in furnace volume and hence weight could be achieved, and it is claimed that there would be no increases in maintenance problems. *Main Gearing:* Compared with naval practice, mercantile marine practice is conservative in its allowable stresses and tooth loading. Improved techniques and modern gear-cutting machines have resulted in more accurate teeth with better surfaces. The loading factor for merchant service gears, however, is between 60K and 80K, whereas the Navy have new designs in production with loadings up to several times this figure. The saving in weight as allowable loading is increased is very great and an increase from 60K to 75K, without changing materials, would effect a 25 per cent weight reduction, while an increase to 120K would halve the weight. *Shafting:* The maximum saving in weight can be achieved by the use of hollow shafting. This, which necessitates in manufacture a boring or trepanning operation, is more expensive than solid shafting, but the saving in weight is most significant. For example, in a single shaft tanker installation, a 40 per cent saving in shafting weight is achieved by the use of hollow instead of solid shafting, while retaining the existing permissible stresses. Metallurgically, the hollow shaft is a sounder proposition than a solid shaft. The present allowable stresses used in the mercantile marine are approximately half the naval design figure. An increase in allowable stress would, of course, effect a further saving in weight and reduce cost. Failures of naval propeller shafting are very infrequent.—*The Shipping World, Vol. 128, 4th February 1953; p. 149, 158.*

Design for Flexible Fenders

Open piled jetties or wharfs are the cheapest maritime berths to construct but, being light and rigid, they must be protected from berthing impacts by fendering systems of adequate flexibility. A mass of 1,000 tons moving at a speed of ½ foot per second has an approximate kinetic energy of 50 inch-tons. If it is assumed that half the weight of a vessel is effective when it comes alongside (that is, that first the bow strikes a fender and then the stern swings round and strikes a second blow), then a vessel of 40,000 tons coming alongside at a speed of ½ foot per second will require a fender of 1,000-inch-tons capacity to receive it safely. Similarly a vessel of 1,000 tons will require 25-inch-tons. Fendering systems of this wide range are now becoming common practice so that open piled jetties can be built to cater for the largest of ships and in comparatively exposed waters, and a detailed description is given of twenty representative designs selected from work done over the past ten years. Post-war developments in the use of rubber in shear (by bonding the rubber to metal plates) and swinging clumps of concrete (essentially a gravity device) are described in some detail. All experience on all types of fendering—of large and small capacities—leads to the conclusion that longitudinal or glancing blows are as important as those at right angles to the jetty line. They are, however, most difficult to cater for and future development work will be mostly concerned with this particular aspect of the fendering problem.—*D. H. Little, Proceedings of the Institution of Civil Engineers, February 1953; No. 1, Vol. 2, Part II; pp. 42-105.*

Diesel Digestion

In considering the essentials of good combustion in a Diesel engine, one must look further than at the distribution of fuel at the right time and place and in a digestible form. The shape of the combustion chamber has a considerable influence, and it should be designed free of spaces to which the fuel particles cannot gain access; for example, the bumping clearance in an open chamber design should be kept to a minimum. Thirdly, the combustion chamber must be charged with as much pure air as possible during each cycle; and what is sometimes overlooked is that this pure air must not be allowed to escape past the piston rings. The above remarks may sound obvious, but the force of them may be shown by the fact that in about six months' experimental work on the direct-injection single cylinder unit of the Admiralty Standard Range 1 engine, the air utilization factor was improved by about 50 per cent. During this process exhaust analysis was frequently taken; whilst the optimum valve overlap period was being determined, two entirely different methods of measuring airflow were in regular use (a sharp edged orifice on a tank and an R.A.E. flowmeter respectively); thermal efficiency was plotted against maximum cylinder pressure; a variety of piston bowl shapes was used, each in conjunction with a choice of several spray patterns. When such work reaps its reward it can be most encouraging; on the other hand one sometimes meets with little encouragement in months of work for some obscure reason; the good result of a few weeks ago cannot be repeated—all very disheartening and perplexing.—*Paper by G. B. Fox, read at a general meeting of the Diesel Engine Users' Association, October 1952.*

The Measurement of Specific Steam Consumption

This paper records some of the methods employed in the measurement of the specific steam consumption during full scale tests of steam turbine installations. The layout of systems and instruments is described, and in addition mention is made of trial procedure, selection of instruments, and accuracy assessment. The shore trials had a number of objectives, one of the most important of which was to prove that the machinery met the requirements of specific steam consumption. Trials at powers corresponding to nozzle control points were carried out to assess the steam consumption over the entire power range. An accuracy of $\pm$ 0·5 per cent for this measurement was aimed at, although this was not achieved at the lower powers. All ancillary equipment, such as boilers and auxiliaries, eventually to be installed in the ship with the engines on test, has usually been available for the trials. The test layout of the machinery and pipe systems simulated as far as possible the ship's arrangements. Running conditions were controlled to simulate the conditions expected in actual service. For each trial, the shaft speed and power were adjusted to the power/speed characteristic expected in the ship, for both the deep- and light-draught conditions. No trial is regarded as satisfactorily completed until the results from three consecutive observation periods agree within some prescribed limit, say, $\pm$ 0·5 per cent. If it is assumed that error is proportional to $1/\sqrt{N}$ where N represents the number of measurements of any particular quantity, then, on average, the results of a trial based on four observation periods are twice as accurate as those of one based on one observation period. Three consecutive observation periods are therefore regarded as the absolute minimum, under steady conditions, acceptable for any trial. Observations for a trial, covering the period in which true steady-steaming conditions were obtained, are averaged. These are then corrected for gauge error from the calibration records, and also for such errors as static-water head in pressure-gauge lines. The specific steam consumption is finally deduced from these figures. From the estimates of accuracy and evidence produced, the accuracy of $\pm$ 0·5 per cent in the measurement of steam consumption appears to have been achieved, except at the lower end of the power range. Two reasons are responsible for the exception. First the measurement of exhaust pressure is outside the prescribed limits; but this has been minimized by taking the average of a number of tapping points. Secondly, the measurement of torque does not meet the requirements, particularly when relatively small-powered units are tested against the available 60,000 s.h.p. dynamometer. Obviously this error is reduced with units of increased maximum power.—*Paper by M. H. Petty, read at a Conference on Steam Turbine Research and Development held at the Institution of Mechanical Engineers, 6th March 1953.*

Heat Release in Coal-fired Combustion Chambers

The heat release from coal, burnt in a mechanically overfired fuel bed, has been investigated in a combustion chamber in which provision was made for measuring successively along the combustion chamber the heat release by the flame and the heat absorbed by the walls of the combustion chamber. Secondary air was applied radially through high-velocity jets above the fuel bed. Means were provided for measuring the rates of air flow and combustion of fuel, gas composition, and temperature conditions in the hot gases and in the walls of the chamber. From the observed conditions, relationships were derived, for a particular fuel, between combustion space required and the following factors: rate of combustion, air/fuel ratio, and aerodynamic conditions. Coefficients of heat transfer and the effective emissivities of the flame were determined from observations, at various horizons of the combustion chamber, of the temperature and composition of the hot gases, the temperature of the surface of the combustion chamber, and the heat transmission to the chamber walls. The results were correlated with the combustion rate and excess air, and appropriate equations relating the variables were derived. Application of the results to a Lancashire boiler indicates that if such an appliance were fitted with overfire jets supplying secondary air at a few inches w.g., efficient combustion could be achieved with 20 per cent excess air, and the overall efficiency of the boiler could be improved by some 8 per cent. Over a wide range of conditions of excess air and combustion rate, flame radiation accounts for 73 to 86 per cent of the total heat transfer, bed radiation for 8 to 21 per cent, and convection for 4 to 11 per cent.—*E. Hammond and R. J. Sarjant, Journal, Institute of Fuel, Vol. 25, March 1953; pp. 364-378, 404.*

Advances in Naval Boiler Design

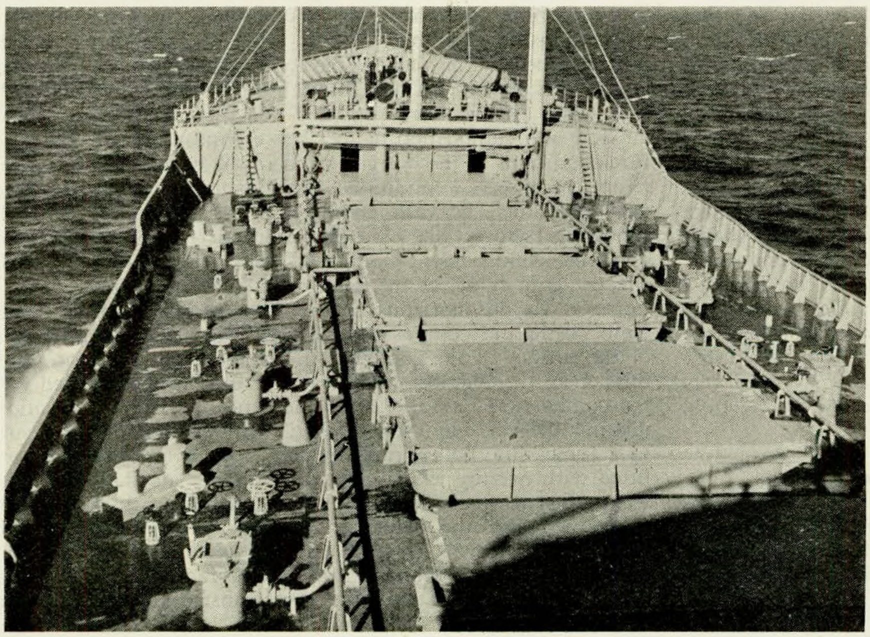
The aims toward which naval boiler design is directed may be summarized as follows: 1. Greater cruising radius. 2. Increased combat effectiveness. 3. Maintenance of reliability. Boiler design should contribute to all three. It should accomplish the first two of three goals through reductions in size and weight permitting, for the same type of vessel, increased carrying capacity for fuel and more effective combat equipment. The extent to which each of these gains are realized, of course, depends upon a compromise based on the combat mission for which the vessel is intended. Increase in energy available to the main turbines through use of higher steam temperatures and pressures permits reductions in, or maintenance of, minimum fuel consumption per shaft horsepower output, resulting in a further increase in cruising radius. Manifestly, these gains must be realized without sacrifice in reliability, since the combat effectiveness of the vessel is a direct function of the trustworthiness of its propulsion plant equipment. A major trend during the last decade has been a shift from the two-furnace superheat control boiler to the single-furnace integral superheater unit, the principal reasons for this change being simpler construction, easier operation and better adaptability to automatic control inherent in the single furnace design. Increasing the energy output through higher steaming rates and steam conditions, while maintaining the similarity in size, has, of course, required higher heat absorption in all sections of the boiler with an attendant increase in heat release of firing rates. The increase in heat release rates and concurrent decrease in boiler size necessitated development of fuel oil burners of greater capacity. Whereas past burner designs available possessed maximum capacities of about 2,500lb. of fuel per hour, present burner installations are capable of burning up to 4,000lb. of fuel per hour per burner, and one unit of 7,000lb. per hour capacity has been developed. Significant advance has accompanied development of forced or controlled circulation boilers for naval use. These units have allowed reduction in size and weight compared with natural circulation boilers of similar capacity, because heating surfaces can be more compactly disposed. For example, fireside tubes can be placed horizontally above the flames—an arrangement largely impracticable in natural circulation units. Current boilers of this type are of about the same total weight as World War II natural circulation boilers, but 40 per cent smaller in size. As compared to current natural circulation units of the same capacity and steam conditions, the weight per unit energy output of forced circulation units is about 24 per cent less, the size or space occupied per unit energy output 10 to 18 per cent less. Boilers of these general designs hold particular promise for adaptation to pressure firing and will probably be the basis of extensive future design and development.—*Paper by R. E. Stone, Abstract in S.N.A.M.E. Bulletin, Vol. 8, January 1953; p. 36.*

Largest Oil and Ore Carrier

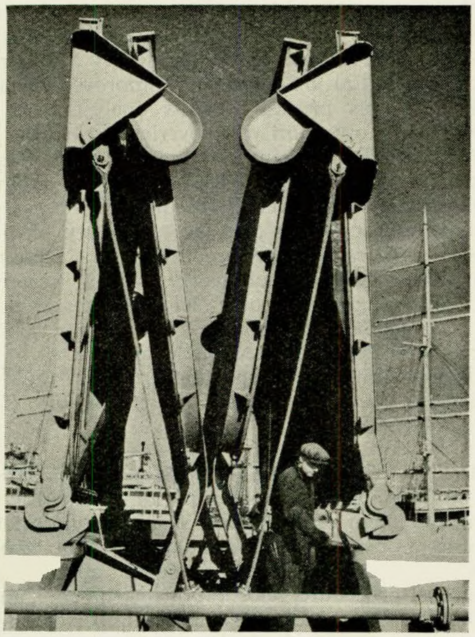
The motorship *Tarfala*, an oil and ore carrier which has been completed by Götaverken for Trafik A/B Grangesberg-Oxelosund, is the largest vessel yet built of this specialized type, having a deadweight capacity of 25,930 tons. The Grangesberg Company has specialized in this type of ship, the first of which, the *Rautas* (12,100 tons d.w.) was delivered by Götaverken in 1944. Ore is carried in hopper-sided holds, with the oil tanks extending on either side of them and beneath them. The principal particulars of the *Tarfala* are:—

Length overall	651ft. 7in.
Breadth moulded	80ft. 6in.
Depth moulded...	46ft. 6in.
Summer draught	31ft. 11in.

The hull is completely welded. Longitudinal framing is employed, with corrugated bulkheads. Two tunnels extending the length of the ship below deck provide covered access forward and aft, and are used for the electric cabling runs. There are two ore holds, each about 200ft. long, 36ft. wide and 33ft. deep. The capacity of the ore holds is about 460,000 cu. ft. and of the oil tanks about 980,000 cu. ft., allowing the ship to load her deadweight in either commodity. Oil cargo is handled by four steam pumps with a combined capacity of 1,600 tons an hour, and ballast by two electrically-driven centrifugal pumps with a combined capacity of 2,000 tons an hour. There are twelve hatches, each 24ft. 6in. by 25ft. 1in. The hatch covers are of a new type developed by Götaverken, and covered by a patent. These hatch covers can be opened and closed without any special posts or winches. They are illustrated below. Each cover is made in two halves, hinged at the centre, and is operated by means of a wire from one of the two warping winches. When the wire is taken in the cover rises in the centre, one half turning about hinges at its end while the other runs along the coaming on small wheels. When the hatch is fully open the two halves of the cover are locked in an almost vertical position at one end of the hatch. The main engine of the



The upper deck of the Tarfala, *showing the patent Götaverken hatch covers*



One of the hatch covers open

Tarfala is the largest welded Diesel engine so far designed and constructed by Götaverken. It is of the firm's two-stroke single-acting type, with nine cylinders of 760 mm. bore and 1,500 mm. stroke, and develops 8,200 b.h.p. at 112 r.p.m. to give a service speed of 14·5 knots.—*The Shipping World, Vol. 128, 25th March 1953; p. 303.*

Wide Range Burner Development

Controls furnished by three different manufacturers were installed at the laboratory in 1938 on a boiler typical of those used during World War II—a double furnace, superheat control, 600lb. per sq. in., 850 deg. F. steam generator. Much was learned from these tests, particularly that, despite rapid manœuvres, pressure and temperature could be held within close limits; but the boiler range could not be covered without considerable manual operation of the burners. Navy boilers, as opposed to those of commercial land and sea installations, operate for larger portions of their lives at loads in the order of 20 per cent full power or less; but they must be capable of increase to maximum capacity on very short notice. In 1938 no burners were available which would cover these ranges without replacing sprayer plates in the atomizers. It was apparent that naval atomizers and registers must be developed if full advantage of automatic control was to be realized. Even this, of course, necessitated improvement in the controls themselves because controls commercially available had been developed without the U.S. Navy's special requirements in mind. The American Navy embarked upon an atomizer and burner development programme which has resulted in the atomizers currently available. The author has laid out here some of the many varieties of atomizers that have been tried in the laboratory. As a result of all this work, several developments have stood out. The use of an atomizing medium such as air or steam was not considered advantageous to the Navy because (1) additional air compressors with their attendant maintenance would have been essential; (2) water supply for steam atomizers became a problem of manufacture and storage. Efforts, therefore, have been mainly concentrated on return flow mechanical type pressure atomizers. Two main factors have been involved, oil pressures of 1,000lb. per sq. in. and auxiliary air pressure for assisting burning at lower rates; with the higher oil pressure larger atomizers can be manufactured which can satisfactorily atomize quantities in the order of 4,000lb. per hr.—*Paper by M. Brunschwig, Abstract in S.N.A.M.E. Bulletin, Vol. 8, January 1953; p. 36.*

Gas Turbine Progress Report—Merchant Vessels

It would appear from a fuel-cost standpoint that the marine gas-turbine engine burning distillate fuel is not now competitive with the best steam plants burning residual fuel. It is certainly not competitive with a Diesel engine which burns residual fuel. Therefore, until it can be proved that the marine gas-turbine engine is superior to the steam or Diesel plants in such factors as maintenance, first cost, reliability, weight, and volume to an extent necessary to offset the increased fuel costs, there is little likelihood that it will widely supplant both the steam and the Diesel plants. However, if and when the gas-turbine plant can either burn residual fuel unrestricted as to turbine-inlet temperature and source, operate at appreciably higher temperatures, or incorporate a successful rotary regenerator, the entire situation may change. Regeneration probably will be used in the great majority of merchant-marine plants. The use of intercooling and/or reheat will depend upon the requirements of the particular vessel. Certainly intercooling is desirable where the plant pressure ratio necessitates the compression process to be done in two compressors. The merchant marine gas-turbine plant would now seem to be limited to special applications. Small passenger vessels which spend a majority of their time in port, such as a cross-Channel boat, might be one such application. There is perhaps another interesting special application of the marine gas-turbine plant. It is well known that most cargo ships are too slow for ideal use in wartime service. The high-speed submarine will make such ships even more vulnerable to torpedo attack in any future war. Unfortunately, because of the hull form and "propeller law", there are practical limitations on increasing a vessel's speed by installing additional horsepower. For example, it would require the power of a C2 cargo vessel to be increased by about 140 per cent to increase the ship's speed by 3 knots. On the other hand, the hull form is such that an increase of about 83 per cent in power would raise the speed of a C3 cargo vessel 3 knots. If it could be shown that a vessel with a speed of 21 knots, had a 25 per cent better chance of not being torpedoed than one of 18 knots, there might be justification for raising the horsepower of suitably designed vessels in time of war. This could be done by designing the steam plant originally so that the turbine rotor could operate at the higher speed, and the reduction gear, shafting, and bearings to absorb the extra "wartime" power. Simple compact open-cycle gas-turbine engines, stored during peace-time, could be installed quickly at the outbreak of the war. Shipowners cannot economically carry excess power around in normal peacetime operation, as it results in increased maintenance, and additional operating personnel and higher wages due to union regulations. This method would therefore seem to offer an economical and quick way of boosting the speed of suitable cargo vessels in time of war.—*W. A. Dolan and A. A. Hafer, Trans.A.S.M.E., Vol. 75, February 1953; pp. 177-184.*

Great Lakes Ore Carriers

During the early stages of the design of the ore carriers *Johnstown, Sparrows Point* and *Elton Hoyt 2nd,* the prospective owners required that, while the vessels were to be completed as oil burners, the arrangement of hull structure and machinery unit locations were to be such that future possible conversion from oil to coal firing could be accomplished in the most expeditious manner. Since the coal would occupy a greater volume than oil fuel, and its use would require the installation of hoppers, conveyors, crushers, stokers, induced-draught fans, etc., it was decided that the structural and machinery arrangements would first be developed for a coal-fired installation. For equal steaming capacity, coal-fired boilers would be materially larger than those for oil fuel, due to grate area requirements, ash pit volumes, etc. The coal bunkers would require hopper bottoms, with conveyors below discharging to coal crushers. A closed conveyor system would be required for lifting the coal from the crushers to the stoker hoppers; and coal feeder pipes from hoppers to stokers would be necessary. The space required for this equipment and for the induced-draught fans had to be considered when arranging the boilers and related equipment for the oil-fired installation. The machinery arrangement was developed with the propulsion unit forward of the boilers and the fuel-oil tanks aft. This permitted an acceptable arrangement of ballast pumps and piping at forward end of engine room. Two boilers are installed in each vessel. They are of the two-drum bent-tube design equipped with side, rear and roof waterwalls, convection superheater and extended-surface economizer. The boilers are designed to be readily converted from oil to coal. In order to provide for sufficient grate area and furnace volume to burn coal properly, the boilers were initially designed for coal burning. After the physical characteristics of the coal-fired boilers were determined, fuel-oil burners were installed. The boiler front in way of the burners is designed so that it may be removed completely and replaced with three spreader-type stokers. The combustion air enters the boilers at the rear (forward end) and travels underneath the boiler and up the double front casing to the burners. Should it be necessary to convert from oil to coal, the boiler front would be removed as mentioned above and the boiler floor replaced with dumping-type grates. The combustion air would then discharge up through the grates to the furnace. All steam generated flows through a multi-pass convection-type superheater. The main propulsion and generator turbines are supplied directly from the superheater outlets and the remaining high-pressure steam discharges through a desuperheater located below the water level in each steam drum. Each boiler is fitted with seven manually operated steam soot blowers, located in the superheater, generating, and economizer tube banks. These soot blowers are supplied with boiler pressure superheated steam.—*Marine Engineering, Vol. 58, February 1953; pp. 70-75.*

French Shipbuilding Research

The French counterpart of the British Shipbuilding Research Association, the Institut de Recherches de la Construction Navale, was founded in Paris in 1948. It has already completed a number of interesting studies, among which is a series of trial methods for testing the manœuvrability of ships. Considerable experimental work has been carried out on the subject of hull vibration, and it is claimed that the analysis of experimental results has led to the possibility of anticipating natural frequencies of a hull from the moment when the preliminary plans and specifications of the ship become known. The possession of such information at this point will naturally be of great assistance in arriving at a judicious choice of propelling machinery for the particular hull. Much work has also been carried out in the analysis of ship trials at sea and their comparison with experimental tank tests, and in the study of the distribution of stresses in the ship's structure. A large number of extensometric measurements have been made, which have resulted in interesting conclusions about the distribution of stresses in launching and under loading. Model tests have been carried out on individual joints at critical points, and it has been found possible to plot a graph for the corners of hatches in cargo vessels to enable greatly reduced concentrations of stress to be obtained without any difficulties in the use of materials.—*The Shipping World, Vol. 128, 11th March 1953; p. 246.*

Device to Aid in Minimizing Bending Stresses

The primary purpose of the device proposed by the author is to rapidly calculate the approximate maximum bending moment of a loaded vessel. It consists of eight sliding pieces moving over graduated bases each of which represents a particular compartment of the vessel. A window is cut in the middle of each slide to indicate the number of tons loaded in the compartment. The right hand side of each slide is marked with bending moment graduations. Those on top of the window are negative and those below are positive. To calculate the bending moment of a vessel the procedure is merely to move the sliding pieces to their corresponding deadweight and the bending moment contribution of each slide is read where the bending moment scale cuts the draught of the vessel. The algebraic sum of all those readings plus a constant gives the bending moment of the vessel amidships. In order to use this device intelligently, standards for maximum, optimum, and minimum hogging moment must be set up. It is contended that this device can be used to better advantage in quickly ascertaining the optimum bending moment than any tabular listing now available. For example, the effect on the bending moment of shifting fuel oil or loading ballast can be very quickly computed by moving the slides the proper distance and noting the increase or decrease of moment in the compartment from which the weight is shifted and comparing it with the increase or decrease in moment in the compartment to which the weight is shifted. The objective of shifting or loading should then be to bring the hogging moment as close as possible to the optimum standard set but in no case to exceed the maximum or be less than the minimum figures set.—*Paper by A. V. Divine, Abstract in S.N.A.M.E. Bulletin, Vol. 8, January 1953; p. 34.*

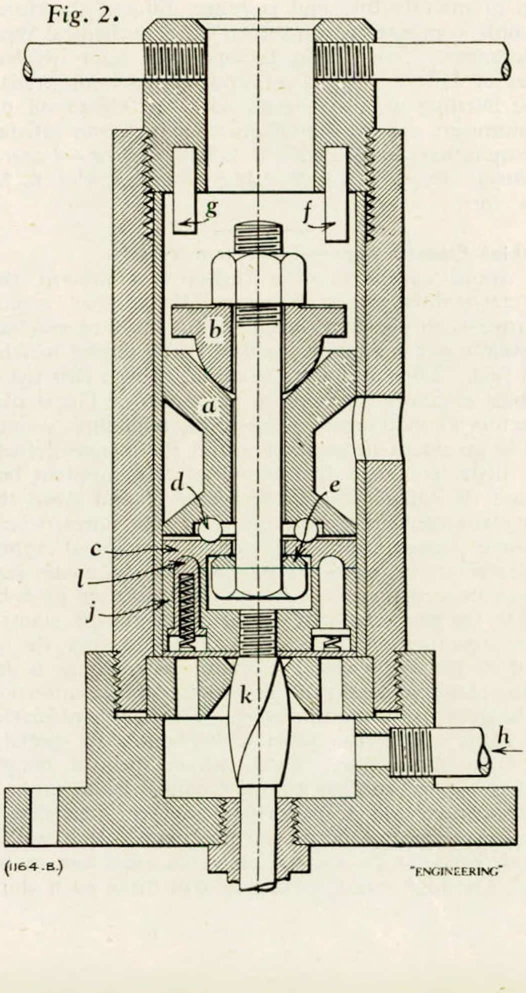
Steering of High Speed Planing Hulls

The subject of steering high speed hulls is one without the deeper background of tank tests and practical use results enjoyed by the older problem of steering displacement hulls. Access to tanks for the designer of fast hulls is rare due to lack of opportunity and/or money. To differentiate between types, high speed hulls may be considered as those with speed length ratios of five or higher. In hulls in planing condition the rudder area becomes a large percentage of the immersed fore and aft plane and in some types it in fact becomes almost 100 per cent of this steadying plane. This fact allows the rudder to control many characteristics not always considered as steering. In this situation the position of the rudder, shape, angle of post to slipstream, and area is able to cause or eliminate unstable conditions in both smooth and rough waters. The paper gives the author's brief observations on rudder effect in high speed turns, rudder positioning, the use of bow rudders, rudder area and rudder fabrication.—*Paper by T. Grenfell, Abstract in S.N.A.M.E. Bulletin, Vol. 8, January 1953; p. 35.*

Testing Nuts and Bolts

A machine for determining the comparative lives of different types of nuts and bolts when tightened up and subjected to vibration has been developed by the Lester Lock Nut and Washer Co., Ltd., Staffordshire. It applies repeated shock loads to the underside of the nut, the point of application of the load moving round the nut with each application. A conventional nut and bolt fixed in the machine and tightened in the normal way gradually work loose under test; they are then retightened and tested again. This gradual loosening, followed by retightening, may occur several times before ultimate failure takes place, though, of course, complete loosening of the nut on its seat would be regarded, in many uses of nuts and bolts, as tantamount to ultimate failure. In other uses—on railway track fishplates, for example—periodical retightening by maintenance staff is acceptable; in these cases, ultimate failure may be due to fracture of the bolt, stripping of the threads, fracture of a cotter, etc. With the Lester vibrator machine the life of a nut and bolt—up to, say, the first complete loosening—is measured in terms of the number of applied shocks. It can also be measured directly in time (minutes and seconds), provided the machine is run at the

Fig. 2.



standard rate of 712 shocks per minute. The vibrator machine, together with the piston unit which forms an inner reciprocating member, is shown in Fig. 2. The piston unit, through which the bolt is passed, consists of the piston itself, *a*, a rocker head *b*, a ring *c*, a ball thrust bearing *d*, and hardened washer *e*. The unit is free to slide vertically in the cylinder, at the top of which is a head, screwed in and provided with a striking pin *f* and a check pin *g* (slightly shorter than *f*). Thus, when the piston unit reciprocates, the rocker head is tilted by the striking pin repeatedly. At each blow the thrust between the underside of the nut and the top of the rocker head is reduced near the point of application and increased at a point diametrically opposite. The check pin *g* controls the tilt of the rocker head when the nut becomes loose. An air compressor from which the outlet valve has been removed is connected to the cylinder at *h*. Thus, when the compressor is running, there is no substantial flow of air (since the outlet valve has been removed, but there are pulsations of pressure which act on the piston, alternatively forcing it up under pressure and drawing it down by suction. Between the piston unit and the air connexion there is a secondary piston *j*, which has a twisted square shank *k* and four spring-loaded pins *l*, the latter engaging in recesses in the ring *c*. When the main piston returns to the bottom of its stroke, the secondary piston gives it a small rotational movement, the movement amounting to 20 r.p.m., i.e. at 712 shocks per minute there are about 35 shocks during each revolution of the main piston. The hardened washer *e* has a prescribed friction area, so as to eliminate variations in torque (as shown on a torque-measuring spanner) due to fortuitous variations in the bolt-head contact surface. Similarly, the rocker head *b* is machined with an inner ring face on which the nut is seated, and the ball thrust bearing also provides a connexion of constant torsional friction.—*Engineering, Vol. 175, 9th January 1953; pp. 54-55.*

Wake Studies of Plane Surfaces

This paper gives the results of experiments to measure the frictional resistance of three smooth plank surfaces, of lengths 50 feet, 18 feet and 10 feet. In the case of the 50-ft plank, four extra sets of experiments were made at points approximately 10 feet, 20 feet and 40 feet from the bow. Precautions were taken to ensure turbulent flow and the resistance was determined by the momentum drag method. The results show that the width of the friction belt did not vary with speed, and that the depth of the wake below the keel was substantially equal to the width of the wake. This width did not vary with draught. New formulas are given for the width of the friction belt and for the velocity distribution within the belt. The second part of the paper outlines a method of correcting for wake-scale effect in estimating ship powers and propeller revolutions from self-propelled model results. This method makes use of the data given in the first part of the paper.—*Paper by J. F. Allen and R. S. Cutland, read at a meeting of the North-East Coast Institution of Engineers and Shipbuilders, 27th February 1953.*

German Coaster with Pressure Charged Engine

The German ship *Fina* is the first of a series of small motor coasters ordered by Montan-Reederei, Hamburg. The vessel, constructed by G. Renck, Jr., K.G., is of 1,016 gross tons and is propelled by a M.A.K. supercharged, four-stroke, trunk-piston engine, having six cylinders with a bore of 285 mm. and a stroke of 580 mm. It develops 1,000 b.h.p. at 300 r.p.m., to give the vessel a service speed of 11½ knots, but the engine in the next vessel delivered, developing a higher power at an increased number of revolutions, will drive through a Vulcan coupling and reduction gearing. The engine is fitted with a Brown, Boveri exhaust gas turbo-blower with two outlets, each leading to three cylinders. The mean effective pressure is 7·8 kg. per sq. cm., the maximum combustion pressure 53 kg. per sq. cm., and the charging air pressure 1·34 kg. per sq. cm. A maximum of 15,500 r.p.m. can be attained by the blower. The fuel consumption is 2½ tons of gas oil per day. When at sea, the whole of the electrical load, including that for the steering gear, is met by a 20-kW. generator which is belt driven from the propeller shafting. There are also two 50-kW. 220-volt D.C. generators driven by Deutz three-cylinder, two-stroke engines scavenged by Roots-type rotary displacement blowers. One of these engines also drives a compressor. A small harbour generator is installed, this being driven by a Deutz two-cylinder, four-stroke engine developing 23 h.p. at 890 r.p.m. Special arrangements are made whereby the main and auxiliary engines may be circulated, when the ship is in cold waters, by water drawn from and returned to the engine room double bottom. The one large hold can be divided to form two. Steel sliding hatch covers of the MacGregor single-pull type are fitted. These are served by Kampnagel three-ton electric deck cranes. The ship has a length b.p. of 67 metres (219ft. 9in.), a beam of 9·8 metres (32ft. 1¾in.) and a loaded draught of 4·92 metres (16ft. 1¾in.).—*The Motor Ship, Vol. 33, February 1953; p. 470.*

Improvements in Ship Performance

After some general considerations affecting the question of ship resistance, the improvements which have been achieved in this direction are dealt with in some detail in relation to various classes of vessel and the question of hull roughness is discussed. Consideration is then given to changes in rudder and stern frame design from the resistance angle and some reference is made to associated problems. The question of screw propeller design is discussed and reference is made to special types of propeller and other means of propulsion. Brief reference is then made to vibration problems, and the question of seakindliness is touched on. Lastly, some comparison is made of the gains discussed above with those obtained from improvements in machinery and improvements in structure.—*Paper by J. F. Allan, read at a meeting of the Institute of Marine Engineers, 10th March 1953.*

Cargo Vessel with Active Rudder

The motor cargo vessel *Falkenstein*, which was recently completed at the yard of H. C. Stülcken Sohn, Hamburg, has the following principal dimensions: Length overall, 134·4 m.; length between perpendiculars, 120 m.; breadth, 16 m.; draught, 7·6 m.; 5,500 tons gross. The Diesel-electric propulsion plant consists of four M.A.K. engines, each developing 1,400 h.p. at 300 r.p.m. Each engine is coupled with a 1,150 kVA. A.E.G. alternator and the screw is driven by a synchronous motor developing 4,640 h.p. at 125 r.p.m. The active rudder will develop 400 h.p. and represents the first installation of its kind in a vessel of this type.—*Hansa, Vol. 89, 20th December 1952; p. 1789.*

Marine Applications of Aluminium

Because aluminium alloy is non-inflammable and non-absorbent, because its inorganic nature does not form a breeding ground for bacteria, and because it also possesses good thermal and sound insulating properties, aluminium provides a fine lining for insulated spaces, particularly if it is combined with a good insulating backing. The Young's Modulus and sound insulation powers of the material can also assist in reducing vibration noise. The extensive use of aluminium alloy for internal decorative work and cabin or minor bulkheads can, therefore, assist in reducing the fire risk and resonance present where wood was previously used as the main decorative medium. The application of aluminium to the upper works of large fast ships can result in reduction in beam while retaining the same transverse metacentric height. This is because of lowering of the centre of gravity and retaining the relationship between the Moment of Inertia of the water plane and the reduced displaced volume. Alternatively the beam can be reduced or a more favourable block or prismatic coefficient can be employed. Warships, whalers, trawlers, cross-Channel ships and fast liners are particularly sensitive to such treatment. The replacement of steel by aluminium alloy results in weight saving of a dual nature, that is, direct and indirect. It can be shown that upon the rate of increase of power as compared with the increase in displacement, affecting, therefore, type, weight and cost of machinery and fuel, rests the determination of the value of the possible total weight reduction. The value of using aluminium ranges from slow-speed to high-speed ships. In many cases it is now possible by employing aluminium superstructures to benefit from higher steam conditions, lighter machinery and smaller engine and boiler space. In general terms, the use of aluminium alloy results in a direct initial weight saving of approximately 50 per cent. The effect of initial saving will assume greater total weight-saving importance in the case of high-speed vessels.—*The Shipping World, Vol. 128, 4th March 1953; pp. 234-236.*

Strength of Threaded Connexions

When a screwed connexion is loaded the bolt or stud stretches and the nut contracts so that complete contact between the two sets of screwed threads is not possible. The brunt of the load is borne by the material near the base of the nut while mutual accommodation between the members is largely accounted for by bending of the threads. The load concentration factor (defined as the maximum value of thread loading expressed in terms of the mean) has been shown to depend upon pitch/diameter ratio, length/diameter ratio, size of nut, form of thread, and degree of lubrication. Fatigue fracture generally occurs near the root of the first or second thread inside the nut. The actual value of the stress concentration factor at this point will be a function of load distribution and of the stress raising effect of the particular thread form chosen. Obviously the geometry of the "notch" is a determining factor, and apart from various methods of artificially strengthening the screwed portion the most rugged threads are those with generous root radii and truncated tips. High endurance figures have been obtained from threads that approximate to the knuckle form. A blunt male thread engaging a shallow spiral groove has also been used, together with a helical bronze spring wire insert between threads. It seems to be generally agreed that the best way of improving the endurance of a screwed connexion is to form or finish the threads by cold rolling. In heavy components this cannot easily be done but there is no doubt that any ingenuity spent in rolling and burnishing the threads would be well repaid. If heat treatment is required it should be done before rolling. Alternative methods similarly based upon the production of a hard compressed surface layer are cyanide hardening, carburizing and nitriding, but in these cases the risk of failure due to grinding cracks and notch sensitivity in the hardened layer may offset other advantages and the processes require care in application. The stud or bolt itself should be as long as possible with carefully designed fillets (re-entrant if necessary) under the head of the bolt, and compound fillets at the points of transition from the waists to the full diameter. The full diameter need only be retained where the members have to be centred or located in position. Ideally also a stud should be provided with a shoulder to take a spanner for preloading the stud end and, where necessary, a damping collar. The endurance limit is sometimes notably increased by extending the screwed length well beyond that required for engagement with the nut. Another way of reducing load concentration which has found considerable application is to use a nut having a lower modulus of elasticity than the bolt. Cast iron, aluminium alloy and magnesium alloy nuts have all been successfully used in this way. The basic principles of heavy duty bolt design are:—(1) To develop the full elastic strength of the bolt throughout its length. This involves thoughtful design and choice of materials in both bolt and nut. (2) To see that the bolt is really tight and that it will remain so. This involves correct design of the members joined, and the avoidance of undue elasticity in washers or other abutments. It is better to preload the bolt to its elastic limit than to risk insufficient tightening. Subject to the need for sufficient plasticity to minimize notch effect, bolt steels should have a high elastic limit and the hardness that goes with it.—*B. R. Byrne, Journal and Transactions, The Society of Engineers, Vol. 43, October-December 1952; pp. 183-198.*

Magnetic Force Welding

In the spot welding process, quality of the weld can be greatly improved by synchronized application of electrode force, thus making it practicable to weld on a rising electrode force wave. The importance of this feature has been increasingly recognized in recent years, and it is now an accepted fact that fast follow-up improves weld quality, decreases expulsion, arcing and tip pick-up. Moreover, application of a rapidly rising electrode force makes it possible to use a lower initial electrode force which has two advantages. Firstly, it increases the heat from a given current because of higher contact resistance, and secondly, it provides concentrated heat from projections on materials such as soft aluminium or magnesium, which would collapse under the electrode force of conventional welders. The ideal source of synchronized force must obviously be the welding current itself, and in principle it is easy enough to visualize an electro-magnetic device passed by the welding current and exerting the required variable electrode pressure. Considerable difficulties were, however, encountered in the practical execution of such a scheme and much patient study and development work were required to develop a device having the operating characteristics and flexibility required in practice. In its present form the device incorporates seven possible adjustments, two of them mechanical, two magnetic and three electrical, which allows a wide variety of force-time curves to be obtained. Forces up to 3,000lb. can be produced by the present device, but there is no inherent limit to this value. Once optimum force-time characteristics have been determined for a given task, it is easy enough to establish a key for the proper setting of the machine. An incidental advantage is that the presence of the magnet system makes the machine less sensitive to the line voltage variations.—*The Engineers' Digest, Vol. 14, March 1953; pp. 74-75.*

Methods of Hard Facing

In the early days of welding when the value of hard facing with various alloys was first recognized, the oxy-acetylene welding process was used primarily. This method ordinarily produces the best quality deposits with the least amount of porosity, cracks and dilution with the base metal. Oxy-acetylene application is preferred when a thin overlay of metal is required. It is especially suitable for precision motor parts because of the close control over the molten metal. It is very good for valve jobs on which a cobalt base material is used. The deposit can be put down very smoothly. The reclaiming of Diesel engine valves, involving the application of a hard-facing alloy to the worn seat, is an excellent example of the proper use of the oxy-acetylene hard-facing method. The first step in reclaiming consists of cleaning the valves thoroughly. Then the hard-facing alloy is applied to the worn seat. A carburizing flame is used and is kept pointed away from the stem to avoid distortion. Then the valve is centred in a lathe, the outside of the valve is trued up and the seat machined to size. Finally the seat is finished in a valve-grinding machine. These reclaimed valves, measuring from 4 to 5¼in. in diameter, have a service life or three to four times that of new valves. In general, particles of scale and foreign matter are eliminated more easily by the oxy-acetylene method, and edges and corners can be formed readily. This is particularly important when grinding the part to a finished dimension is necessary. Electric arc application is advantageous for large areas over irregularly worn surfaces, or where a fine finish or precision dimensioning is not necessary. Speed of deposition is greatly accelerated over the oxy-acetylene method. It should be remembered, however, that the single layer deposit by the arc method tends to result in a deposit somewhat diluted with the base metal, softening the deposit and reducing its resistance to abrasive wear. The dilution effect can be minimized by depositing a thin first layer followed by a second layer thick enough to restore the worn part to its original dimensions. In general, the arc method is particularly suitable for hard facing all heavy parts where exceptionally thin, smooth deposits are not required.—*J. J. Barry, The Welding Journal, Vol. 32, February 1953; pp. 119-126.*

ENGINEERING ABSTRACTS

Section 3. SHIPBUILDING AND MARINE ENGINEERING

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Preheating of Mild Steel Welds

This paper describes a recent investigation concerned with the rôle of micro-crack-prevention in connexion with the benefits induced by the preheating of welds in mild steel. In recent years it has been reported that too rapid cooling following the arc welding of mild steel may produce fine-scale fissuring of the deposited metal with adverse effects on various mechanical properties. Detection of such fissures requires the microscopic examination of carefully polished sections. Once formed, the defects are not susceptible to healing by thermal treatment. It is believed that microcracking ordinarily occurs only after the cooling weld has reached a temperature range below that usually associated with the decomposition of austenite in mild steel. While the mechanism of formation is not altogether clear, it seems certain that the hydrogen content of the weld is a factor of prime importance. Welds deposited with "low-hydrogen" electrodes possess remarkable resistance to microcracking. In welds made with the more popular hydrogen-rich varieties it appears that microcracking may be prevented by measures which ensure sufficiently slow cooling in the low-temperature range. In welding practice, preheating has long been recognized as an effective method of preventing embrittlement in difficult-to-weld applications. It is most commonly employed in the welding of the more hardenable steels but can often be used to advantage in work on mild steel as well. While the effectiveness of preheating has been abundantly apparent, there has been cause to feel that the reason for its effectiveness has not been altogether clear. The usual explanations have attributed the benefits to elevated-temperature effects concerned either with the transformation of austenite in the heat-affected zone or with a general broadening of the latter region. In the light of such explanations it has been disturbing to note in welding practice that moderate preheating is surprisingly effective in the case of mild steel welds. In a California shipyard, for instance, it was reported that a disproportionate number of embrittlement problems affected welds deposited on the night shift when the temperature was somewhat lower than during the day. The problems were generally solved by employing heating torches "to take the chill off the work" before welding. It is difficult to believe that the small alterations in elevated-temperature conditions produced by mild preheating could be of such decisive benefit. On the other hand, one might expect even mild preheating to exert a strong influence on the cooling rates at lower temperatures. Since it is thought that microcracking ordinarily occurs as a result of too rapid cooling in the latter range, one is inclined to wonder if the benefits of preheating may not often be realized, in part at least, through the prevention of microcracking.—*A. E. Flanigan and T. Micleu, The Welding Journal, Vol. 32, February 1953; pp. 99s-106s.*

New Wood Laminating Process

A new laminating process that uses air pressure instead of clamps to hold the wood together forms laminated oak sections that are reported to be stronger than steel sections of the same weight. According to the American periodical "Business Week", the process is making it possible to build larger wooden ships that are three times stronger than when solid wood was used. The process, developed by Higgins Inc., New Orleans, laminates oak and fir planks under heat and pressure. The planks are screened twice before being used, and are then dressed and sized in the mill, and ends cut for the scarf or diagonal joints. A number of short planks are jointed to form one long one. The ends are placed together and the joint placed in a scarfing machine that applies heat and air pressure. The method gives an even pressure over the entire length of the planks, and provides a more uniform shrinkage of the wood. To make a large section, such as a bow stem, a number of long planks are run through a glue machine and then placed, one at a time, into a special jig. The coated planks are then locked in the jig, which gives the "bundle" the required shape. Large tarpaulins are placed over the jig and live steam applied to bring the temperature of the assembled planks to about 110 deg. F. for a duration of about ten hours. The tarpaulins are then removed, the jigs unlocked, and the solid pieces trimmed and planed to the desired tolerance. The ribs of a ship are made up of several long planks glued flat against each other, being shaped to exact curve and welded together in a special jig.—*The Shipping World, Vol. 128, 4th March 1953; p. 240.*

Norwegian V.P. Propeller

The whaler *Enern* is the first Diesel-engine whale-catcher in the Norwegian whaling fleet and the first vessel of her type to be fitted with a variable pitch propeller. The builders are A.M. Liaaen Skipsverft og Mek. Verk, Aalesund, Norway. The propeller for the *Enern* has a diameter of 3·2 metres, or 10ft. 6in., the hub diameter being one metre or about 3ft. 3⅜in.

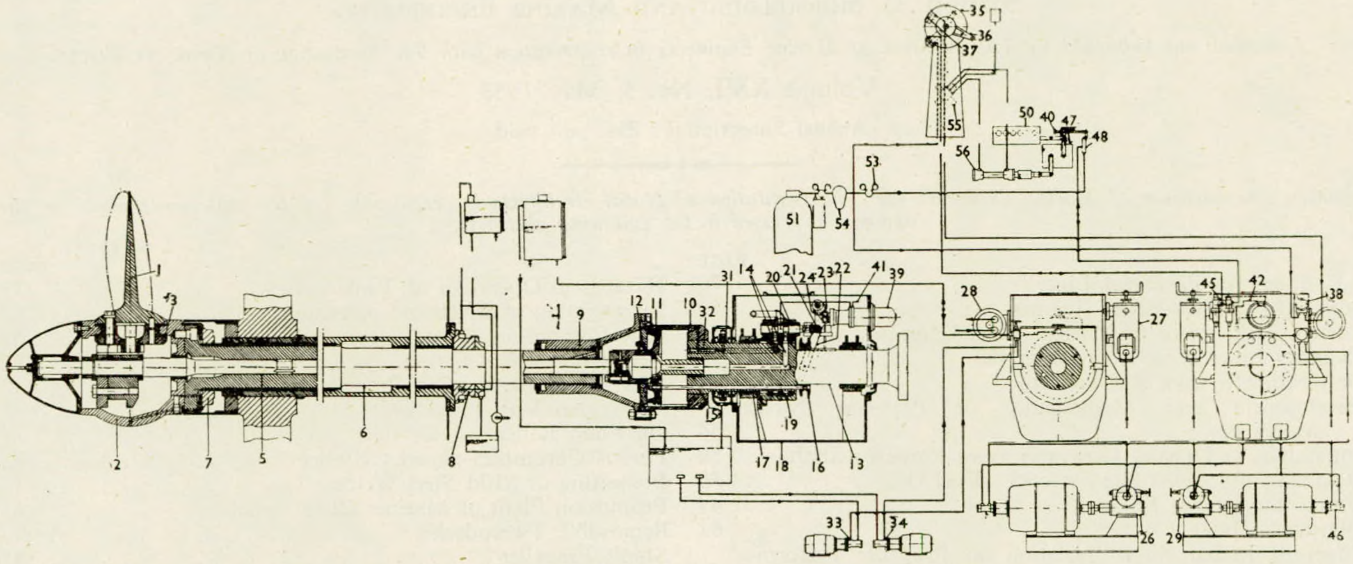


Diagram showing mechanism of Liaaen controllable pitch propeller

1.—Propeller blades. 2.—Propeller hub. 3.—Crank discs. 5.—Actuating rod. 6.—Propeller shaft. 7.—Outer stern tube stuffing box. 8.—Inner stern tube stuffing box. 9—Servomotor coupling. 10.—Servomotor. 11.—Servomotor piston. 12.—Servomotor stuffing box. 13.—Intermediate shaft. 14.—Bores for inlet and outlet of oil. 15.—Connecting rods. 16.—Sleeve. 17.—Bearings for intermediate shaft. 18.—Frame for pitch control gear. 19.—Oil distribution box. 20.—Slide valve. 21—Transverse rod. 22.—Parallel arms. 23.—Pinions. 24.—Control arms. 26.—Main oil pressure pump. 27.—Oil filter. 28.—Oil cooler. 29.—Stand-by oil pressure pump. 31.—Lubricating oil inlet ring. 32.—Bores for lubricating oil. 33.—Propeller hub lubricating oil pump. 34.—Stand-by lubricating oil pump. 35.—Control column. 36.—Control column lever. 37.—Compressed-air pitch indicator. 38.—Transmitter for pitch indicator. 39.—Auxiliary servomotor. 40.—Air reduction control valve. 41.—Remote control inlet pipe. 42.—Oil pipe from main oil pressure system to auxiliary servomotor. 45.—Air/oil pressure converter valve. 46.—Stand-by servomotor. 47.—Main engine governor. 48.—Transmitter valve. 50.—Fuel oil regulator. 51.—Starting air receiver. 53.—Reduction valves. 54.—Air filter. 55-56.—Remote control for engine governor.

The complete equipment with the hydraulic control mechanism weighs approximately 16 tons. This propeller is driven by a Nordberg-Sulzer type engine of 2,700 b.h.p. at 225 r.p.m. for a ship's speed of 17 knots. Each root is secured to a rotatable disc (3) having a crank sliding in a guide block moved by a rod (5) passing through the hollow propeller shaft. A stainless steel stuffing box (7) is provided at the outer end of the shaft and a cast-iron box (8) is fitted on the inner end of the stern tube. A coupling (9) at the forward end of the shaft is secured by the S.K.F. oil-pressure method and the flange is attached to the actuating rod (5). The piston (11) of the servomotor has a rod which passes through the cylinder cover and a stuffing box (12). There are two radial bores (14) in the intermediate shaft (13) and these holes are the inlet and outlet respectively for the servomotor oil. Three rods, one (15) of which is shown, connect the servomotor piston to a sleeve (16) on the intermediate shaft. The bottom of the frame (18) acts as an oil container and encloses the pitch control gear, which consists of an oil distribution box (19) containing a slide valve (20) which allows the oil to pass under pressure to one side or the other of the servomotor piston. The oil is led through two ring chambers, each communicating with one of the radial holes (14). The slide valve is connected to a transverse rod (21) adjoining two parallel arms (22). The lower ends of the arms are linked to a sleeve (16) and the upper ends are in the form of toothed sectors which, through the pinions (23) and a control arm (24), are coupled to a remote control device. The main oil pressure pump (26) is of the Imo type, drawing oil from the sump in the frame and delivering it through a filter (27) to the distribution valve (20). A standby pump (29) is provided. The hub is supplied with a special grade of oil which is delivered to the inlet (31) and passes through holes (32), reaching the hollow propeller shaft and filling the hub, where it remains under constant pressure. The pumps (33, 34) are in duplicate, one being a standby. There is, however, an overhead tank which ensures sufficient oil pressure when the working pump is stopped. The control column (35) on the bridge has a lever (36) which alters the pitch of the blades and a compressed-air indicator (37) is provided. The engine revolutions are controlled from the bridge by hydraulic telemotor gear (55, 56). The remainder of the mechanism in the complete device is indicated in the table.—*The Motor Ship, Vol. 33, March 1953; p. 527.*

Fuels for Gas Turbines

It is generally agreed that the gas-turbine engine can expect no widespread application in the merchant marine, industrial, or locomotive fields until it is able to burn residual fuel oil. Gas-turbine engines will and are being used extensively in the industrial field where natural gas is an economic fuel. The ability of the gas-turbine engine to burn coal will further enhance its position in the industrial and locomotive fields if and when it is feasible. Inasmuch as this prime mover can now operate on gasoline, kerosene, distillate fuel, and natural gas, and as the problems yet to be solved in burning coal are well known, this discussion is limited to residual fuels. The actual design or fabrication of suitable combustion equipment for burning residual fuel oil is, in general, similar to that for distillate fuel. The heavier fuel may result in a greater tendency for temperature stratification at the outlet of some types of combustion chambers. The real problem is caused by burning an ash-bearing fuel which introduces the hazard of deposition in the combustion system, on turbine blades, and all other surfaces exposed to the exhaust gas. Numerous land gas-turbine engines have been run on residual fuel but they have either had relatively low turbine-inlet temperatures (below 1,200 deg. F.) or have had selected grades of such fuel. Even these units, in general, have experienced some difficulty. The Oerlikon Company's gas-turbine engine has been running on residual fuel for a long period, with some blade deposits (about 0·04 inch). Every two weeks these are easily washed off and no traces of corrosion remain. N.G.T.E. ran residual fuels in a jet engine with not only blade fouling but also a significant degree of blade alloy corrosion. The Metropolitan Vickers Gatric gas-turbine engine was run at idling speed for 20 minutes on residual fuel and then compressor stalling occurred.

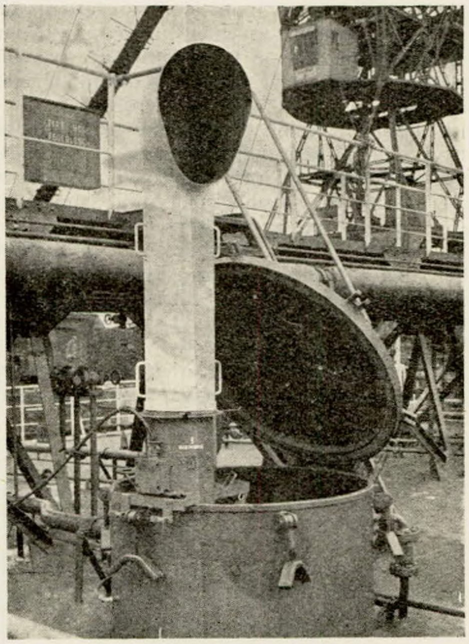
A heavy carbon deposit was found in the combustion chamber and on the high-pressure-turbine blading. Compressor stalling had been caused by partial blocking of the high-pressure-turbine blade passages. The experimental C. A. Parsons and Company 500-h.p. gas-turbine engine experienced compressor surge after 8 hours of operation on residual fuel which caused ash deposition, resulting in turbine blockage. Other companies, such as General Electric, Elliott, Westinghouse, and Brown Boveri have had considerable experience in burning residual fuels. As serious as the ash-deposition problem is, it is secondary to corrosion resulting from such deposition. The deposits may be cleaned off but the corrosion process is irreversible. Corrosion results from the presence of vanadium (in various forms, including vanadium pentoxide) (V_2O_5), salts (especially Na_2SO_4) and sulphur (S) being present in the fuel ash. High-temperature boilers and Diesel engines also suffer from these same impurities. It has been suggested that the residual fuels used in Diesel engines be limited to a low-ash and sulphur (2 per cent or less) content. Generally, sodium is in the fuel as a natural constituent but it may be introduced in the refinery process, or by salt-water contamination. This latter contaminating process will make it more difficult to burn residual fuel in the marine gas-turbine plant than in its land counterpart. The disastrous effects resulting from salt water in boiler fuels are well known. Most residual fuel oil has a considerable ash content. A large percentage of this ash may be composed of V_2O_5 and alkali compounds which combine to form low melting slags. Not only is vanadium present in almost every crude, but it is often the predominant impurity. A study also indicates that vanadium concentrations do not seem to occur in the absence of sulphur. Sodium is also a likely constituent, and, as has been mentioned in the case of marine service, may additionally be introduced by sea water. The relative percentages of these deleterious elements will vary according to the source of the fuel oil. It can be seen that this problem is complicated in marine plants because a vessel may receive fuel from every part of the world and, therefore, must be able to burn any fuel likely to be encountered.—*A. A. Hafer, Trans.A.S.M.E., Vol. 75, February 1953; pp. 127-136.*

Gas Turbine Progress—Marine

The gas turbine engine has now progressed to the point where its use is advantageous in a number of naval applications. These include small short-range boats, landing craft, PT boats, emergency power generation, and portable power equipment. Booster gas turbine engines combined with base-load Diesel, steam, or gas turbine plants are also attractive. In general, all of these applications but the base-load plant will utilize a simple-cycle engine. For merchant marine propulsion the gas turbine plant is not now economically superior to either the Diesel or steam plant except in special applications. The choice of a prime mover is basically a question of economies. After analysing such factors as engine first cost, maintenance, life, fuel consumption, and cost of operating personnel, the type of prime mover most advantageous for a particular installation is usually apparent. Such an analysis in the case of the naval prime mover is somewhat more involved as it is difficult to give an economic value to certain engine features. Any progress made towards higher turbine temperatures, ability to burn residual fuel, or a radical reduction in regenerator size will make the use of the marine gas turbine plant that much more advantageous. It should not be anticipated that the gas turbine engine will replace both the Diesel and steam plants in the near future, but that it will supplement them. It is safe to say that the number of marine gas turbine engines in service in several years will number in the hundreds. A large proportion of these will be small naval units. There is every reason to believe that progress will continue at a high rate during the next decade.—*W. A. Dolan and A. A. Hafer, Trans.A.S.M.E., Vol. 75, February 1953; pp. 169-175.*

De-aerating System for Tankers

The first vessel to be equipped with the new Siemens-Schuckert ventilating unit for cargo oil tanks is the motor tanker *Irvingbrook,* built by the Howaldtswerke A.G. for the Irving Oil Co., St. John's. It is claimed that, with this equipment, considerably faster and more reliable ventilation of cargo tanks can be effected than with the employment of canvas wind sails. The device comprises a three-phase induction motor enclosed in a pressure-resistant flame-proof casing, driving a ventilating fan which operates as a blower. The unit is installed on a portable frame which can be fixed either at the coaming of the manhole or placed in the Butterworth opening.



Siemens ventilating unit installed in manhole

Fresh air is introduced through a ventilator rotatable to any position and is delivered at a high speed towards the bottom of the tank. The return flow is mixed with the gas-laden air and is driven out through the manhole. The capacity of the fan is 3,000 cubic metres of fresh air per hour and, assuming the air volume of the tank to be changed six times hourly, a tank of 1,000 cu. m. capacity will be gas-free in about two hours, or sufficiently so as to allow the tank to be entered. As the fan motor is an A.C. machine, it will require to be connected to a three-phase distribution system on board.—*The Motor Ship, Vol. 33, March 1953; p. 525.*

Hazards in Operation of Tank Vessels

This paper is a discussion of some hazards encountered in the handling and transportation of bulk petroleum and petroleum products by tank vessel, with comment on how these hazards are controlled by vessel design, equipment, and safe working practices, together with some suggestions as to how the hazards might be better controlled. As a specific example, a fire occurred in the fire room of a tank vessel when Diesel oil was spilled on a superheated steam line which ran under the grating. Had the line been unlagged, there would have been no fire. The line was, however, lagged and there were breaks in the lagging in a way of a flange that allowed the oil to run down along the pipe. The lagging held the oil against the hot pipe permitting slow oxidation to proceed and it subsequently ignited. Corrective action was to so shield the steam line with metal as to deflect any oil that might be spilled on it. In new design, superheated steam lines on which petroleum products might be spilled accidentally should be suitably protected. Design should anticipate the possibility of a fireman removing a burner with the oil still turned on. A chronic offender in the fireroom as regards oil spillage is the fuel oil strainer. It should be so designed as to make unnecessary the removal of the covers or should be provided with a safety device that will make it impossible to remove the cover from the strainer in use. Strainers should be equipped with large easily cleaned drip pans. On Standard Oil Company of California tank vessels, vapours are contained within a closed venting system known as a vapour control system. Details of this system are shown in Fig. 1. If the tanks are out-breathing as when cargo is being loaded, vapours are discharged at the masthead vents, a location removed from probable sources of ignition. If the tanks are in-breathing, as when cargo is being discharged, air is likewise drawn from the masthead vent. The vapour control system differs from the conventional venting arrangement in two important respects. First, special weighted ullage hole covers are provided, which are equipped with the usual pin for securing when the vessel is underway. The pin is removed when handling cargo or ballast and the lead weight holds the cover closed against pressure from within the tank. Should this pressure exceed two pounds to the square inch the cover will lift and relieve the pressure to prevent possible structural damage. Second, a float type of gauge tape is provided on each cargo compartment. The top of the tape is covered by a gastight housing which is not shown in the drawing. The ullage is read through a glass window in the housing. These tapes allow the liquid levels in the tanks to be ascertained without necessitating the opening of the tanks' ullage covers. These two features, the weighted plug and the tape, together with the conventional venting system, effectively control vapours within the limits of the capacity of the system and, in the opinion of the author, contribute very greatly to the safety of operations.—*Paper by G. D. Washburn; Abstract in S.N.A.M.E. Bulletin, Vol. 8, January 1953; pp. 11-20.*

First Tanker for All-gas Turbine Propulsion

A contract of outstanding significance has been placed with Cammell Laird and Co., Ltd., of Birkenhead, by the Anglo-Saxon Petroleum Co., Ltd. (operators of the Shell tanker fleet), of London. This was for a single-screw vessel, of 18,000 tons deadweight, to be propelled exclusively by gas turbines—the first tank ship in the world with all-gas turbine propelling machinery. The ship will be constructed to a specification prepared by the owners, and the machinery to a specification compiled jointly by the British Thomson-Houston Co., Ltd., of Rugby, and the owners' research and development departments. The British Thomson-Houston Company will design and prepare the detailed plans of the gas turbines and A.C. electrical equipment, while Cammell Laird and Co., Ltd., will prepare the general arrangement plan of the engine room. This interesting vessel, which is scheduled for completion in 1956, will be one of the "general purpose" class of tank ships, of which fifty are either on order or under construction for the Shell fleet. As the other vessels will be powered by either steam turbo-electric machinery or by geared steam turbines, it will be possible to obtain a direct comparison of the operating

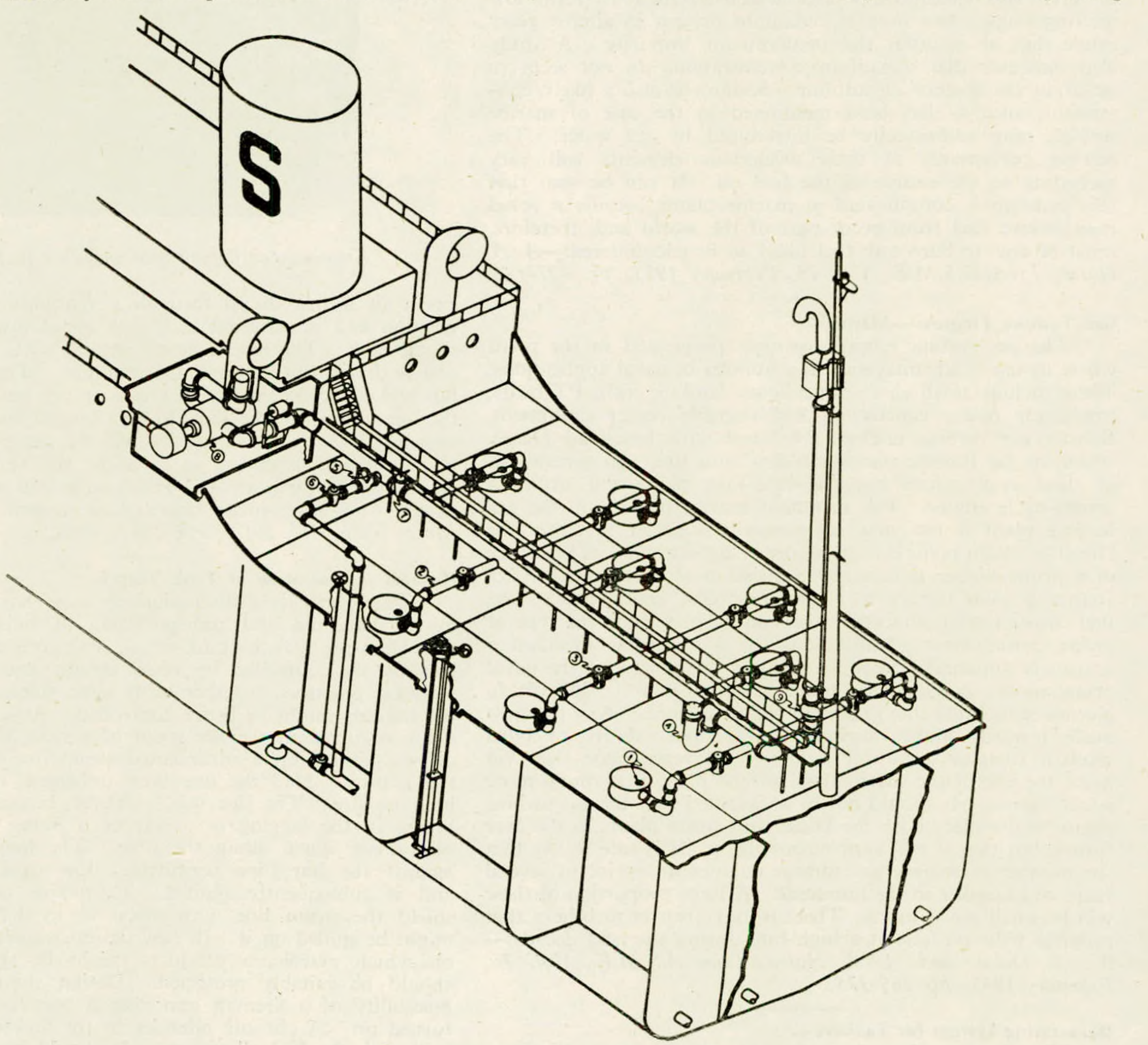


FIG. 1—*Vapour control system*

and maintenance costs of the three forms of propulsion. The all-gas turbine-driven tank vessel is the latest development in a series of experiments in marine propulsion with gas turbines which have been carried out since the war by the Shell marine research department, headed by Mr. John Lamb, O.B.E. The first success was achieved in 1951 with the *Auris,* which became the world's first merchant ship powered by a gas turbine engine, fitted in place of one of the vessel's four Diesel engines. This gas turbine functioned so well, that early in 1951, it propelled the *Auris* across the Atlantic without any assistance from the Diesel engines which were held in reserve. In common with all tankers being built for the Shell fleet, the all-gas turbine ship will be of the most modern design, incorporating newly invented equipment to ensure the safe handling of highly volatile cargoes and quick turn-round in port. The gas turbine components will be manufactured jointly by the British Thomson-Houston Co., Ltd., and Cammell Laird and Co., Ltd. The turbines will be tested at the latter company's Birkenhead works, under the supervision of the British Thomson-Houston Co., Ltd., before installation in the ship. The machinery will comprise two gas turbo-alternator units driving the single propeller shaft through two electric motors and a two-pinion single-reduction gearbox. Power for auxiliary requirements at sea and in port will be drawn from the propulsion gas turbines, including that required for the cargo pumps, which are capable of dealing with 2,000 tons of oil per hour. A Diesel alternator will be provided to supply power for the ship's services when the gas turbine alternators are not in use. Operating on the open cycle, each unit will have two-stage compression, with intercooling. The h.p. line will comprise turbine, compressor and geared auxiliary alternator with exciter set; while the l.p. line will comprise turbine, compressor and propulsion alternator. The initial starting of the turbines will be effected by steam turbines of 150 h.p. each. Steam for cargo heating and tank cleaning will be supplied by a small oil-burning boiler, in conjunction with waste heat boilers taking exhaust gas from the heat exchangers of the turbine. The propeller motors will be of the double-cage induction type. Astern running will be obtained by reversing contactors in the main electric circuit, and, consequently, with no change in the turbine direction of rotation and with full power available for astern running. Between 100 r.p.m. and approximately 55 r.p.m., the speed of the propeller will be controlled at the turbine. Lower propeller speeds will be obtained by reducing the voltage applied to the propeller induction motors, the controls being in the alternator exciter circuits. The four centrifugal cargo pumps and one of the two stripping pumps will be electrically driven, at 3,000 volts, from the propulsion alternator. The other stripping pump is of the steam reciprocating type.—*The Shipbuilder and Marine Engine-Builder, Vol. 60, 1953; p. 163.*

6,500 h.p. Marine Gas Turbine

The world's highest powered marine gas turbine, developing 6,500 h.p., built under a development contract for the Admiralty, has recently completed extensive shore tests at the Rugby works of the English Electric Co., Ltd. Prior to the completion of E.L.60A, as this turbine is officially known, the most powerful marine gas turbines were the 3,500 s.h.p. unit at Pametrada, the Allis-Chalmers 3,500 s.h.p. unit, and the Elliott 3,000 s.h.p. set, the two latter having been built in the U.S.A. for the American Bureau of Ships. These earlier units have all given satisfactory performances on test, but none of them were destined for actual marine propulsion service. The parallel flow cycle (Fig. 2) was chosen because it enabled a relatively small power turbine to be designed. This has to pass only one-third of the total mass flow of the cycle and was thus suitable to run at the 5,600 r.p.m. of the ship's existing alternator. The cycle was also an attractive one for manœuvring, as any power from full to zero could be provided by manipulation of the throttle and the blow-off valves, without interfering in any way with the speed, mass flow, or pressure ratio of the compressor. The charging set turbine has eight symmetrically-disposed gas inlet connexions and was designed to develop 13,700 h.p. at

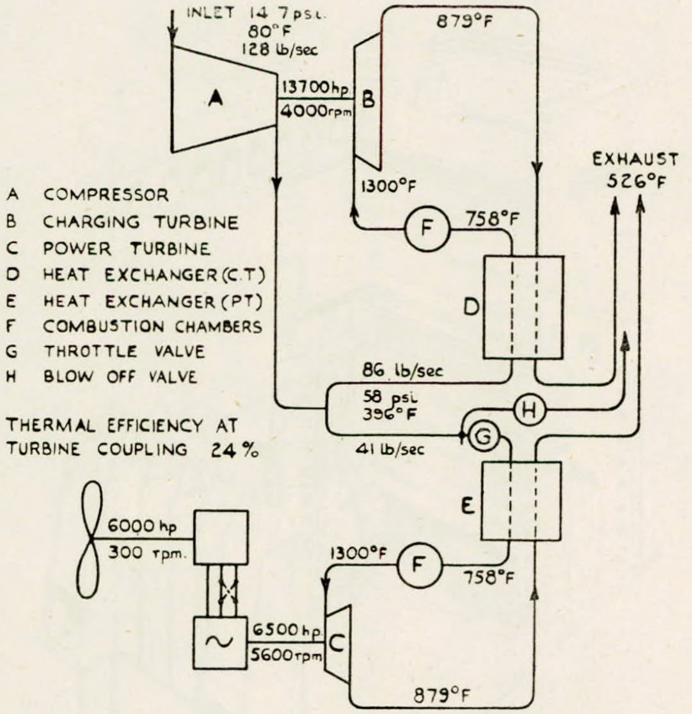


FIG. 2—*Cycle diagram showing design conditions. Note the power absorbed by the compressor*

4,000 r.p.m. under full power conditions. The 6,600 h.p. at 5,600 r.p.m. power turbine is geometrically similar in all respects including blade form and blade heights which are scaled down in the linear ratio of 0·688 to 1. It has, however, only four gas inlets. Both turbines were originally designed as six-stage axial-flow machines with vortex blading, but in final assembly the first stages were removed by the simple expedient of parting off the blades in a lathe, leaving the roots in position to provide a smooth unbroken surface on the rotor periphery. This action was taken so that the turbines might better match the actual output of the compressor. The turbine rotors were each built up by welding from three separate austenitic steel forgings, single forging of this size being unobtainable. The design of weld involved much research and a pair of full-size dummy rotor discs were first welded and subsequently sectioned. The success of the rotor cooling system referred to later has shown that the use of austenitic steel is unnecessary and future rotors of this size will be in the more readily available ferritic material. The rotor and the more heavily stressed parts of the blades are protected from direct impingement of the hot gases by a series of disc cover rings, built in segmental sections for expansion,

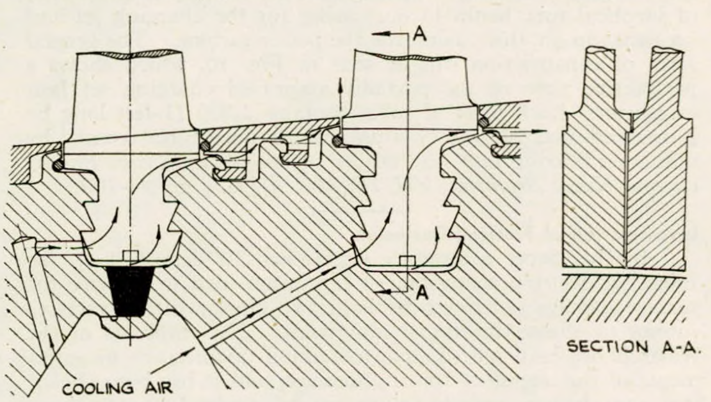


FIG. 7—*Section showing the path of the rotor cooling air and the weld between two rotor sections*

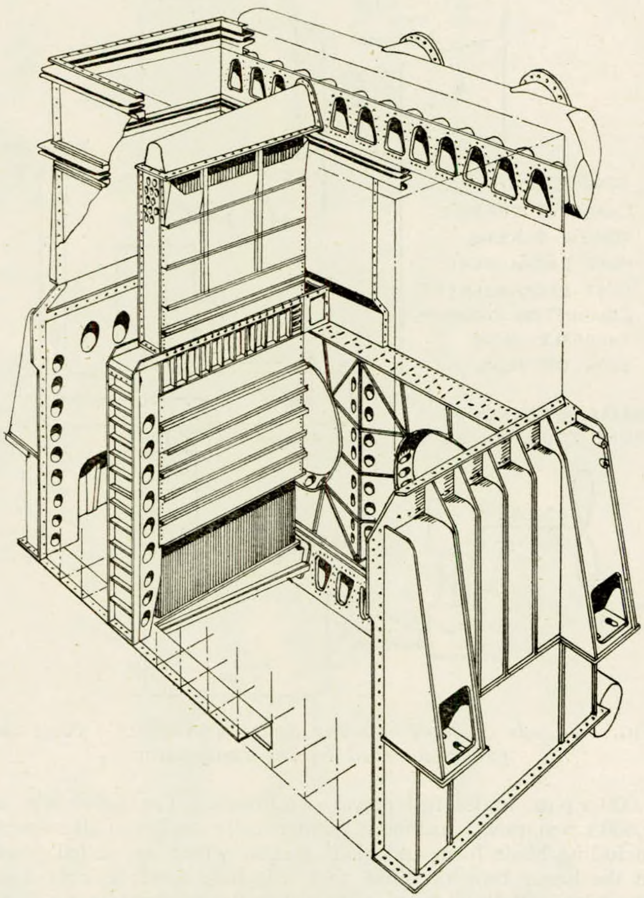


FIG. 10—*Perspective view of the charging set heat exchanger in a partially assembled condition. Only one of the twelve tube banks are shown in position*

and cooled by an air stream passing through the annular space between the disc covers and the periphery of the rotor. This cooling air supply is tapped from the compressor discharge and fed into the centre of the rotor through radial holes located between the h.p. end labyrinth glands. It is then distributed from the centre of the rotor through a number of small radial holes feeding into the blade roots and beneath the disc covers (Fig. 7). A series of calibrated fusible plugs, examined after about one hundred hours of full gas temperature running, showed that this air cooling system (which used less than 2 per cent of the main cycle air) was able to maintain the rotor temperature everywhere at least 250 deg. C. (450 deg. F.) below that of the hot gases. The heat exchangers were built up from a series of identical tube banks in one casing for the charging set and six banks in another casing for the power turbine. The general form of construction can be seen in Fig. 10, which shows a perspective view of the partially assembled charging set heat exchanger. Each bank of tubes contains 1,800 11-feet long by $\frac{3}{8}$-inch o.d. and 22 S.W.G. aluminium bronze tubes secured by induction brazing into steel header plates.—*The Marine Engineer and Naval Architect, Vol. 76, March 1953; pp. 94-100.*

Investigation of Fretting Corrosion

In this paper a series of experiments is described on the subject of fretting corrosion, an expression used to describe the surface damage occurring between two closely fitting surfaces subject to slight vibrational movement. The emphasis of the research has been placed upon obtaining quantitative measurements of the degree of fretting damage, and it has been shown that considerable reproducibility can be obtained, thus making it possible to correlate this measurement with such variables as total number of oscillations, load, or atmosphere humidity Humidity variations have been shown to have a pronounced effect on both the form and total amount of damage, and this probably accounts for the large discrepancies between the observations obtained by different workers. Although most of the measurements have been made using lapped carbon-steel specimens, some results were obtained with other materials such as chromium, nickel, and gold. Nickel plating has been advocated as an anti-fretting surface, but the experiments clearly show this to be ineffective, unless it can be assumed that the nickel plating has the additional effect of reducing the slip between the two surfaces. The use of lubricated phosphated surfaces can be strongly recommended for inhibiting fretting corrosion, from the results obtained with this form of surface preparation rubbing against a steel flat. Additional experiments have been carried out with carbon-steel surfaces to establish the significance of the absorption and diffusivity of oxygen in surrounding liquid media.—*Paper by K. H. R. Wright, read at a General Meeting of The Institution of Mechanical Engineers, 20th February 1953.*

Removable Tweendeck

This invention describes a removable platform providing a 'tween deck in a ship's hold. In Figs. 1 and 2 the hold (1) of a ship is divided by two vertical planes (2 and 3) into three parts (4, 5 and 6). The actual number of such parts will vary in different cases. In the invention a first series of panels (7) are provided to cover one part of the hold (4) hinged to the

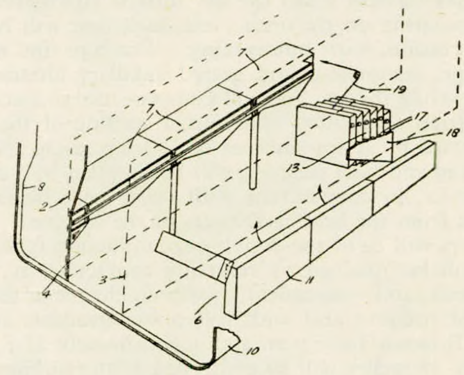


FIG. 1

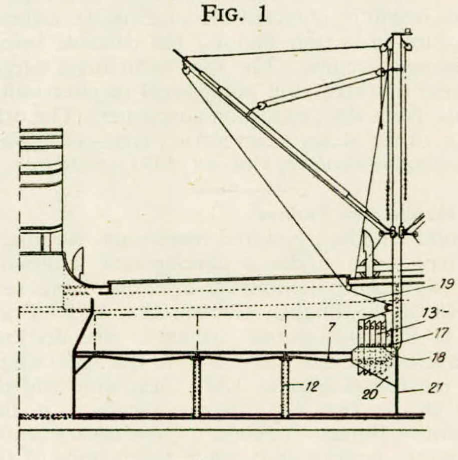


FIG. 2

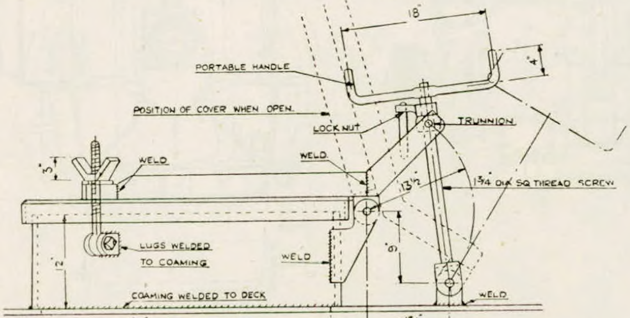
side wall (8). The other outboard part of the hold (6) is provided with a similar set of panels (11). Vertical girders (12) are provided in the planes (2 and 3) separating the hold parts, to support the ends of the panels. The central section (5) is covered by another set of panels (13). These are preferably capable of stowing vertically at one end of the hold, and may with advantage be of the MacGregor type. The panels shown are of a type described in British Patent No. 631,003.—*Patentee: H. Kummerman. British Patent No. 684,455. World Shipbuilding, Vol. 3, February 1953; p. 35.*

Effect of Radial Pitch Variation on Propeller Performance

This paper is a description of the results of an investigation, by means of strip-theory calculations, of the comparative effects of various forms of radial pitch-variation on the efficiency and performance characteristics of a marine propeller working in a uniform stream, and also in various non-uniform wake-streams representative of twin-screw and single-screw wake patterns. In this investigation, strip-theory calculations were first of all made for an actual four-bladed aerofoil propeller of good modern design, as fitted to a large twin-screw liner, and the results were found to be closely in agreement with expectations, and with the tank-test figures. The following conclusions are drawn: From the point of view of overall efficiency and apart from any consideration of cavitation or flow breakdown, there appears to be no material advantage to be gained from the adoption of a radial variation of pitch, both in a uniform stream and in a variable-wake stream. In particular, it seems that there is no special advantage to be gained from the application of the various alternative methods of design, based on the principle of minimum energy loss, which have been examined, as any gain which might be achieved in the ideal or hydrodynamic efficiency by the adoption of these procedures is very small, and is overshadowed by the effects of blade-efficiency (i.e. section-drag losses, etc.) introduced by the changes in angles of incidence where the wake concentration is high. On the other hand, the above results suggest that no special loss in efficiency is to be expected from the adoption of moderate pitch-variations which are favourable from the point of view of cavitation or flow breakdown, and this leaves the designer considerable freedom in the matter of adopting such alternative pitch-variation lines from root to tip of the blades as might be considered desirable from this point of view. It appears that the quantity designated relatively-rotative-efficiency has a real meaning, in terms of the methods of analysis usually adopted, and its value can be estimated by calculation, in the manner described in the paper. The numerical values obtained agree reasonably well with the experimental data. The nominal wake factor is slightly larger than the corresponding thrust-identity wake factor, and this in turn is somewhat greater than the equivalent torque-identity wake factor. It appears that the differences between these factors are not greatly dependent on pitch-ratio. The value of the nominal behind-efficiency η_M, for a propeller working in a variable-wake stream, appears to increase slightly as the value of W_N is increased, for the same value of J_N or nominal mean wake speed V_a. This is reflected in the value of the relative-rotative-efficiency, which increases with increase in the wake intensity, or as the variation in W' increases.—*Paper by Professor L. C. Burrill and C. S. Yang, read at a meeting of the Institution of Naval Architects, 26th March 1953.*

"Easifast" Hatch Covers

The patented "Easifast" hatch cover is adaptable to suit round, oval or rectangular hatches, the cover being strengthened by perimeter and other stiffeners to permit the reduction in the number of fasteners necessary to secure oiltightness. For example, a 4-ft. diameter hatch has three fasteners compared with the eight fasteners normally used with the older type of hatch. In the prototype hatch of this size the cover was tested to a head of 29 feet before showing signs of leakage (Lloyd's test for the hatches is 8-ft. head). The raising and lowering of the hatch cover is accomplished by means of a screw gear, the lower end of which is secured to the deck or to the hatch coaming (depending upon the height of hatch coaming required). Two of the cover stiffeners are extended over the hatch side to form a lifting lever, and a trunnion in which the screw works is fitted between the ends of these stiffeners. The turning of the screw in this trunnion raises or lowers the cover as required. Normally the hatch cover remains stationary in any position, but to provide additional security a lock nut is fitted above the trunnion to lock the cover at any position. This lock nut is fitted with a hinged drop handle which when not in use falls down between the stiffeners so that when the screw is operated the lock nut rises and falls with the trunnion and does not need to be separately operated. The hatch cover can be fully opened in about half a minute by one man and lowered in a few seconds, while the smaller number of fasteners means that the final tightening can be done in about one-quarter of the time required for a normal cover of the same size. The accompanying sketch shows a typical arrangement of a circular type hatch with a low coaming, but the design is readily adaptable to suit hatches of varying shape and heights of coamings.—*The Shipping World, Vol. 128, 25th March 1953; p. 307.*



Circular type hatch with "Easifast" cover

Propulsion Plant of Mariner Class Vessel

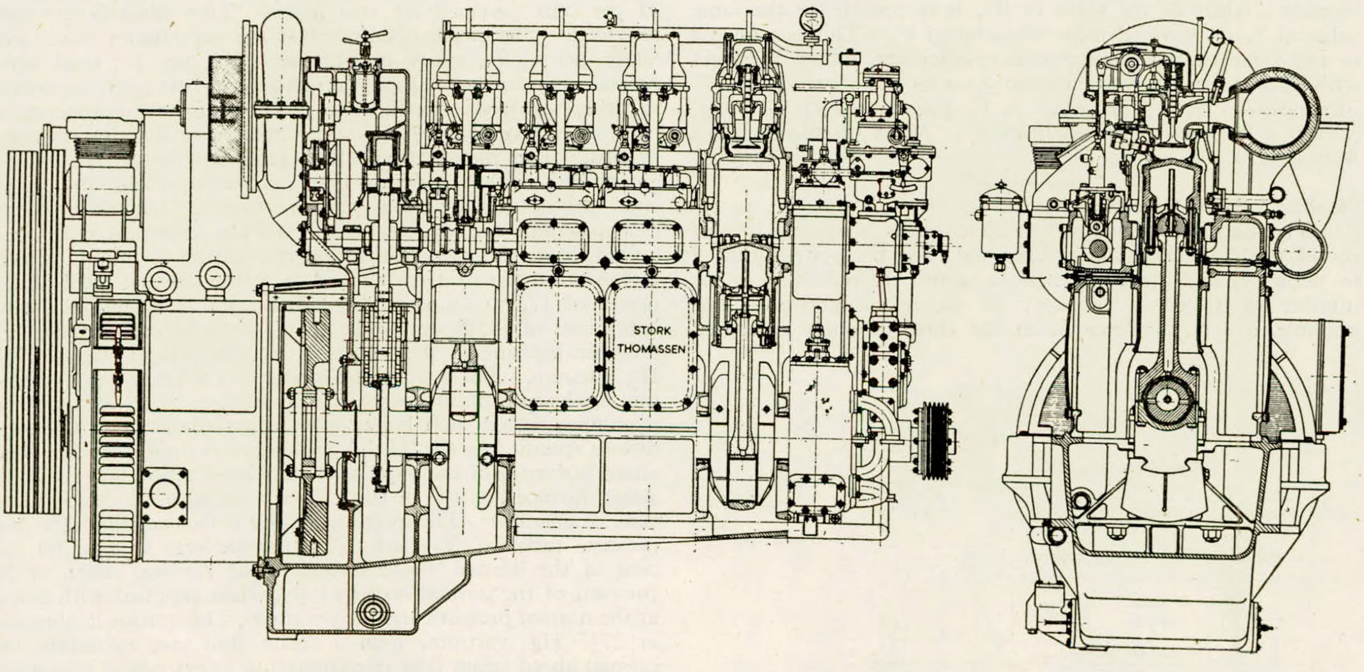
The main propulsion unit of Mariner class vessels consists of a 17,500 s.h.p. cross-compound, double reduction, geared turbine of the latest marine design and high efficiency, driving a propeller through a line of shafting at about 102 r.p.m. when delivering 17,500 s.h.p. The unit is capable of continuous operation at 19,250 s.h.p. under within A.B.S. design criteria. The turbines are designed to operate at normal power with 585lb. per sq. in. gauge pressure, 855 deg. F. total temperature steam at the throttles, and exhausting at 28½" Hg. vacuum with sea water at 75 deg. F. The electric generating system consists of two turbo-generators supplied with steam at the throttle at 585lb. per sq. in. gauge pressure, and 830 deg. F. total temperature and exhausting to individual condensers at 28½" Hg. Each unit is designed to deliver 600 kW., 450 volt, 3 phase, 60 cycle alternating current at 0·8 power factor continuously, and 25 per cent overload for two hours. Two main boilers are provided. The steam conditions at the superheater outlet are 600lb. per sq. in. gauge pressure and 865 deg. F.; total temperature, under normal power conditions; manually operated superheat control is provided to prevent the steam temperature from exceeding 865 deg. F. under maximum evaporation conditions. The main turbine and gear unit is of the cross-compound, double reduction type, consisting of one high pressure turbine and one low pressure turbine, including astern elements, and one double helical, double reduction gear connected to the turbines with flexible mechanical couplings. Each turbine and gear unit is designed to deliver normal rated ahead power of 17,500 s.h.p. at about 102 r.p.m. with initial steam conditions of 585lb. per sq. in. gauge pressure and 855 deg. F. total temperature at the throttle and exhausting under a 28½" Hg. vacuum. The unit is also designed for continuous operation at the maximum rated shaft horsepower of 19,250 and the maximum design shaft horsepower in accordance with the conditions specified by the (U.S.A.) Maritime Administration. The ahead turbine is of the impulse or impulse-reaction type, and the astern turbine consists of suitable impulse wheels located in the exhaust chamber. The main condenser is located below the low pressure turbine. The astern turbine delivers at least 80 per cent of the normal ahead torque, when turning astern at 50 per cent of the normal ahead r.p.m., when supplied with steam at the normal pressure and temperature. This torque is obtained at 27½" Hg. vacuum, with a steam flow not exceeding the normal ahead steam flow corresponding to extraction operation. The ahead and astern turbines are so designed that there is no overheating under continuous astern operation at 70 per cent of the normal ahead r.p.m. and approximately 35 per cent of normal ahead power with inlet steam supplied at the normal design pressure and temperature. The condenser vacuum for this operation is maintained at 27½" Hg. Thermometers are fitted at the throttle, h.p. turbine steam chest, first stage shell and all bleeder points. The ahead turbine is provided with sufficient hand controlled nozzle valves to obtain a relatively flat water rate curve from 12,000 to 19,250 s.h.p. Connexions are provided to permit independent operation of the high pressure and low pressure turbines for emergency purposes. Blank flanges, gaskets and piping are provided to facilitate this emergency operation. The high pressure turbine operating alone develops a shaft horsepower of 6,600 at 75 r.p.m. of the main shaft and the low pressure turbine operating alone develops a shaft horsepower of 4,000 at 64 r.p.m. of the main shaft. Steam for emergency operation of the low pressure turbine is taken from the desuperheated steam system. The design of all turbines and the connexions is such that with rated load and normal steam conditions, the inlet steam velocity does not exceed 9,000ft. per minute. Steam passages are smooth and free from burrs or other obstructions likely to cause eddies. At rated load and normal steam conditions, the exhaust velocity does not exceed 21,000ft. per min. for condensing operation.—*The Log, Vol. 48, March 1953; pp. 50-51.*

Dutch Two-stroke Engine

The new Stork-Thomassen two-stroke engine has a bore of 240 mm., a stroke of 360 mm., and a maximum speed of 600 r.p.m. The engine is shown in longitudinal and cross sections in the illustration. The engine has been designed with uniflow scavenging along accepted lines, the scavenging air being introduced through a number of ports in the cylinder wall during the lower part of the stroke and the exhaust gases being expelled through a central overhead valve in the cylinder head. As is known, the difficulty with two-stroke engines is to introduce the scavenging air in such a way that after the inlet and exhaust period the cylinder is as far as possible charged with clean air, while at the same time measures must be taken to prevent this air from mixing with the exhaust gases of the previous power stroke. In this respect the new design appears to have its merits, since the results achieved are such that the smoke limit of the engine lies at a mean effective pressure (*pe*) of 7·2 kg. per sq. cm. (103lb. per sq. in.). To enable a high mean effective pressure, which is generally accompanied by a high heat load, also to be carried with safety as the designed normal engine load, a number of special features have been incorporated in the design, particularly with regard to cooling. So as to be on the safe side a normal (or commercial full load) mean effective pressure of 6 kg. per sq. cm. (85lb. per sq. in.) has been adhered to for the time being. For scavenging air supply comparative tests were carried out with both Roots and centrifugal blowers; as a result of these tests it was decided to employ the latter. The speed of the centrifugal blower used on the standard engines is about ten times that of the crankshaft.—*P. C. Brunting and G. Wieberdink, Gas and Oil Power, Vol. 48, March 1953; pp. 55-57.*

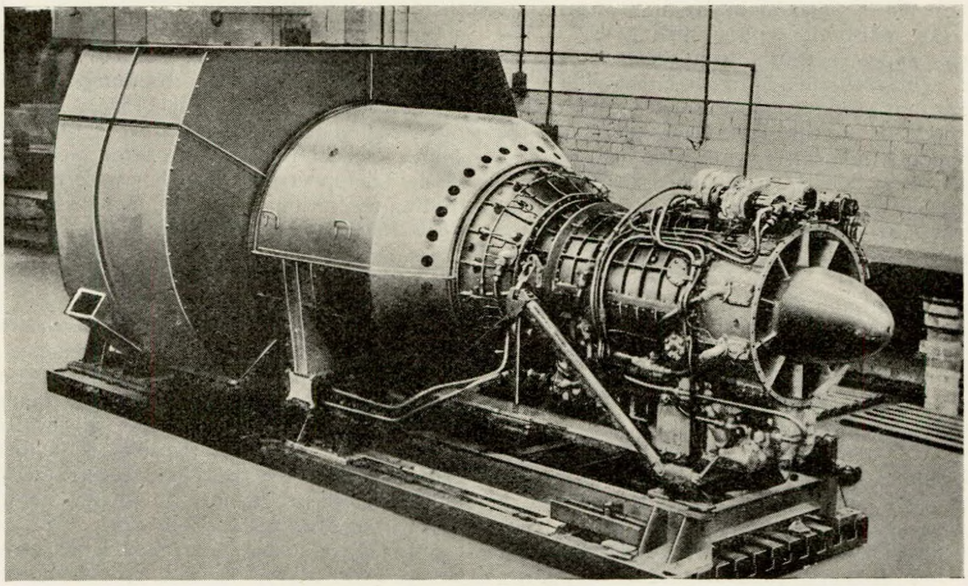
The G2 Gas Turbine

Details have been issued by the Metropolitan-Vickers Electrical Co., Ltd., of the G2 gas turbines which they have constructed for the Admiralty. Following the successful trials of the Metropolitan-Vickers Gatric gas turbine in MGB 2009, an order was placed for four basically similar but larger gas turbines to give further operating experience at sea. These G2 gas turbines are now at sea in the Coastal Forces craft *Bold Pioneer* and *Bold Pathfinder*. Another two gas turbines of this type have been ordered by the United States Navy for trial and evaluation purposes. The Gatric gas turbine was of the open cycle type, without heat exchanger, and consisted of a gas generator which was based on the F2 jet engine, and a power turbine. The G2 gas turbine employs the same cycle as the Gatric engine. The gas generator (compressor, combustion chamber, compressor turbine and auxiliaries) is, with certain modifications, that of the Beryl jet engine. The G2 gas turbine is designed to develop 4,800 s.h.p. Pending operational experience at sea, however, the maximum power has been declared at 4,500 s.h.p., with a maximum continuous rating of 4,000 s.h.p. The G2 has been designed for a total life of 1,000 hours, of which 300 may be run at maximum power. These figures are low by marine standards, but the engine, of course, is intended for development purposes, while its use as a booster allows a shorter life requirement. A complete G2 unit consists of a gas generator, a power turbine with exhaust duct, and a reduction gear. The power turbine, which is mechani-



New Stork-Thomassen two-stroke engine

cally independent of the gas generator, drives the propeller shaft through a single-helical, side pinion reduction gear. The 4,500 s.h.p. gas turbines have a much better power/weight ratio than the 2,500 s.h.p. Gatric unit. The gas generator is supported by a built-up frame on either side of the compressor, bolted to the engine bearers; the frames are so pivoted that they can swing outwards to allow for radial expansion. The forward end of output is effected solely by regulating the fuel quantity to the combustion chamber of the gas generator; the speed of the gas generator, which depends upon the fuel quantity, determines the temperature, pressure, and flow rate of the gas at the inlet to the power turbine. A lightweight compressed air motor, supplied with high-pressure air at 450lb. per sq. in. gauge from storage bottles by way of a reducing valve, is used for starting



The Metropolitan-Vickers G2 naval gas turbine, showing the protective aluminium casing

the compressor is supported under the inlet branch by a small pivoted frame which permits axial expansion. Alignment of the unit is maintained by a sliding key under the compressor outlet branch. The compressor has eleven axial flow stages, giving an overall pressure ratio of 4·03 at an air-mass flow of 62·6lb. per sec. for a maximum speed of 7,830 r.p.m. The rotor consists of an inlet end piece of manganese-molybdenum steel, a drum front extension, a drum centre portion and drum conical extension pieces of aluminium alloy, and a turbine end shaft of nickel-chrome-molybdenum steel. The compressor is of aluminium alloy. The rotor is carried by two bearings, one ball and one roller, which are lubricated by an oil mist. The moving and fixed compressor blades are of aluminium alloy; the moving blades are carried in longitudinal serrated grooves in the rotor and the fixed blades in dovetail grooves in the casing. Provision is made for the removal of salt deposits from the compressor blading by a distilled water spray. The single stage compressor turbine has a molybdenum-vanadium steel disc mounted on a steel shaft attached to the compressor drum conical extension piece. The forward end of the power turbine is carried by two semi-flexible legs attached to brackets on the outlet branch and mounted on a cross beam bolted to the engine bearers; the aft end of the turbine is supported from the gearbox. The maximum design speed of the power turbine is 5,200 r.p.m. The mean gas temperature at the power turbine inlet, measured by four thermo-couples, is 660 deg. C.; due to stratification of temperature across the annulus, this figure is some 20 deg. C. higher than the average temperature calculated from fuel flow. The power turbine drives the propeller through a side pinion single-reduction gear encased in a lightweight rigid casing of all-welded steel construction. The overall gear ratio is 4·73 and the maximum propeller shaft speed is 1,100 r.p.m. The bearings for both the pinion and wheel shafts are of the roller type; a Michell thrust block is incorporated in the aft end of the gear case. Solid flange couplings join the pinion and power turbine shafts, and the slow speed wheel shaft to that of the propeller. The engines are designed to operate on pool gas oil. The control of power the gas generator; at a speed of approximately 2,000 r.p.m. the unit becomes self-sustaining and the motor is no longer required.—*The Shipping World, Vol. 128, 7th January 1953; pp. 13-14.*

Ship's Propeller

This invention is intended to produce a lighter propeller by a more scientific calculation of the propeller thickness required from the aspect of bending strength. Fig. 3 shows a

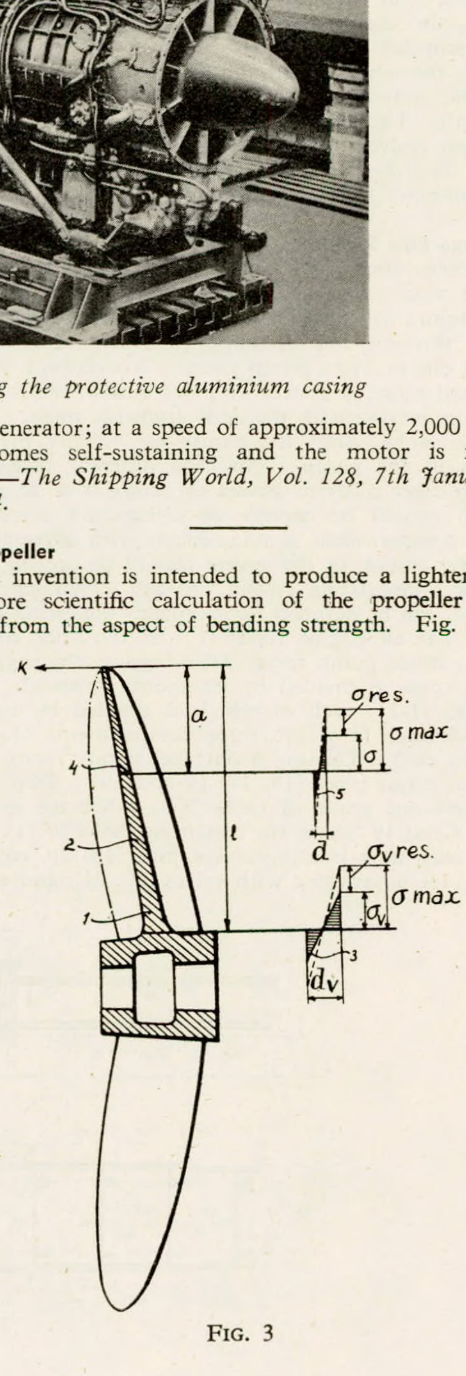


FIG. 3

four-bladed propeller. Assuming that the magnitude and distribution of the propulsive, tangential, centrifugal, and frictional forces acting upon a blade are known, it is possible to determine the stresses, in particular the bending stresses, at the root (1) of the upper blade (2) when the root is of given dimensions. In the drawing the graph of moments (3) has been given for this purpose. The forces mentioned thus set up a maximum bending stress σ_v. The maximum strength of the material may be known and represents the maximum bending stress σ_{max} which the material can withstand without giving way. The difference between σ_{max} and σ_v is thus a "stress reserve" referred to as $\sigma_{v,\,res}$, which represents an additional stress which the blade can absorb when any unexpected force acts on the blade. At a section (4) of the blade the calculable forces also cause the blade to bend, producing a moment (5) with bending stress (σ). The thickness (d) of the section is so calculated that an additional force (K) acting at the tip of the blade, which sets up the stress $\sigma_{v,\,res}$ at the foot will also set up a stress σ_{res} at the section (4), also corresponding to σ_{max} of the material. Thus, when the propeller is operating, a force K will give rise to the σ_{max} along the whole length of the blade: there is no most unfavourable section, and the blade will tend to bend along its whole length. The blade will thus be considerably lighter and thinner than conventional blades.—*Patentee: M. M. H. Lips, Drunen. (British Patent No. 679,593). World Shipbuilding, Vol. 2, November 1952; pp. 150-151.*

Tanker Cargo Pipe System

The cargo space of a tanker is normally divided into a number of water tight tanks by transverse bulkheads and one or two longitudinal bulkheads. The contents of the tanks are discharged through deck discharge pipes by means of pumps, situated in one or more pump rooms. To connect the pumps to individual tanks, a system of pipes and valves is provided. This, in conjunction with the deck discharge pipes, also serves to load the tanks, when the pumps are usually bypassed by connexions in the pump room. As different tanks may be required to carry different grades of cargo, it is desirable that the system should be capable of filling and discharging a number of compartments simultaneously with different cargoes, and that any section of the system should be capable of forming part of a circulating system through which sea water, or other liquid, can be pumped for cleaning. Referring to Fig. 7, the tanker has an engine room (12) at the after end of the ship, and a single pump room (13) adjoining the engine room. The cargo space is divided by transverse bulkheads (14) into eleven tanks (1-11) each of which is divided by two longitudinal bulkheads (15) into three compartments (1a, 1b, 1c, 2a, 2b, 2c, etc.). Leading from the pump room are four longitudinal cargo pipes (16, 17, 18 and 19). Two pipes (16 and 17) serve one group of tanks (1-5), while the second pair of pipes (18 and 19) serves the remaining tanks (6-11). Associated with each tank is a cross-over pipe (1p, 2p, etc.). Each cross-over pipe is provided with valves (20, 21), and with three outlet pipes (22a, 22b and 22c) connecting the pipe with the three sections of the associated tank. Each outlet pipe has a valve (23). The cross-over pipes divide the longitudinal pipes into a number of sections. At the beginning of each section of each pipe a valve (34) is provided. Another valve (35) is provided at the end of the sections between pipes 1p and 2p and pipes 6p and 7p respectively. The ends of the shorter longitudinal pipes (18 and 19) are connected to adjacent points of the longer pipes (16 and 17). Each of these connexions is controlled by a valve (26). The various longitudinal and cross-over pipes are fitted in the normal way with expansion fittings (27). Also leading from the pump room (13) are four deck discharge pipes (28, 29, 30 and 31). These discharge into cross pipes each having an outlet valve (32) on either side of the ship. Each of the deck discharge pipes (28-31) has a strainer (33). The upper plan also shows the valve wheels (36) for controlling the various valves and the tank hatches (37).—*Patentees: Shell Refining and Marketing Co., Ltd., London. (British Patent No. 678,259). World Shipbuilding, Vol. 2, November 1952; p. 152.*

Development and Maintenance of Post-war Naval Machinery

A review of the pre-war maintenance policy of the Fleet and the interdependence of design and maintenance shows how it became necessary to alter the naval maintenance and design policies so that advanced designs of naval machinery could be developed for warships capable of maintenance by a ship's personnel with the tools and equipment provided on board. Following these changes in policy certain principles were adopted and reorganizations took place which, when taken together, have enabled extremely rapid advances to be made in nearly every type of warship machinery. This involved advances in the practice and techniques of manufacture and some examples of the results of post-war naval machinery designs are given. The ultimate aim of naval maintenance and repair is to enable a warship to maintain its efficiency, be always available when required and to "keep the seas" for as long as operational commitments demand, and the paper concludes with a review of maintenance and repair developments in the Fleet affecting both post-war and earlier naval machinery designs.—*Paper by Commander(E) A. F. Smith, R.N., read at a meeting of the North-East Coast Institution of Engineers and Shipbuilders, 20th March 1953.*

Torsional Fatigue Strength of Marine Shafting

The paper deals with tests on $9\frac{3}{4}$-in diameter forged mild steel shafts similar to those normally used in merchant vessels. Details of production of the test shafts, the testing machine and methods of calibration, test and control are given and the results of the large-scale tests compared with theoretical and small-scale test results obtained by other investigators. Supplementary chemical, physical, metallurgical and small-scale fatigue tests were carried out to determine the properties and structure of the test shaft material, the strain-hardening and penetration of

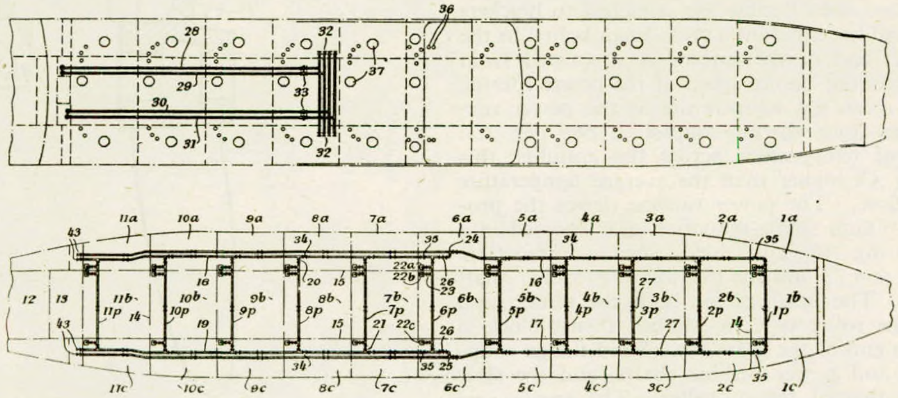


FIG. 7

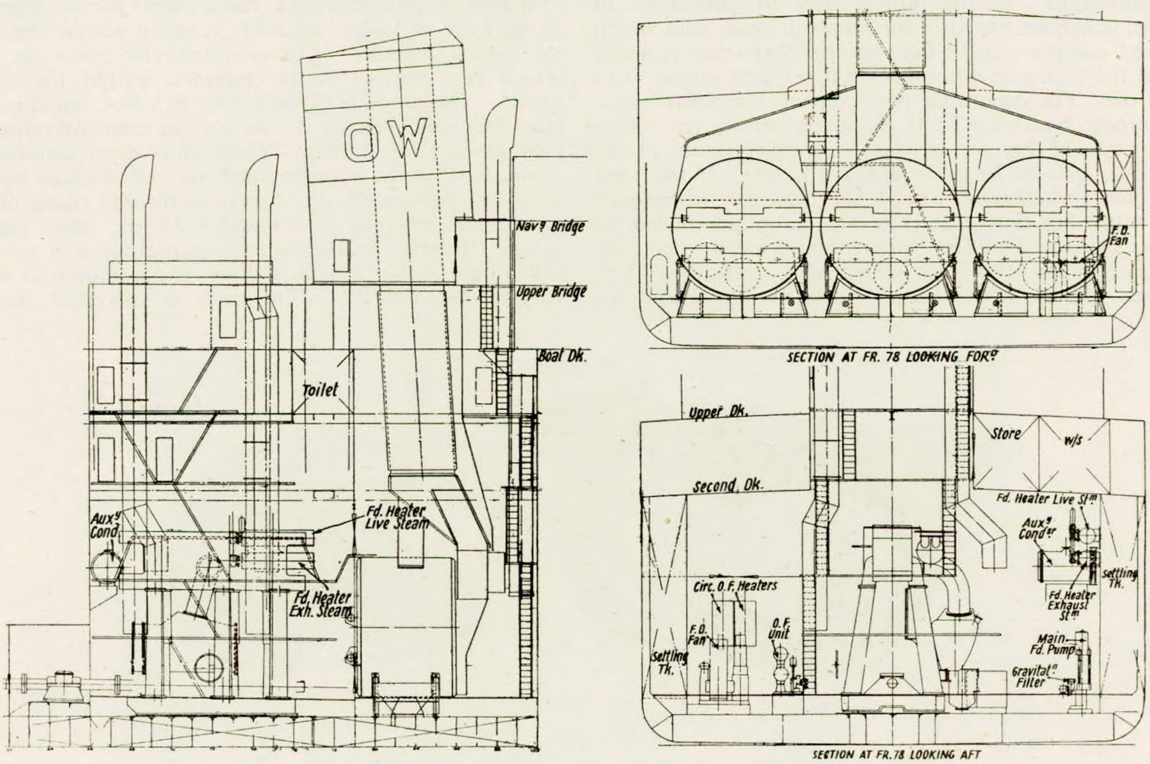
the fatigue cracks, the unnotched size effect and the influence of such factors as surface finish and mild understressing in unnotched torsional fatigue. Among the conclusions drawn by the author are the following: The torsional fatigue strength of 9¾-in. diameter forged mild-steel shafts is considerably reduced by reducing the fillet radius, the rate of strength reduction increasing with the sharpness of the fillet. A fillet radius (r) of $\frac{1}{16}$th of the shaft diameter (d) reduces the fatigue strength by 25 per cent while a fillet radius of 1/64th of the shaft diameter reduces it by 40 per cent. There is no appreciable size effect for unnotched specimens of the same material subjected to reversed torsional fatigue loading. The size effect for notched specimens is not yet known and is at present under investigation. Variations in surface roughness from a commercial smooth turned finish to a polished surface finish do not appreciably affect the torsional fatigue strength of ½-in. diameter unnotched mild-steel specimens. The presence of uniformly distributed "A" segregates which were found from sulphur prints and micrographs to be coincident with the fillet did not appear to influence the direction of crack propagation. The distinct forms of crack initiation and propagation were observed in the large scale specimens, apparently depending on the fillet radius. With specimens having a sharp fillet, that is $r/d = \frac{1}{16}$ and less, a large number of cracks were initiated around the fillet and developed very slowly, whereas with large fillets a single crack was usually initiated and developed rapidly. The strain-hardening produced by the fatigue testing is highest in the region of the stress concentration and along the lines of crack propagation.—*Paper by T. W. Bunyan and H. H. Attia, read at a meeting of The Institution of Engineers and Shipbuilders in Scotland on 7th April 1953.*

Steamship with 3,900 i.h.p. Reheat Engine

The single-screw cargo steamship *Bohème,* built by Bartram and Sons for Rederi A/B Wallenco of Stockholm, is equipped with the highest powered reciprocating steam unit to be completed by the North-Eastern Marine Engineering Co. (1938), Ltd. Indeed, the designed service output of 3,900 i.h.p. at 94 r.p.m. is well within the range which nowadays is almost the exclusive preserve of the Diesel engine. The *Bohème* is an open shelter-decker built to the requirements of Lloyd's Register of Shipping, the Swedish Board of Trade and the Ministry of Transport. She is intended for the owners' tramp chartering, with special provisions for the Narvik iron-ore trade. The principal particulars are as follows: —

Length between perpendiculars	445ft. 0in.
Breadth	62ft. 0in.
Moulded depth	40ft. 2in.
Deadweight capacity on 26ft. 11½in. draught	10,400 tons
Service speed with 3,900 i.h.p.	12·5 knots

The N.E.M. reheater engine has cylinders of 27-inch, 44-inch and 76-inch bore with a stroke of 51 inches. The cylinders are arranged in the customary N.E.M. disposition of h.p., i.p. and m.p., which provides improved balance, very accessible valve gear and easier steam passages. The designed output is 3,900 i.h.p. at 94 r.p.m. in service when burning 34 tons of fuel per day, and 4,500 i.h.p. at 98 r.p.m. on trials. Cam-operated poppet valves are fitted to the h.p. and m.p. cylinders while a Martin and Andrews balanced slide valve is fitted to the l.p. cylinder. A number of new features have been incorporated in this particular engine, in view of its somewhat higher running speed. In particular the lubricating system has been modified as a result of investigations carried out over the past eighteen months. A closed lubricating system is now employed for the top end brasses, oil being pumped by T and K lubricators through flexible pipes attached to the indicator rods. The bottom end bearings are supplied with oil through banjo rings attached to the crankwebs, a supply being taken to the inside of the bearings. Oil cups are retained for reserve purposes and these can be fed from spare lines provided in the T and K lubricators. This arrangement should result in a considerable economy in lubricating oil. Another modification is the provision for h.p. and i.p. valve gear tappet adjustment, while running. The expansion at these points is considerable and if left at the correct thirty thousandths of an inch when hot, the valves would be slightly opened when the plant cooled down. All eight valves can be adjusted in twelve minutes. A third modification is the provision of a fabricated l.p. piston, from considerations of weight and balance. Solid top end brasses and welded cross-



Elevation and sections through the machinery spaces of the Bohème head shoes are now standard practice in N.E.M. reciprocating engines and have been found very successful.—*The Marine Engineer and Naval Architect, Vol. 76, April 1953; pp. 135-141.*

Difficulties in Driving Generator from Propeller Shaft

A recent small vessel was equipped with a Diesel engine which was coupled to the propeller shaft through a reversing gear. The speed of the engine was variable between 110 and 320 r.p.m., and it was decided that the drive to the main generator should be taken from the propeller shaft between the engine and the gear box. The advantage of this arrangement is that no battery is necessary, since the main generator is producing electrical power all the time the main engine is running, and at other times power can be drawn either from the shore supply or from a small auxiliary generator. In practice, however, it was found that when the engine speed fell below 170 r.p.m. the lights had a pronounced flicker, which became intolerable at the lowest speeds. The cause of this flicker was found to be an insufficiently high moment of inertia of the flywheel, with the result that the speed of the engine was affected by the power strokes of the individual pistons. Since it was not possible to install a larger flywheel, an electrical solution was required. This took the form of connecting an electric motor in parallel with the generator. As the generator output falls, so the motor, which by virtue of its inertia continues to turn at speed, becomes a generator, and feeds power into the circuit. This "electrical flywheel" was found to eliminate the flicker.—*W. van den Born, Schip en Werf, Vol. 20, 2nd January 1953; p. 13; Journal, The British Shipbuilding Research Association, Vol. 8, March 1953; Abstract No. 7,312.*

Use of Boiler Oil in Diesel Engines

For several years the Fiat Company has guaranteed its medium and long stroke engines, both of the single-acting and double-acting types, to operate continuously on boiler oil. The Soc. Italnavi runs its eight motor ships exclusively on bunker oil C. The consumption of boiler oil is usually 7 to 8 per cent higher than that of Diesel oil, due mainly to the lower calorific value of boiler oil (about 5 per cent), combined with a slight diminution in thermal efficiency due to the slower and retarded combustion. Taking into account the impurities in the boiler oil, which are expelled during purification, and which do not exceed one per cent, it may be said that when running on boiler oil the total consumption is 8-9 per cent greater than with Diesel oil. The maximum pressures are somewhat lower, being in the neighbourhood of 48 to 50 kg. per sq. cm., compared with 54 to 65 kg. per sq. cm. The temperature of the exhaust gases is usually 20 to 30 deg. C. higher. At one time there was some apprehension due to the high rate of wear of the piston rings, but now that these are chromium plated no difference is experienced between the wear occasioned when running on boiler oil or Diesel oil. They last at least 8,000 to 10,000 hours. If the correct adjustments are made and the right temperatures maintained, the cylinder liner wear is between 40 per cent and 70 per cent greater when running on boiler oil, that is to say, about 0·17 to 0·18 mm. per 1,000 hours, compared with 0·10 mm. on Diesel oil. The injectors have a slightly shorter life because the nozzle holes are subject to more rapid wear on account of corrosive elements in the boiler oils, but the additional loss involved is very small. It may be taken that the cylinder liner wear averages 65 per cent more than that with Diesel oil. With the Fiat engines of large size having the upper section of the liner (also the lower section of double-acting engines) in steel, with inserted rings of special cast iron, only the lower section and the rings in the upper section need be replaced. It is accepted that the life of a liner running on Diesel oil is six to seven years (with a liner wear of 0·8 to 0·12 mm. per 1,000 hr.) compared with from 3½ to 4 years with boiler oil, the mean wear being 0·16 to 0·20 mm. per 1,000 hours. Taking all the factors into consideration, therefore, it is found that with single-acting engines of high power the net economy in the fuel bill when operating on boiler oil is 26·3 per cent. If the steam required for heating the oil is obtained from an exhaust gas boiler the economy is greater. An examination of figures shows that the total additional expenditure for all purposes involved with the employment of boiler oil represents about 12 per cent of the fuel economy so that even if additional expenses come into question, due to any special circumstances, there is still a great saving to be effected. It is also to be noted that after about 200 hours' running the cost of the additional installation is covered.—*Paper by E. Cotti and A. Marie, read at the Internal Combustion Engine International Congress, Milan, 1953. The Motor Ship, Vol. 34, May 1953; pp. 61-62.*

New Compressor Type

Experience gained at the Slough laboratory of the British Internal-combustion Engine Research Association by testing Roots-type pressure chargers, and knowledge of engine air requirements, showed that this type of compressor was far from ideal as a positive-displacement pressure charger. It was, therefore, decided to develop a more suitable design. The casing of the new compressor is of figure-of-eight form, similar to that of a Roots-type compressor, but without ports. The rotors are in the form of hollow cylinders running within this casing and are geared together. On each cylinder rotor are two lobes which run, with working clearance, within the bore of the casing. On one side of each lobe is a slot, equal in length to the lobe, which allows a lobe on the contra-rotating rotor to pass at each revolution. These slots communicate with the centre bore of the cylinders and are also used as ports for the inlet and delivery of air. Ducts in the end plates of the outer casing are arranged to connect with the inner bores of the rotors. Inserted in the bore of each rotor is a tubular valve, illustrated in Fig. 2, which shuts off the port over any desired part of the rotation. The cycle is described pictorially in

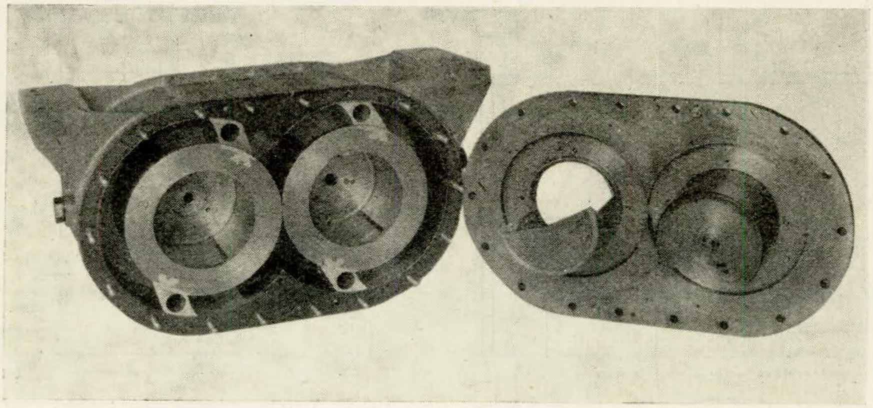


FIG. 2—*Tubular valve*

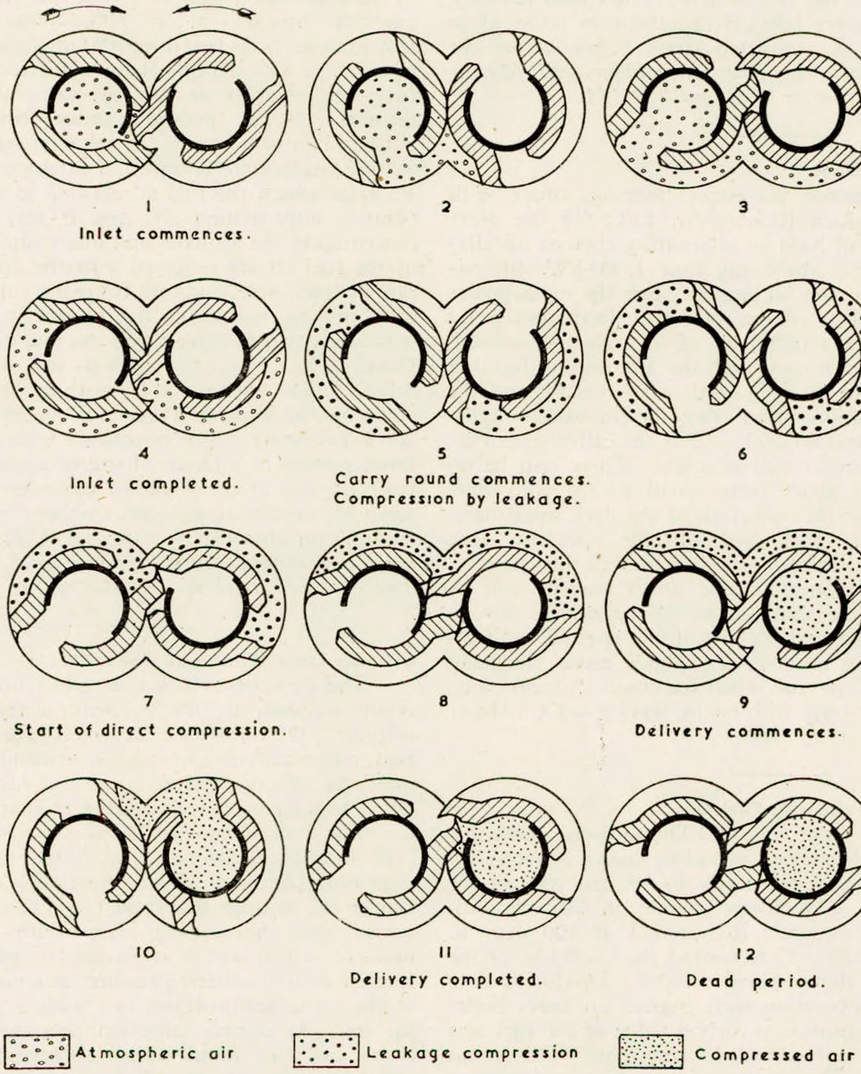


FIG. 3—*Diagrammatic path of a single air charge through two-lobe compressor*

Fig. 3, which shows the path of a single charge through one complete cycle. Diagrams 1 to 4 (Fig. 3) show how the inlet air is drawn into the compressor through the left-hand rotor, thus filling the spaces between the lobes. This air is divided into two volumes, known as the "carry round" volumes, and taken round the periphery of the rotors until the position shown by diagram 7 is reached, when the lobe of the right-hand rotor dips into the pocket of air carried round by the left-hand rotor, and direct compression commences. Compression continues to the desired pressure, and is controlled by the position of the tubular delivery valve, after which delivery takes place, as shown by diagrams 9 to 11. When the position shown in diagram 11 has been reached, the delivery is completed, and the valve closes to stop back-flow. Diagram 12 shows the "dead" period, during which the lobes move into position to start the next cycle. It will be noted that the rotor positions 1, 4 and 9 are similar and show the beginning of three cycles. The pressure-volume diagram describing this cycle is shown in Fig. 4. It is descriptive only, and intended to indicate where the positions shown in Fig. 3 occur in the compression process. Thus, although the toe of the diagram depicts the saving obtained by leakage recovery, it does not give a true estimate of the work saved. At point 1, the inlet port is opening and stays open through points 2 and 3 until just before point 4, when the port starts to close. After this, the air is carried round the periphery of the rotors without the volume changing, but with the pressure rising as the air leaking past the lobes from the high-pressure side is trapped. Only a very small proportion escapes past the second lobes into the inlet space, as the pressure ratio across these lobes is much less. At point 7, compression begins and continues through point 8 to point 9, where the valve starts to open. As with piston-type compressors, there is a slight over-compression while the valve is opening, after which delivery is at a steady rate through point 10 at the delivery pressure plus the delivery loss. Just before point 11, the delivery loss rises slightly before falling, as, at first, the

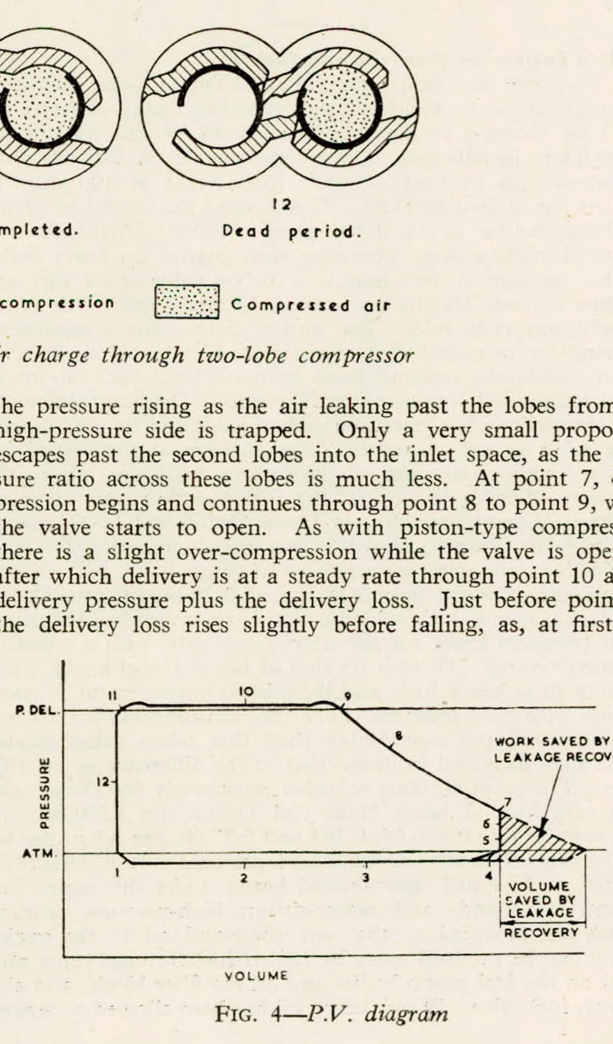


FIG. 4—*P.V. diagram*

valve starts closing before the rate of delivery has been reduced, and then the rate of delivery falls off rapidly from point 11 to point 12, when delivery is completed about 5 deg. before the valve closes.—*D. W. Tryhorn, The Shipbuilder and Marine Engine-Builder, Vol. 60, March 1953; pp. 139-144.*

A.C. Plant for Passenger Liner

The transatlantic motor passenger liner on order with Swan, Hunter and Wigham Richardson, Ltd., for the Norwegian American Line will have an alternating current installation, the generating plant comprising four 1,000-kW. alternators, each driven by a Ruston oil engine. For the main power circuits the busbar voltage will be 440, the alternators being three-phase machines with a frequency of 60 cycles per second. The smaller electrical machinery and the fluorescent lighting will operate on a 220-volt 60-cycle single-phase circuit and the complete lighting installation, apart from the fluorescent lighting, will be at 110 volts and 60 cycles. For the call system, etc., a 12- or 24-volt A.C. circuit will be used. There will be no direct current on board apart from small rotary converters situated in deckhouses for the operation of the deck machinery. It is possible, however, that hydraulically operated deck machinery may be utilized instead of electrical, and this question is now being examined. In that case direct current will be wholly eliminated from the ship. Most of the electric motors will run at constant speed and will be of the normal squirrel-cage type. The pumping units in the engine room are being split up in different sizes, so that when the engines operate, say, at half-load only one pump will be in service.—*The Motor Ship, Vol. 33, March 1953; p. 519.*

Doxford Engines on High-viscosity Fuel Oils

There are now nearly one hundred Doxford-engined ships operating on heavy boiler oil and including many conversions. They are showing savings approaching £8,000 per annum for 3,000 b.h.p. installations, and mostly operate on fuel oils with a viscosity up to 1,500 seconds Redwood I at 100 deg. F. Owners are advised by Doxford's to restrict the viscosities of the fuels they bunker to this figure where possible. Doxford's have proposed to shipowners operating their engines on heavy boiler oil that the sum of the Conradson carbon value of the fuel and sulphur content should not exceed 12 per cent. This is an entirely empirical rule. The author gave some comparisons showing the principal characteristics and chemical analyses of various fuel oils, ranging from marine Diesel fuel oil up to 3,500 seconds oil obtained from Curaçao. The main difference is in their viscosity, Conradson carbon value and cetane number. In a table in the paper the consumption and pressures of Doxford engines running on the various fuels were given, the primary difference being that the fuel injection was advanced 3 degrees of crank angle for the Haifa, Ordoil and 3,500 seconds fuel and the injection pressure was also increased. The fuel nozzles have the same arrangement of seven holes, but the size of hole was increased from 0·25 inch to 0·28 inch for the Ordoil and 3,500 seconds fuel. Combustion is smoother and the maximum pressures lower for the heavier fuel oils, with a constant injection timing. There is no sign of heavy Diesel knock when burning these heavy fuels and there is no improvement in combustion with pilot injection. The fuel consumption is only one per cent or 2 per cent higher than that when using marine Diesel fuel, increased in proportion to the difference in calorific value. The consumptions recorded respectively for Diesel fuel, Admiralty fuel oil, heavy Haifa fuel, Ordoil and 3,500 seconds fuel were 0·34, 0·355, 0·36, 0·364 and 0·372lb. per b.h.p. per hr. Mechanical efficiency was about 86 per cent. It is Doxford's practice to fit a fuel steam-heated heater under the engine fuel pumps, and hand- and motor-driven high-pressure priming pumps are provided so that any congealed oil in the engine piping can be pumped away by heated fuel through vents provided on the fuel pump bodies and on the filter blocks and also on each fuel valve. Where heavy oil has been allowed to remain in the pipes for a week or more, this clearing out of the piping may take upwards of one hour. The fuel oil should reach a temperature prior to injection into the engine such that the viscosity is 150 seconds Redwood I and even with the heaviest fuels the viscosity at the fuel valve should be less than 300 seconds. It has become standard practice to centrifuge the boiler oil in two stages: in the first stage water, sand and the heavier sludges are removed at a temperature of about 185 deg. F., after which the fuel is reheated to a somewhat higher temperature approaching 200 deg. F. and in this second stage of centrifuging the asphalts and other similar unburnable fractions of the fuel oil are removed. In the author's view, the average rate of liner wear, judging from the running of a large number of ships on boiler oil during the past four or five years, is between two to three times the rate of wear when running on Diesel fuels. For 1,500 seconds fuel having a sulphur content below 3 per cent and a Conradson carbon value of under 10 per cent, the maximum wear rate is between 0·007 and 0·009in. per 1,000 hours. The maximum wear for both the upper and lower pistons of a Doxford engine appears to be some 10 inches from the end of the stroke of the inner piston ring of each piston. Maximum wear occurs further down the stroke than when running on Diesel oil.—*Paper by P. Jackson, read at the International Combustion Engine International Congress, Milan, 1953; The Motor Ship, Vol. 34, May 1953; p. 60.*

Cylinder Liner Wear with Boiler Oil

The elements which may affect liner wear, apart from the cetane number, are the viscosity, asphalt, carbon residues and sulphur. Of these the viscosity is the most important factor. Bad pulverization can lead to considerable increase in liner wear; for effective pulverization the viscosity should be brought down to a figure corresponding to that of Diesel oil at 15 deg. C. The Fiat Company limits the viscosity to 6 deg. Engler (180 seconds Redwood I) with the recommendation that for maximum efficiency of pulverization it should not exceed 2 deg. Engler (57 seconds Redwood I). The little-known fact is mentioned that the viscosity rises with pressure and that, for instance, a fuel which at 50 deg. C. has a viscosity of 10 deg. Engler, at atmospheric pressure, as a viscosity of 45 deg. Engler at the same temperature, but with a pressure of 700 kg. per sq. cm. At normal injection pressures in large engines, viscosity can vary rapidly with a variation in temperature. For instance, with a bunker fuel of 60 deg. Engler at 50 deg. C., if the normal temperature of heating is reduced 20 deg. C. (this temperature being 150 deg. C.) the viscosity is doubled. The practical result is that the boiler oil should be heated to a degree dependent on the ignition pressure. As practically all bunker oils employed in marine engines have viscosities from 8 to 60 deg. Engler at 50 deg. C., it is necessary to keep them respectively between a minimum of 120 deg. C. and a maximum of 150 deg. C. Changes in the temperature of heating should be avoided, otherwise they influence pulverization. In general, it is considered desirable that the fuel before injection should not have an ash content above 0·02 per cent or 0·03 per cent. The asphalt which can be partially eliminated with the purifiers and clarifiers should be dealt with at a temperature not exceeding 80 to 90 deg. C., since at higher temperatures the asphalt tends to become fluid and its ejection by separation is rendered impossible.—*Paper by G. Simonetti, read at the Internal Combustion Engine International Congress, Milan, 1953. The Motor Ship, Vol. 34, May 1953; p. 63.*

Boiler Oil in Two-stroke Engines

At the beginning of operation on boiler oil the liner wear increased between 10 per cent and 30 per cent, but some time ago there were indications that the increase in liner wear may reach 100 per cent. Some exhaust valves had been pitted and valve stems corroded and the pitting was overcome by stelliting the valves. A table is given in the paper showing the amount of sludge found in purifiers and clarifiers per 1,000 litres, and this varies from 82 gr. in the purifier and 15·5 gr. in the clarifier per 1,000 litres, with fuel having a viscosity of 400 seconds Redwood I, up to 639 gr. per 1,000 litres in the purifier and 156 gr. per litre in the clarifier, in a fuel of 3,300 seconds Redwood I viscosity. The amount with fuels of assumedly the same viscosity varied substantially. The author considers that the ash content may be the possible cause of the cylinder liner wear and is discouraged to learn that the difference in ash content before and after centrifuging falls within the analytical error.—*Paper by D. T. Ruys, read at the Internal Combustion Engine International Congress, Milan, 1953. The Motor Ship, Vol. 34, May 1953; p. 62.*

Oil Fired Boiler

In a boiler of the type shown in Fig. 1, the tubes adjacent to the burner side will generate steam at higher rates than the remaining tubes of the bank. This condition prevails particularly at high rates of steaming at which a distinct temperature gradient exists in the flue gases flowing over the tube bank, the gas temperature decreasing from the end of the bank receiving the predominant portion of the heat toward the other end. This unequal distribution of heat to the bank of steam generating tubes may be caused by a number of influences, such as flow of the products of combustion longitudinally to the drum or bank, whereby the products of combustion impinge against a wall and pile up there to be deflected into the adjacent end of the tube bank. This results in a greater portion of the flue gases passing through the end of the bank so affected. As outlined in Figs. 1-3, the burners mounted in openings (12) direct the flame through the front furnace wall (9) into the furnace in direction longitudinal to the steam drum (1) and to the banks of tubes (3) and (5) and toward the rear furnace wall (10). The products of combustion pile up against wall (10), and the pressure of the flue gases, therefore, is higher there than toward the burner wall (9). This results in relatively increased rates of flow of the flue gases over each tube bank portion adjacent to wall (10) so that the rate in steam generation is increased in the portions affected. Obviously the same unbalanced heat flow to the main tube bank will occur if the flue gases are forced by baffles to enter the bank at one end. The result of the increased steam and water discharge from the steaming tubes receiving the greater proportion of the heat of the flue gases is to raise the water level in the steam and water drum (1) above these tubes, while the water level above the remaining tubes is at a substantially lower level. The resulting water level in the drum may be such as shown by the line (27) in Fig. 3. As a consequence the place where the greatest proportionate steam release occurs in the drum has the least steam space above the water level for the separation of steam from water, which, of course, is objectionable. To overcome this

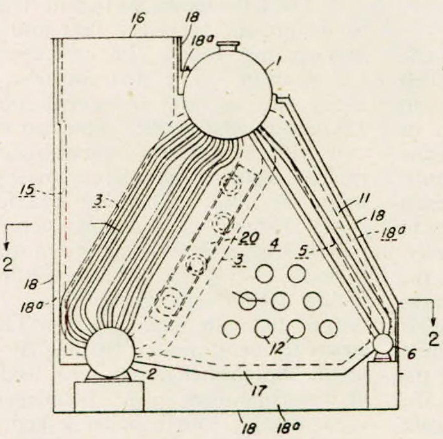


FIG. 1

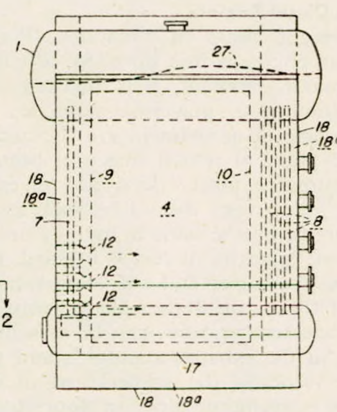


FIG. 3

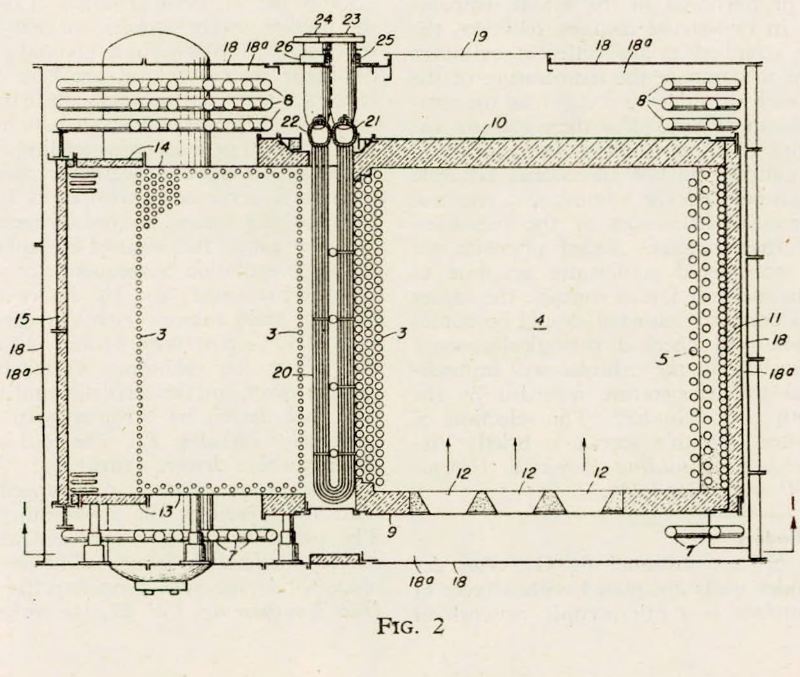


FIG. 2

objection the end of the steam drum (1) experiencing the greatest proportional steam generation is provided, according to the invention with a downcomer area to the water drum or drums (2 and 6) which is greater than the downcomer area at the opposite end of the steam drum. Such a greater downcomer area (8) withdraws a proportionately greater amount of water from the overloaded rear end of the steam drum and thereby lowers the water level adjacent to that end. By appropriate choice of the two downcomer areas (8 and 7) the water level in the steam drum (1) may be caused to be substantially the same from one drum end to the other. In this manner a greater average steam releasing space becomes available, which is most desirable. Furthermore, the greater amount of water withdrawn from the most active portion of the steam drum will be delivered to the most active portion of the water drum, thereby ensuring ample supply of water to the tube bank where the circulation is most rapid. Also there will be a substantial reduction in the longitudinal flow of water through the steam drum (1) from the overactive to the underactive end. In the lower water drums (2 and 6) a minimum end flow is desirable since the water in it is thus quieter and the tendency for solids to settle out increases. In the upper drum, the improvement effected contributes to the maintenance of a more nearly horizontal water level.—*British Patent No. 687,711, issued to Combustion Engineering-Superheater, Inc. Engineering and Boiler House Review, Vol. 68, April 1953; pp. 128-129.*

Correct Adjustment of Marine Diesel Engines

As M. Cariou in a recent issue of "Navires, Ports, Chantiers" points out, when an engine is first installed, or when it has been subjected to a major overhaul, it is necessary to adjust it so that the same amount of power is developed in each cylinder. The items capable of adjustment are the quantity of fuel injected, and the point at which injection begins. It is not practicable to measure the power developed in each cylinder directly, so that the adjustment should be such as to result in the maximum pressure being the same in each cylinder. If this is achieved, and the same quantity of fuel is injected, the cycle in each cylinder will be the same, and the temperatures of the gas in each cylinder at the moment the exhaust valve is opened will be the same. These temperatures can be measured only by thermometers placed in the exhaust manifolds and the readings obtained are average values of the temperatures of the gases of combustion and the scavenging air. In four-stroke engines the scavenging air is fairly evenly distributed between the various cylinders, and the temperatures recorded by the thermometers are sensibly proportional to the actual temperatures of the exhaust gases; in two-stroke engines, however, the quantity of scavenging air supplied to the different cylinders varies, and it is not possible to compare the temperature of the products of combustion in each cylinder by comparing the temperatures recorded on the thermometers. For these reasons, the use of thermometer readings as a measure of the maximum pressure attained in each cylinder is not considered accurate enough for the initial adjustment of the engine, and recourse should be had to the direct measurement of the pressures. Although the readings of thermometers placed opposite the exhaust openings are not considered sufficiently accurate to serve as a basis for the adjustment of Diesel engines, the values obtained when the engine is correctly adjusted should be noted; and these temperatures should be checked periodically, since any abnormal conditions existing in the cylinder will immediately result in a change in the temperature recorded by the thermometer associated with that cylinder. The selection of suitable types of thermometers for this service is briefly discussed.—*Journal, The British Shipbuilding Research Association, Vol. 8, March 1953; Abstract No. 7,286.*

Porous Chromium-plated Cylinders

In the Porus Krome process invented by Dr. Van der Horst in Holland, the cylinder walls are plated with a layer of porous chromium. The surface is a microscopic network of pores and cracks, which hold the lubricating oil, and so ensure good lubrication and reduce cylinder wear. The process has been adopted in several countries; and in 1949, for example, cylinders plated with porous chromium were provided for new ships totalling 161,600 tons, and had replaced cast-iron cylinders in 147,200 tons of existing shipping. The new cylinder liners have been fitted on the Norwegian ship *Oslofjord,* and will also be incorporated in her sister ship, the *Bergenfjord,* to be built by Swan, Hunter and Wigham Richardson, Ltd. In the U.S., the Van der Horst process is being extensively developed. The layer of porous chromium, whose thickness ranges from 0·005 to 0·015 inch, extends the life of a liner 3 to 7 times, according to the type of engine. The layers are deposited electrolytically on the base metal and no intermediate layers are used. The cylinder is placed in a chromium bath and acts as the cathode while the chromium is being deposited by a current of 40,000 amperes. Then the cylinder is placed in another bath and the current is reversed; it is during this part of the process that the pores are formed. As it is important to avoid friction between two very hard surfaces, the chromium should never be plated on both the cylinder and the piston rings.—*Holland Shipbuilding, Vol. 1, No. 10, 1952; p. 12. Journal, The British Shipbuilding Research Association, Vol. 8, February 1953; Abstract No. 7,176.*

New Great Lakes Ferries

The Chesapeake and Ohio Railway is adding two new ships to its present car ferry fleet and lengthening and repowering two existing vessels. The two new ships are the s.s. *Badger* and s.s. *Spartan.* The *Spartan* was commissioned in November 1952, and is now in service on the railroad's Milwaukee-Ludington lake route. The two sister ships are 410 feet long, and will carry thirty-two railroad freight cars (or 150 automobiles), plus hundreds of passengers, at normal speeds of 18 miles per hour. They are said to be the world's largest and fastest freight car and auto-carrying steamers. Each vessel is powered by an 8,000 h.p. steam plant that includes four Foster Wheeler D-type two-drum coal-fired boilers, arranged in pairs facing a transverse firing aisle. Each has a normal evaporation of 29,500lb. steam per hr. They are designed for operating steam temperatures of 750 deg. F. and 450lb. per sq. in. gauge at the superheater outlet, with feed temperatures at 225 deg. F. at the economizer inlet. In normal operation, three boilers are used, while a fourth boiler is kept available for inspection and cleaning. This arrangement is necessary to the full-time operation of the ship. Each boiler is provided with a coil-type 2,500lb. per hr. desuperheater. Coal handling equipment from the bunker to the stokers consists of two pan-type conveyors that receive coal through manually operated hopper-gates and discharge into two coal crushers located in the coal bunker space. These coal crushers discharge through watertight gates into two "L"-type conveyors which, in turn, discharge through four screw-type conveyors feeding individual day bunkers over the stokers. General Regulator combustion control is provided with three separate control units for each boiler. These controls are: air volume control by regulation of the forced draught fan inlet vanes, fuel control by stoker feeder regulation, furnace draught regulation by sequence control of the induced draught fan turbine speed, and the boiler outlet dampers. Each ship has two main engines with a normal rating of 3,500 shaft h.p. each at 125 r.p.m. with 440lb., 740 deg. F. steam at the throttle and 27½ inches vacuum. There are also two main generators on each ship, for the auxiliary and hotel load, rated at 500 kW. each and driven by turbines with steam at 440lb. per sq. in. gauge and 740 deg. F. The feed systems each consist of two steam turbine driven, centrifugal type feed pumps with differential pressure control so arranged that the control is automatically served by the maximum pressure boiler in operation. The propeller shafting for each ship is supported in water-cooled line shaft bearings. The shafts are equipped with four-blade solid-type steel propellers for operation in icy waters.—*Heat Engineering, Vol. 27, December 1952; pp. 192-196.*

ENGINEERING ABSTRACTS

Section 3. Shipbuilding and Marine Engineering

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Heat Transfer Through Boiler Tubes

Results of induction-heated experimental boiler studies of tube-metal temperatures are shown in this paper for heat-transfer rates from 140,000 to 300,000 B.Th.U. per sq. ft. per hr. and for pressures from 250 to 2,500lb. per sq. in. Clean tube conditions and different kinds of deposits were also studied. The approximate relationship between metal thickness, heat-transfer rate, and temperature drop through the metal are shown in graphic form. Approximate relationship between scale thickness and temperature increase in the tube metal at a given heat-transfer rate and pressure is shown for four scales. Some of the factors which affected the accuracy of the results are: eccentric position of the induction-heating coil, distortion of the heat-flow pattern by the thermocouple hole, poor thermal contact between thermocouple and tube metal, conduction through the thermocouple wires, non-uniform distribution of steam bubbles and circulation pattern over the heated area of the tube, stray e.m.f. in the thermocouple from the high-frequency induction field, variation of heat-conductivity coefficient with temperature, both of the tube metal and the deposit, diverging heat pattern in the wall of the tube, and change in film coefficient between metal-water and deposit-water interfaces due to changes in surface area and roughness. In spite of the many variables present in this method of testing, the results are in good agreement with the results of previous experimental work by other investigators, the paper states. However, the observation made by previous investigators that porous deposits have low heat conductivities and dense deposits have high heat conductivities does not appear to be confirmed by this investigation. The relative positions of the deposits investigated based on their heat-conductivity coefficients also coincide with generally accepted field experience.—*Paper by C. Jacklin, read at the 1952 A.S.M.E. Annual Meeting. Paper No. 52-A-30.*

Rotary Air Preheater for Gas Turbines

This paper is confined to a discussion of the rotary type of regenerative heat exchanger (Ljungström type), although several of the problems discussed could apply equally well to other types. This paper is concerned with a regenerator used as the air preheater in a simple open-cycle gas turbine. However, the conditions relating to working temperature, temperature and pressure gradients, and other factors are so severe that they may cover several other applications beyond the gas turbine. Advantages of the regenerative-type heat exchanger are that small hydraulic diameters can be readily attained, and that when the normal counterflow arrangement is used the matrix is self-cleaning to some extent, particularly during that part of the operating cycle when high velocities and pressure differences are experienced. Against this must be taken the problem of providing an efficient seal between the air and gas flows. In comparison, if flame-trap matrixes are employed in both, the recuperator can be designed to have hydraulic diameters of the same order as those in the regenerator. The matrix, however, is not self-cleaning, and thus continuous outside provision must be made to keep it in working order. This necessarilv involves an equivalent power loss, but one which can justifiably be compared with the auxiliary power losses of the regenerator. The separation of the gas and air flows, however, must be made in the matrix itself, which complicates the problem of cheap and simple manufacture.—*Paper by A. T. Bowden and W. Hryniszak, read at the 1952 A.S.M.E. Annual Meeting. Paper No. 52-A-74.*

Quick Closing Valves

This invention relates to a quick closing valve. In Fig. 8 the valve body (1) contains a movable valve closure member (2) entrained by a nut (3) which engages a screw-threaded spindle (4). The latter can slide axially through a gland (5). The spindle carries a handle (6) and is pressed by a spring (7) to slide into the valve body (1). The links (8 and 9) of a toggle are connected together by a knuckle pin (10), the links (8) being pivoted at (11) to a bridge (12). The latter is entrained by the spindle (4), and the links (9) are pivoted at (13) to the valve body (1). A trip member (14) is pivoted at one end to the knuckle pin (10) of the toggle, the other end being engageable with a fixed seat (15). The levers (16) are designed to swing around the respective pivot (13) and are pivoted at (17) to a piston rod (18) which is connected to a piston (19). This

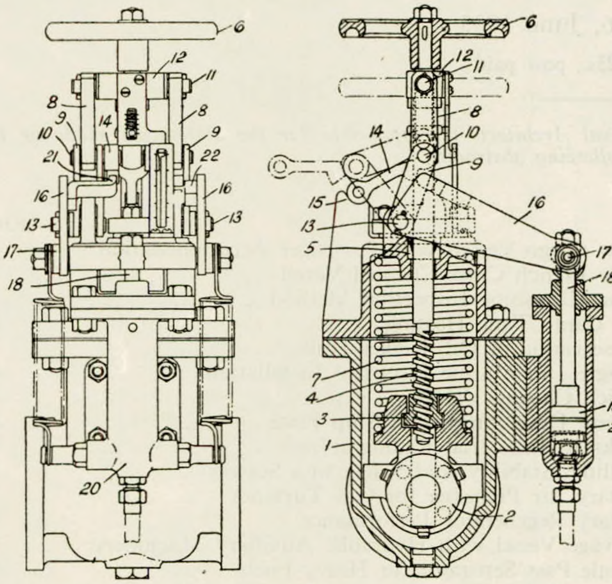


FIG. 8

piston slides in a cylinder (20). One lever (16) carries an abutment (21) which can engage the trip member (14), and the other lever carries an abutment (22) which can engage with the links (9) of the toggle. In normal operation, the toggle links (8 and 9) are almost in alignment, the trip member (14) being engaged with the fixed seat (15) and the piston (19) being at the lower end of the cylinder (20). When the valve is in this position, the valve closure member (2) may be moved between the valve-closed and the valve-open positions by rotating the spindle. When the valve is open and is to be closed rapidly, the piston (19) is moved upwards by admission of operating fluid to the lower end of the cylinder (20), the levers (16) swing about the pivots (13), and the abutment (21) comes into contact with the trip member (14) from its seat (15). The abutment (22) thereupon breaks the toggle and the spring (7) slides the spindle (4) into the valve body (1) and thus effects closure of the valve. When the valve is to be reset, the operating fluid in the cylinder (20), below the piston (19) is allowed to escape to exhaust.—*British Patent No. 688,141, issued to James Howden and Co., Ltd. Engineering and Boiler House Review, Vol. 68, April 1953; p. 131.*

Fatigue in Ship Structures

The incidence of fatigue may be considered of small importance for ship structures. Stresses of a high enough amplitude to produce fatigue failure may not be applied frequently enough during the lifetime of a ship, which may explain the fact that, from such information as can be obtained, fatigue failures in ships seem rare. Even if they occurred more frequently than they appear to do the problem may not be serious, since a fatigue failure—unless it sets off a brittle fracture—would be expected to propagate comparatively slowly and the damage, if it occurred after a number of years in service, could be expected in general to be repaired at small expense. It would not be reasonable on account of fatigue to alter design practice with regard to certain common details in ships if such alterations were likely to lead to considerable increases in the cost of building. Even if fatigue failures could be totally eliminated the cost of such measures would be a multiple of the probable cost of repair. To take a concrete example: whilst on the basis of these tests continuous fillet welds are undoubtedly a better proposition from the point of view of fatigue than intermittent welds, the increased cost of welding, if all intermittent welding were replaced by continuous welding, could not be justified. On the other hand, if as has been shown the fatigue strength of butt welds reinforced by straps is not only not increased but is likely to be reduced, such butt straps could readily be dispensed with, and in this way a definite improvement in the fatigue strength of the structure and perhaps a small saving in cost can be obtained. If butt welds, however, were of such poor quality that they must be considered to be insufficiently strong even under static loading, the remedy should not be seen in reinforcing straps but only in an improvement in the quality of welding. If there should be other compelling reasons for using strap reinforcements for butt welds, an angle strap of the Type K is much to be preferred. Scalloped stiffeners attached by intermittent chain welding have not shown up very favourably in this investigation. Whilst it is quite possible that a different form of scallop might give better results than those obtained in the present series of tests, they could really only be justified if appreciable saving in dead weight and cost can be achieved by using scallops, unless fatigue as a likely form of failure is totally ignored.—*Paper by R. Weck, read at a meeting of the Institution of Naval Architects, 26th March 1953.*

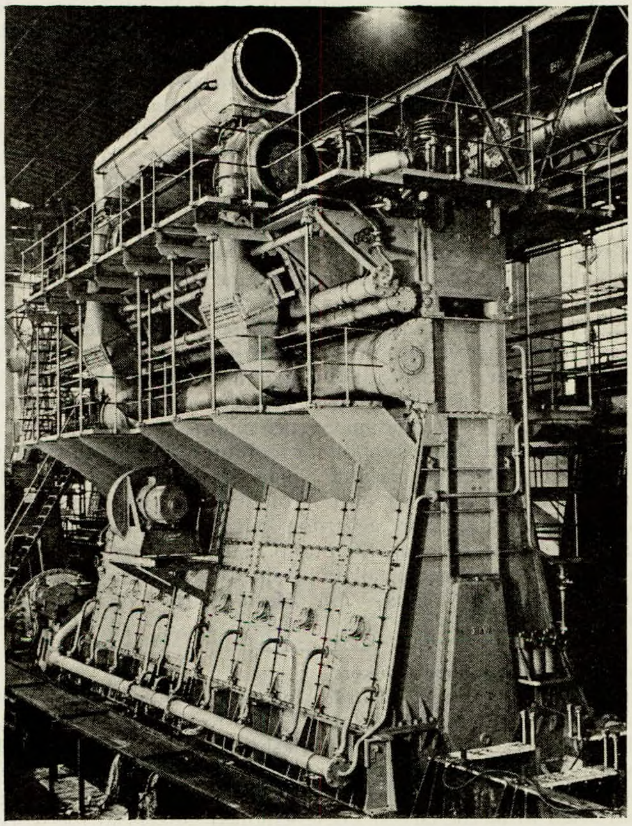
Mariner Class Dry Cargo Ships

The design of the *Mariner* calls for a dry cargo vessel substantially larger than previously built American ships. Along with the greater size is greater horsepower from her engines and a large cargo carrying capacity, both in dry cargo and refrigerated cargo. The *Mariners* are full scantling dry cargo ships, 560ft. 10in. long, 76ft. beam and 44ft. 6in. deep, with engine room and house amidships and a raised forecastle deck. The ships have seven cargo holds, five of which are double ganged, with heavy lift equipment at the forward ends of holds No. 4 and 6. They are transverse framed with solid floors under each frame. The ships are characterized by a raked stem, cruiser stern and a tapered stack. Kingposts instead of masts support the cargo handling equipment and the standing rigging. There are two complete steel decks, the main and second, while below the second deck is a third deck extending for a distance of approximately 265 feet from the bow. Forward of the forepeak bulkhead is a steel chain locker located between the main and third decks. A watertight steel flat is located under the second deck running from the afterpeak bulkhead to the stern of the ship parallel with the baseline. At a point approximately 335 feet from the bow and extending aft for 65 feet at the 26½ feet level above the base line is another partial deck and below this, beginning at the same point and extending aft for 177 feet is another partial deck. Both of these are cargo spaces. Transversely, the ship is divided by nine watertight bulkheads extending to the weather deck. Between the main and second decks, tonnage openings have been cut as required. Double bottoms begin at a point 35 feet from the bow and extend aft for a distance of 425 feet. They are subdivided to provide tank space for ballast and fuel oil.—*The Log, Vol. 48, March 1953; pp. 45-46.*

Cargo Liner with Supercharged Diesel Engine

Trials have recently been completed of the m.v. *Songkla*, built by Burmeister and Wain for the East Asiatic Company. This vessel is the first to be specially designed for propulsion by the new B. and W. two-stroke turbo-charged Diesel engine. The average speed attained was 18·84 knots, corresponding to the guaranteed speed of 16·8 knots on loaded trials. Compared with a 9-cylinder engine installation without turbo-chargers, developing 8,300 b.h.p. at 115 r.p.m., a saving of 30 per cent in weight and 20 per cent in length is obtained for the main engine, corresponding to a reduction in the length of the engine room by three frame spaces. A further reduction is obtained in the size of the starting air receivers, compressors, fresh and salt water cooling pumps; and a reduction in the power consumption corresponding to the reduction in the number of cylinders from 9 to 7 for the main engine. The *Songkla* has been built to the highest class of Lloyd's Register of Shipping, with dimensions corresponding to a mean draught of 27ft. 2in. Her main characteristics are as follows:—

Length b.p.	460 feet
Beam	62 feet
Depth to upper deck	38 feet
Depth to 2nd deck	29ft. 9in.
Draught (about)	27ft. 2in.
Deadweight (about)	10,100 tons
Loaded speed (about)	16·8 knots



View of main B. and W. engine

The ship is a single-screw motorship with two continuous decks, raked stem, and cruiser stern. Three bipod masts are fitted, and in addition there are two samson posts forward of the bridge. The hatchways are served by twelve 5-ton cargo derricks and two heavy derricks of 60 and 20 tons capacity respectively. At the after end of the boat deck there are two 3-ton deck cranes. The main propelling engine of the m.v. *Songkla* is a B. and W. direct reversible, single-acting, poppet valve, two-stroke, 7-cylinder crosshead engine with turbo-chargers. The engine has a cylinder diameter of 740 mm. and stroke of 1,600 mm., and is capable of developing 8,750 b.h.p. in normal continuous service, corresponding to 9,870 i.h.p. at 115 r.p.m., with an indicated mean pressure of 8 kg. per sq. cm. This is an increase of 35 per cent in b.h.p. compared to the non-turbo-charged engine of the same main dimensions and number of revolutions. A feature of the B. and W. supercharged engine is the system by which supercharging is achieved. Scavenging air is supplied solely by two centrifugal blowers, each direct-coupled to a turbine driven by exhaust gas from the engine. From the turbines the gas is led to an exhaust gas boiler which also serves as a silencer, and which is able to generate the same quantity of steam as for an unsupercharged engine of the same output. Air coolers are inserted between the blowers and the scavenging air receivers, and the scavenging air is cooled to a temperature of about 10 deg. C. above the temperature of the cooling water. In the unlikely event of both turbo-chargers breaking down simultaneously, a small electrically-driven emergency blower is installed, capable of giving the vessel about 60 per cent of normal speed. Apart from the fitting of the turbo-chargers and the omission of the chain-driven scavenging air blower the engine is of normal design.—*The Shipping World, Vol. 128, 15th April 1953; pp. 369-371.*

Steadiness of Route in Restricted Waters

It is well known that either because of the shallow depth or the presence of banks, the manœuvring behaviour of a ship in restricted waters differs completely from the manœuvring behaviour of the same ship in water of practically infinite width and depth. The problem has taken on greater importance because of the increase in size of ships which must pass through certain canals, in particular the Suez Canal. The Paris Model Basin has been carrying out tests on this project for the past fifteen months. The straight-line towing tests with yaw have demonstrated that the hydrodynamic forces on the hull are radically changed by the shallowing of the water and the presence of the banks. From these tests have come coefficients of magnification of the yawing force of the order of 10, comparing motion in the open sea and in a canal. The point of application of this force is shifted aft, and the moment about the centre of gravity is multiplied by approximately three. The forces on the rudder, when at an angle other than zero, are modified only a very little. It is deduced, therefore, that the efficacy or steering ability of the rudder is very much reduced compared to that when the ship is in open water. To this effect there is added the action of the bank, which produces a repelling force on the bow of the ship and an attracting force on the stern. These preliminary tests have indicated the great importance of the rapidity of laying the rudder. However, this can be studied only on a self-propelled model. The Paris Model Basin has set up a canal model of sufficiently large dimensions to enable self-propelled models to be tested. The first tests with this model showed that the phenomena were very similar to those which are observed in the full scale. However, the manœuvring behaviour of the model differed essentially from that of the ship in the following respects. The operator who steers the model is in a much less favourable position to notice the beginning of a swing than is the pilot who stands upon the bridge of the ship. Consequently, the yaw of the model is much greater than that of the full-scale ship. It appears that the behaviour of the model cannot be compared directly with that of the ship, but this comment does not eliminate the possibility of comparing two models and of classifying them from the standpoint of ease of manœuvring in an arrangement which should be the same as that of the actual ships. To carry the matter farther, it is necessary to define the parameters which may be measured on the model, selecting those whose value enables a firm conclusion to be formulated concerning the ease of manœuvring the ship. The action of the Institute of Research has, therefore, been directed toward the determination of the behaviour of actual ships as a means of enabling characteristic parameters to be selected and of defining desirable values for them. The difficulty with this is that it is evidently not possible to undertake a systematic test on a ship in transit in the (Suez) Canal. It is only possible, at the best, to make the most of such information as can be gathered in the course of a transit made under ordinary conditions. The first attempts showed that it was most difficult to take sufficiently detailed data to make it possible to reproduce the transit conditions completely. One test which involved an attempt to fix the position of the ship with reference to the banks by range-finder observations while simultaneous records were made of the heading and rudder angle, gave only a very doubtful reconstruction of the actual conditions. One other experimental difficulty has been encountered, having to do with the quite different manner in which individual ships are handled. As an example, the number of rudder shifts in the course of a single hour of running in straight sections of the Suez Canal are given for two ships:—

	First ship	Second ship
Shifts of rudder angle less than 15 degrees	8	122
Shifts of rudder angle from 15 to 30 degrees	63	21
Shifts of rudder angle exceeding 30 degrees	3	0

At the present time it is not possible to determine whether these differences arise from individual styles in handling the ship on the part of the pilots or whether they can be related to certain characteristics of the ships themselves. On this point one is led to ask whether the greater frequency of rudder shifts at small rudder angles on the second ship do not really stem from the fact that it answers its rudder much better and that an incipient yaw can be checked much more quickly and with a smaller rudder angle. Finally, the Institute of Research has requested the Paris Model Basin to test a ship which has made a large number of Suez Canal transits and which is considered to be a ship easy to handle in the Canal.—*Collected French Papers on the Stability of Route of Ships at Sea. Issued by The Institute of Research in Naval Construction, Paris. Navy Department, The David W. Taylor Model Basin, Translation 246, January 1953; pp. 47–49.*

Anti-fouling Paints for Aluminium Alloys

The development of technique in anti-fouling aluminium ship bottoms is traced. Early attempts to use paints containing copper and mercury compounds failed because both corroded the aluminium. Paints containing organic compounds such as 10-chloro 5:10 dihydrophenarsazine (D.M.) may be used, but compounds developed so far, while not attacking aluminium, have only limited anti-fouling power. In the interim of the development of better organics, an excellent procedure is the use of multiple layers of zinc chromate primer protecting the aluminium, and overlaid with, say, cuprous oxide in a vinylite type medium. As a result of work at the Naval Research Laboratory, Washington, anti-fouling paint, specified in 1944, contained copper resinate as a suspension agent and mercurous chloride as anti-fouling agent. The varnish medium was an alkyd resin and a high-viscosity chlorinated rubber to give films with good flexibility. The specification was amended three times. Recently the U.S. Navy has changed over completely to paints based on vinyl resins for coating aluminium PT boat hulls and other craft. The paint system specified consists of a wash primer based on polyvinyl butyral resin and pigmented with zinc tetroxychromate followed by an undercoat comprising a polyvinyl copolymer and resin, with cuprous oxide as the sole pigment and poison. As a result of trials by the Admiralty begun in 1949, it has been shown that one coat of etch primer or pretreatment with A.C.P. Deoxidrine followed by one coat each of Admar anti-corrosive (white lead, basic lead sulphate and iron oxide in a linseed stand oil phenol-formaldehyde varnish medium) and one coat of a cuprous oxide plasticized resin varnish type anti-fouling paint allowed only very little corrosion of the aluminium alloy after two years' exposure. The paint coating remained almost intact when a red lead Vinylite undercoat applied over an etch primer was substituted for the Admar anti-corrosive paint. Specimens in which the anti-fouling paint containing cuprous oxide was applied direct to the aluminium alloy (AW6) showed severe corrosion after three months' immersion, which became increasingly worse up to two and a half years. The above paint systems were compared with a proprietary system as control, in which the anti-fouling paint contained organic compounds as the sole toxic constituents. In each case, except where the cuprous oxide anti-fouling paint was applied direct to the aluminium alloy, the anti-fouling performance and corrosion resistance of the systems under trial were superior to those of the control. It is concluded that paints containing organic compounds as the only poison constituents have so far fallen short of the required standard of anti-fouling efficiency for aluminium alloy vessels. The most satisfactory coatings to date consist of corrosion inhibiting primers and undercoats of the zinc chromate type followed by an anti-fouling paint containing sufficient copper compounds as the poison to give complete anti-fouling protection for at least six months. Further developments are to be expected as more than one authority has work in hand to produce better anti-fouling paints for this requirement; further organic poisons are being investigated, and other types of medium are under consideration.—*Paint Manufacturer, Vol. 23, 5th, 10th, 29th January 1953. Abstract in Light Metals Bulletin, Vol. 15, 20th March 1953; pp. 180-181.*

Salvage Vessel with Hydraulic Auxiliary Machinery

An order has been received by the A/S Stord, Norway, for a salvage ship to be propelled by a Wichmann four-cylinder two-stroke engine driving a variable pitch propeller at 325 r.p.m. The vessel is unusual in several respects, having a combination of hydraulic and electrical auxiliaries. She is classed to the requirements of Det Norske Veritas and is strengthened for navigation in ice. The length is 31·635 metres o.a. (104ft.), the breadth being 6·9 metres (22·75ft.) and the depth 3·45 metres (11·4ft.). On deck are a windlass, a winch for general lifting purposes and a towing winch, all of the hydraulic type. There are two pumps in the engine room for the supply of oil under pressure, one driven from the main engine and the other coupled to an electric motor. Current for the electric auxiliaries is supplied by two Pelapone 60 b.h.p. Diesel engines, each coupled to a Thrige alternating current, three-phase 37-kW., 220 volt generator. The speed is 1,000 r.p.m. and there is also a 6-kW. auxiliary set. The salvage plant includes an Atlas compressor with a capacity of 4 cubic metres per minute, the pressure being 7 kg./cm.^{2}, and there is a diving compressor with a capacity of 2 cubic metres per minute. The salvage pump is of the centrifugal type with an output of 200 tons per hour. The auxiliary plant includes a 20-ton bilge pump, a 25-ton fire pump, a 3-ton fuel transfer pump, a starting air compressor and a water pressure installation, also a refrigerating machine for the ship's provisions.—*The Motor Ship, Vol. 33, March 1953; p. 519.*

Largest Diesel-engined Tankers

Six ships are now on order for various Norwegian owners to the new standard design for a 32,250-tons deadweight tanker which has been developed by Eriksbergs mek. Verkstads A/B, Gothenburg. Like all the other vessels to be built at this yard, these tankers will have Diesel propulsion, and are likely when completed to be the largest Diesel-engined tankers in the world. Their nearest rivals are the French-owned and built *Bérénice* and *Bethsabée*, of some 31,200 tons deadweight, which are already in service. Like the French ships, the Eriksberg tankers will have two Diesel engines direct-coupled to twin shafts, the engines being of Eriksberg-B. and W. type. The principal particulars of the Eriksberg tankers are given below, and those of the *Bérénice* are added for comparison. It is noticeable that while the French ships have been designed to have the greatest draught permissible at present for passage through the Suez Canal, Eriksbergs have adopted a much shallower section, and the depth of the larger ships is actually the smaller:—

	Eriksberg design	*Bérénice*
Length b.p. ...	625ft. 0in.	619ft. 1in.
Breadth moulded...	86ft. 0in.	85ft. 0in.
Depth to main deck	45ft. 9in.	46ft. 5in.
Loaded draught (summer) ...	34ft. 3in.	35ft. 3in.
Tank capacity ...	1,500,000 cu. ft.	1,400,000 cu. ft.
Fuel capacity ...	2,650 tons	2,950 tons
I.H.P.	13,800	13,100

The Eriksberg design specifies a ship qualifying for the highest class of Det Norske Veritas, longitudinally framed and with corrugated bulkheads. An unusual feature is the adoption of a straight stem: this choice has presumably been made in order to keep down the overall length as far as possible. The hull shape is full, as is common with modern large tankers and ore carriers, the block coefficient being 0·79. There are two longitudinal bulkheads, dividing the eleven tanks into centre and wing sections. There are two main pump rooms, one immediately forward of the engine room and the other between Nos. 3 and 4 tanks, while an auxiliary pump room for ballast and fuel oil pumps is situated just abaft the forward collision bulkhead. There are four cargo pumps, each with a capacity of 750 tons per hour. The two in the forward pump room are steam driven, and the other two are electrically-driven centrifugal pumps, the electric motors being situated in the engine room and driving the pumps through the bulkhead. The main cargo piping is of 14-in. diameter, with 10-in. branches to the main and wing tanks. The main propelling machinery will consist of two Eriksberg-B. and W. Diesel engines of two-stroke single-acting type. Each has six cylinders with a diameter of 740 mm. and stroke of 1,600 mm., and the combined output is 13,800 i.h.p. at 115 r.p.m. Auxiliary machinery is largely steam driven, and three oil-fired boilers are included in the design. Electric power is supplied by two 330-kW. Diesel generator sets, and there is also a steam dynamo of 75 kW. capacity.—*The Shipping World, Vol. 128, 18th March 1953; p. 281.*

New Cargo Vessels with Passenger Accommodation

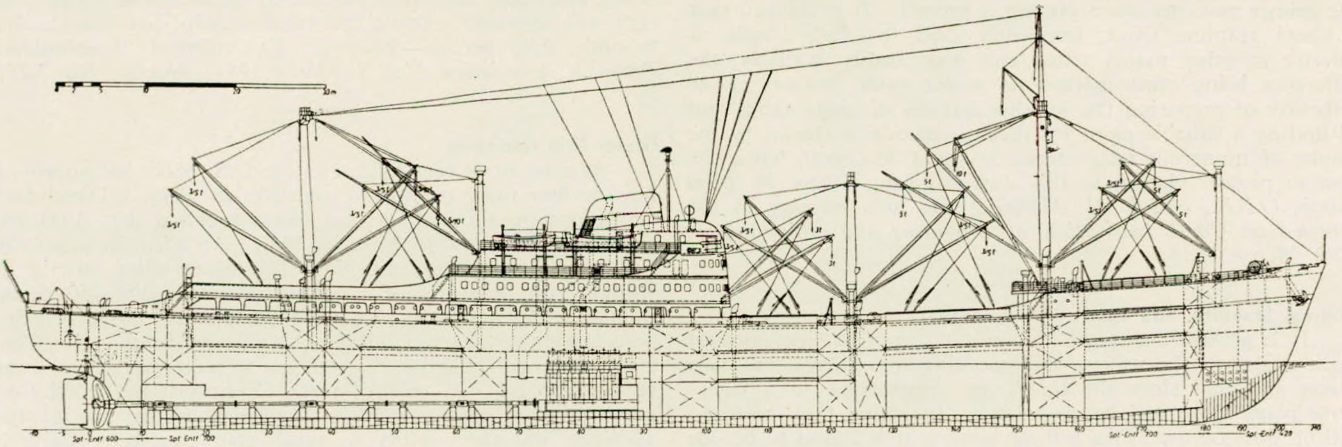
The second series of *Santa* vessels of The Hamburg-Südamerikarische Dampfschiffahrts-Gesellschaft Eggert and Amsinck KG, is now building at the yard of the Howaldtswerke Hamburg A.G. These vessels, of which the first, *Santa Teresa,* was recently placed in service, have the following principal dimensions:—Length, o.a., 146·04 m.; Length b.p., 135·14 m.; Breadth moulded, 18·6 m.; Draught, 8·61 m.; Deadweight, 11,710 tons; Displacement, 16,910 tons; Gross registered tons, 8,996; Net registered tons, 6,682. There is accommodation for 28 passengers in twelve single berth and eight double berth first class cabins. The main propulsion plant consists of a six-cylinder single-acting two-stroke cycle crosshead Diesel engine of M.A.N. design, built by the Howaldt yard under licence. At 125 r.p.m. the engine develops 4,000 b.h.p., the speed of the vessel being 13 knots in the fully laden condition. Cylinder diameter is 700 mm. and stroke is 1,200 mm., with a m.e.p. of 6·2 kg. per sq. cm. The bronze propeller is of 4·85 m. diameter. There are two auxiliary Diesel sets generating 130 kW. and 250 kW. respectively. An emergency Diesel set of 25 kW. is also provided.—*Hansa, Vol. 90, 28th February 1953; pp. 343-348.*

Dimensionless Frequency Parameter in Unsteady Flows

This report presents some theoretical results in connexion with the unsteady flow research programme at the David Taylor Model Basin. The complex and relatively unknown field of time-dependent hydrodynamic phenomena is approached from a general point of view. Only a few special flows are discussed emphasizing the diversity of unsteady flow problems. Since time effect occurs only in the acceleration term of the momentum (or Napier-Stokes) equation, an analysis of the two types of acceleration is presented in detail. A dimensionless ratio of the local and convective accelerations is introduced. It is shown that with the magnitude of this ratio, the unsteadiness of the flow can be described and characterized. Previously published results on hydrodynamic impact represent one limiting case for the above "measure of unsteadiness" ($S \rightarrow \infty$). Steady (time-independent flows) are associated with zero value of this measure ($S = 0$). A detailed report on flows with intermediate values ($0 < S < \infty$) is under preparation. It will be published in the near future in connexion with recent oscillator experiments. A short discussion of accelerationless flows is included. —*V. G. Szebehely, Navy Department, The David Taylor Model Basin, Report 833, November 1952.*

B.S.R.A. Resistance Experiments on the *Lucy Ashton*

Sir Maurice Denny's paper, which was read before the Institution of Naval Architects at the International Conference of Naval Architects and Marine Engineers 1951, dealt with the details of the experimental equipment and the conduct of the ship trials of the *Lucy Ashton,* carried out by the British Shipbuilding Research Association. In addition, some of the full-scale resistance results were given in view of their interest and importance and in order to demonstrate the quality and accuracy of the experimental observations. The present paper is complementary and deals primarily with the correlation of the measured full-scale resistance of the ship with the results of tests carried out in an experimental tank on a series of six geometrically similar models ranging in length from 9 to 30 feet. The translation of the model results to the full scale has been effected in the usual manner, using Froude's skin-friction coefficients, and also by several modern skin-friction formulations. The scope is restricted to correlation on the basis of resistance for the four naked hull conditions of the ship, which included tests with two different paint surfaces in conjunction with the effect of fairing the seams of the shell plating. An account is given of the various corrections which had to be applied to the ship results and also the analysis of the comprehensive measurements of hull surface roughness made on the ship. Full details of both ship and model results are given in appendices.—*Paper by J. F. C. Conn, H. Lackenby and W. P. Walker, read at the meeting of the Institution of Naval Architects in London, 25th March 1953.*



Santa Teresa *type vessel*

Skin Friction Determination Using Wall-sided Models

Two fundamental assumptions made by Froude can be stated as follows: (a) Skin friction and wave resistance are separate entities which do not mutually affect each other; (b) skin resistance could only be determined by plank experiments, and residuary resistance (chiefly wave resistance) could only be determined by subtracting the skin resistance from the total resistance. The writer has of necessity retained assumption (a) but has reversed assumption (b). An attempt has been made to produce a friction line, that is to say, a line corresponding to zero beam, by subtracting the calculated wave resistance from the total resistance. The apparatus used was a specially designed gravity dynamometer, based on the simple form described by the writer in a previous paper. A line free from edge effect under fairly full beam conditions has been obtained by tests on a family of three models, all of 15-in. draught, and having beams $1\frac{1}{2}$ inches, $2\frac{1}{4}$ inches, and 3 inches respectively. Care was taken to ensure turbulent flow. Finally the wave profiles of the models were recorded photographically and compared with the calculated profiles.—*Paper by R. T. Shields, read at a meeting of the Institution of Naval Architects, 27th March 1953.*

Preservation of Oil Tanker Hulls

The problem of preserving the hulls of oil tankers, including the interiors of cargo compartments, is discussed and the measures already adopted or being considered to reduce wastage are described. The effect of corroded surfaces in contact with the sea upon a ship's speed is referred to and an estimate given of loss in the ship's earning capacity from this cause. Protection of the outside of the hull is shown to depend for its success as much upon initial preparation of the surface as upon paint. Experiments with flame cleaning are described and the advantages of this method of surface preparation for large areas treated *in situ* illustrated, while the beneficial effect of applying the primer paint to warm plating is proved. It is claimed that red-lead graphite paint, frequently used for ships' hulls, is inferior to other paints tested and now readily available, the difference being most marked on under-water surfaces. The difficulty of preparing the interior surfaces of cargo tanks and of finding a suitable paint for their protection is shown by the results of numerous tests carried out and described, while the cost to protect plating in this way is given.—*Paper by John Lamb, O.B.E., and E. V. Mathias, read at a meeting of the North-East Coast Institution of Engineers and Shipbuilders, 30th March 1953.*

Rolling, Stability, and Safety in a Seaway

It is generally assumed that when a ship is proceeding at right-angles to the waves, only pitching movements will take place, and that since the forces are symmetrical in a transverse plane there will be no rolling. In actual fact, however, rolling does occur. This rolling movement is found to be due to variations in the stability of the vessel, since the metacentric height varies owing to the movement of the water. Theoretical analysis shows that the central upright position of the ship is unstable when the frequency of the variation in metacentric height coincides with the natural frequency of the ship or one of its harmonics. An experimental investigation in which a model was equipped with an eccentric weight rotating in its longitudinal centre plane confirmed this result. The experimental results are analysed and discussed at some length. The behaviour of a ship in a sea which makes an acute angle with the course is briefly considered. The danger that a ship may capsize in a heavy sea is discussed, and it is shown that the righting moment is considerably reduced when the ship is on the crest of a wave; it is considered that if there is a following sea, and the relative speed of the ship with respect to the wave is small, there is a danger that the ship may be carried on the crest of the wave for a length of time sufficient for capsizing to take place. A mathematical analysis of the rotary motions of a ship, namely yawing, rolling, and pitching, shows that they are interrelated, and that one can give rise to the others.—*O. Grim, Hamburg Shipbuilding Experiment Station, Memorandom No. 269. Journal, The British Shipbuilding Research Association, Vol. 8, March 1953; Abstract No. 7,229.*

Corrosion Resistant Steel for Tankers

A considerable amount of damage is done to the interior of oil cargo tanks by the corrosive substances present in the oil, and by rusting caused by the carriage of sea water as ballast. L. F. Dert in a recent article in "Schip en Werf" suggests that the time and expense involved in carrying out the repairs made necessary by this corrosion can be considerably reduced by employing a corrosion-resistant steel containing chromium and aluminium as alloying agents. This steel has the further advantage that it has a higher tensile strength and elastic limit than the conventional shipbuilding structural steels, so that the weight of the hull can be reduced without impairing the strength. As an example, the cost of building a 17,000-ton d.w. tanker in which the cargo tanks are built of this steel is compared with the cost of building the ship entirely from the normal structural steel. The cost of using the new steel is estimated as about $12\frac{1}{2}$ per cent greater. The reduction in the weight of the hull will permit an increase in deadweight of $2\frac{1}{2}$ per cent, which will increase the annual profit by 6 to 8 per cent, according to the freight rates obtained. The main advantage of using this corrosion-resistant steel is, however, apparent when the ship has been in service for, say, twelve years, when about 65 per cent of the cargo tanks of the vessel built of structural steel will require replacement, against 5 per cent in the vessel built of corrosion-resistant steel. At the conclusion of these repairs, the total cost of the tanker built of special steel is found to be about 15 per cent less than that of the tanker built of normal steel. The normal tanker requires further considerable repairs when it has been in service a total of 20-25 years, and these are often considered uneconomic. This new steel will, however, extend the vessel's useful life considerably beyond this period.—*Journal. The British Shipbuilding Research Association, Vol. 8, March 1953; Abstract No. 7,251.*

Diesel Fuel Additives

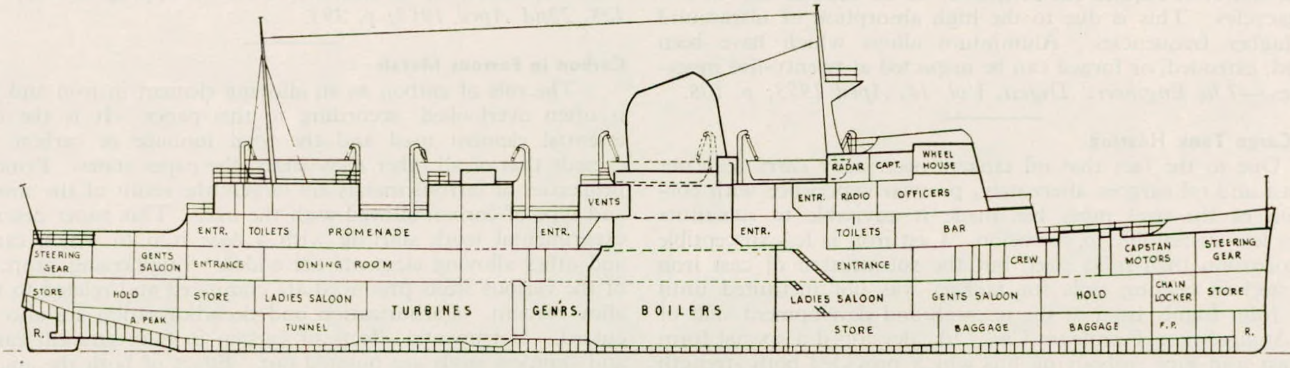
As a result of year-long tests the U.S. Navy has approved, for the first time, the use of additives in Class I Diesel fuel. Consumption of Diesel fuel has grown fourfold since 1941 and authorities foresee a doubling by 1960. An adequate supply of fuel is a problem; for the Navy heretofore relied strictly on straight run distillates of a certain cetane quality—50 cetane number as a minimum. No additives were permitted to bring up a distillate of a lower number. Navy specifications for Class I fuel imposed difficulties for refiners to meet the general competing demand. In view of these conditions, the Ethyl Corporation conducted a four-year research on the use of fuel additives and their ability to raise cetane number. Early in 1951 the U.S. Navy sponsored a test programme involving the additive, or ignition improver, to straight run distillates having a lower cetane number. Use was made of Ethyl Corporation's additive in conjunction with Venezuelan straight run Diesel fuel of 46·5 cetane number. Between 0·18 per cent and 0·20 per cent by volume of the ignition, improver was added to each gallon to bring the average cetane quality to 53·5. Prime test criterion was that the added cetane quality should equal that in the natural state; that this quality should be stable and be retained in storage; and it must not create any harmful engine operating conditions (wear, cleanliness, operation, smoke, noise, odour): nor must it be a health hazard. A squadron of twelve submarines tested the fuel with the additive and individual operating time ran upwards of 1,700 engine-hours on the many weeks' test. None of the engines was preconditioned for the fuel. At the same time the Naval Engineering Experiment Station conducted fuel analyses which found the treated fuel to be stable.—*Motorship (New York), Vol. 38, March 1953; p. 38.*

New French Cross-Channel Vessel

A new French passenger vessel, the twin-screw steamship *Lisieux,* has joined the small fleet of vessels which maintain the cross-Channel service between Newhaven and Dieppe. The *Lisieux* has been built at the Granville yard at Le Havre of Forges et Chantiers de la Mediterranée. In general design and appearance the *Lisieux* is a miniature version of the *Côte d'Azur,* which was completed two years ago for the service between Folkestone and Calais, and she can be employed as a relief ship on this route. Her principal particulars are as follows:—

Length o.a.	312ft. 7in.
Length b.p.	308ft. 4in.
Breadth o.a.	45ft. 0in.
Depth moulded to B deck ...	24ft. 0in.
Gross tonnage	2,800 tons
Displacement at full load ...	2,221 tons
Horsepower	22,000 s.h.p.
Speed	24 knots

Conditions at Newhaven place limits on both the length and the draught of ships on the Newhaven-Dieppe service, while the schedule demands high speeds. In the *Lisieux* these conditions have been combined with adequate accommodation space by the use of unusual lines. A very long and fine entry blends into a flat run aft, the latter to counteract the inevitable tendency to squatting at high speeds with so short a length. Below the waterline, therefore, the point of maximum beam is well abaft amidships. Above it, however, the lines are of normal full shape, and the transition has been achieved by the use of a prominent knuckle forward. The hull is constructed of Siemens-Martin steel, and partly welded. Light alloys have been used for the deck houses and the funnel, and the eight boats are also of light alloy. Four of the boats are carried on the after deckhouse, with the chocks fitted on projecting sponsons, and these boats can be moved down to the deck below in winter. As in the *Côte d'Azure,* the funnel is of the FCM-Valensi "Strombos" type. The main propelling machinery consists of a twin-screw arrangement of Parsons geared-steam turbines. With the limitation on draught, small propellers have to be used (9·18ft. diameter) turning at the high speed of 400 r.p.m. Single reduction gearing is therefore employed. Two boilers of the FCM 47/60 type are fitted, delivering a total of 132,000lb. of superheated steam per hour at 425lb. per sq. in. and 705 deg. F. The FCM 47/60 is the latest type of boiler developed by Forges et Chantiers de la Mediterranée, and it is claimed that steam can be raised in thirty minutes. A bow rudder is fitted, and this and the rudder aft are controlled by hydraulic telemotors.—*The Shipping World, Vol. 128, 8th April 1953; pp. 353-354.*



Profile of the Lisieux

Risks in Industrial Radiography

During the course of the discussion on Mr. E. J. Duffy's paper, "Radiography in a Clyde Shipyard", recently presented to the Institution of Engineers and Shipbuilders in Scotland, Dr. J. M. A. Lenihan, M.Sc., Ph.D., stated that while the author had stressed the importance of protective measures for X-ray and gamma-ray workers, there is general apathy in this matter. It should be realized that a few minutes or even a few seconds may be sufficient for an accidental overexposure, and he wondered if any attempt was made to measure by means of films or pocket ionization chambers the radiation received by operators. The author in his reply suggested that it is the responsibility of those in charge of the radiographic department to familiarize themselves with the various necessary precautions to be taken to avoid accidental exposure. In addition, steps should be taken to measure the amount of radiation being received by the operator, and firms not possessing a pocket ionization chamber should be encouraged to take advantage of the services provided by the National Physical Laboratory, Teddington. This service may be employed either continuously or intermittently, and the author recommends the former at least during the first few months in order to check that the operator is not being unwittingly exposed to radiation.—*Shipbuilding and Shipping Record, Vol. 81, 12th March 1953; pp. 327-328.*

Measuring Wall Thickness of Long Tube

An illustrated description is given of an apparatus for the direct measurement of the wall thickness of a tube at a point along its length which is not easily accessible. The apparatus consists of a beam of channel section carrying an alignment telescope at one end and a measuring anvil at the other. A measuring stylus is held directly over the anvil by a thin-walled small-diameter support tube, the other end of which is connected to the beam by a simple form of universal joint. An illuminated target is situated near the plane of the measuring stylus and displacements of the stylus and target relative to the measuring anvil can be viewed through the bore of the stylus-support tube by the alignment telescope, and measured by an optical micrometer incorporated in the telescope. The instrument is first standardized by inserting a slip gauge between the stylus and the anvil and aligning the telescope on to the target in this position. Then the stylus and target are inserted into the tube through an open end, the tube wall thus separating the stylus and anvil in place of the slip gauge. The telescope is again adjusted, and the difference between the new and the standard reading gives the difference between the thickness of the tube and the gauge. The optical micrometer can indicate target displacements up to 0·1 inch in units of ± 0·0005 inch, and the accuracy of the instruments is within ± 0·0005 inch. The apparatus is not restricted to the wall thicknesses of tubes, but with some modifications might be adapted to obtain measurements at inaccessible positions on many other types of products.—*H. R. Davenport, Ministry of Supply, A.I.D. Development Report No. Metro/4, December 1951. Journal, The British Shipbuilding Research Association, Vol. 8, February 1953; Abstract No. 7,197.*

New Ultrasonic Inspection Method

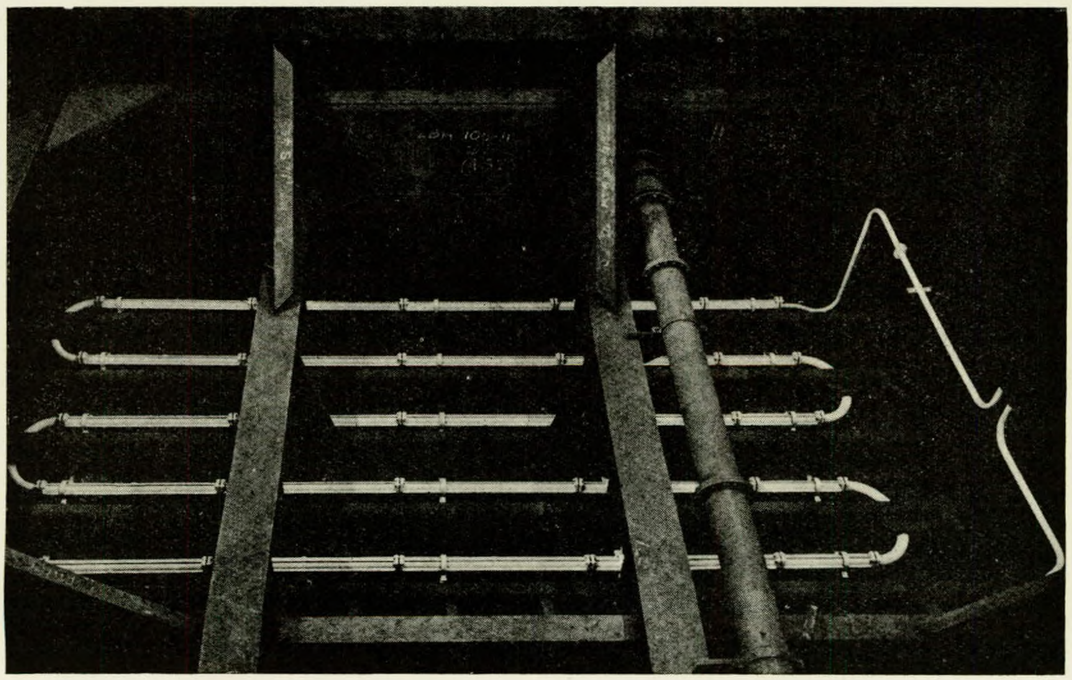
In the contact-scanning method employed in the ultrasonic examination of large metal objects, it is usually necessary to machine or grind the surface of the metal before it can be inspected. With a new system of inspection, the metal is immersed in rust-inhibited water, and the crystal scans the metal without being in contact with it. The several inches of water between the crystal and the metal make it possible for the crystal motion to be mechanized without concern for the small contour changes in the metal under inspection. Also, a unit operating according to the new system should be operated in conjunction with an echo type flaw detector to give the echo height standardization. The use of the mechanized crystal scanning allows electronic circuits to calculate the position of the crystal relative to the object being inspected. Once this position is known in electrical terms, the data can be fed to a television-type display tube. Changes in the horizontal position of the crystal cause similar changes in the horizonal position of the beam in the cathode ray television tube. The vertical position of the cathode ray tube beam is determined by the elapsed time between sending the ultrasound pulse and the received echoes from the metal object being inspected. Ultrasound frequencies of from five to twenty-five megacycles are used. The higher frequencies are claimed to give the best resolving power, because they will permit the distinguishing of flaws which lie close to the surface. Some metals, such as low carbon steel, however, require the frequency to be kept as low as five megacycles. This is due to the high absorption of ultrasound of higher frequencies. Aluminium alloys which have been rolled, extruded, or forged can be inspected at twenty-five megacycles.—*The Engineers' Digest, Vol. 14, April 1953; p. 109.*

Oil Cargo Tank Heating

Due to the fact that oil tankers necessarily carry seawater ballast and oil cargoes alternately, postwar experience with corrosion of the steel pipes has made it advisable to substitute pipes less susceptible to corrosion. Cast iron is less susceptible to corrosion than mild steel, but the substitution of cast iron for steel in heating coils for tankers was not instituted until Mr. John Lamb, head of the research and development staff of the Anglo-Saxon Petroleum Co., Ltd., developed a special form of cast iron pipe embodying fins which provided both strength and additional heating area. The first ship to be fitted with cast iron heating coils was the *Amastra,* in 1946. After six years the installation is still in use, and shows no sign of pitting or corrosion. The system was patented and in 1949 Colvin-Smith, Ltd., were entrusted with the sole licence in the United Kingdom for further development and installation. The pipes are usually supplied in 6-ft. lengths with four fins 1 inch high, spaced equidistantly round the external surface of the pipe and running longitudinally between the flanges, which are mated and connected by four bolts. The patent cruciform section ensures efficient heat transference and gives added strength to the pipe. The heating surface of cast iron finned pipe is 1·66 sq. ft. per linear foot, as compared with the figure of 1·04 sq. ft. per linear foot for plain round pipe. The owners of the *Merchant Baron* specified the following heating surfaces: Centre tanks 1 sq. ft. per 120 cu. ft. tank capacity and 1 sq. ft. per tank capacity for the wing tanks. Using finned cast iron pipe, a total length of 4,820 feet is required as compared with a total length of 12,950 feet of plain 2-inch bore steel pipe. The comparative weights are 70 tons for the cast iron installation and 49 tons for a steel pipe installation. The patented cast iron pipes are supported at intervals of not more than 8ft. by $2\frac{1}{2}$in. × 2in. × $\frac{3}{8}$in. angle welded between the longitudinals and secured to these angles by 2in. × $\frac{3}{8}$in. steel clips. The pipes are arranged at not less than 4 inch distance from the shell of the ship, and the whole installation is free to move with expansion or contraction.—*The Shipping World, Vol. 128, 22nd April 1953; p. 393.*

Carbon in Ferrous Metals

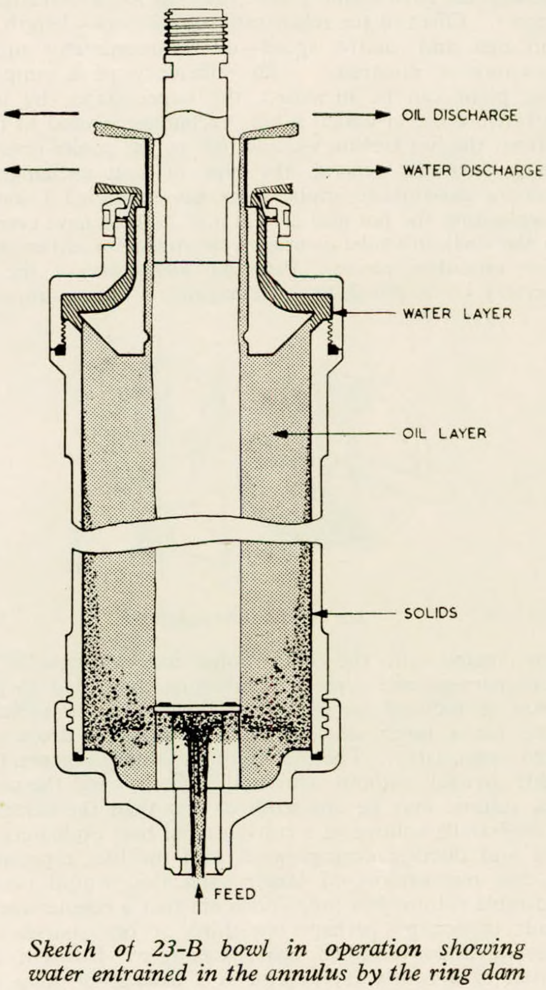
The rôle of carbon as an alloying element in iron and steel is often overlooked, according to this paper. It is the most essential element used and the total tonnage of carbon steel exceeds that of all other alloy steels, the paper states. Principal properties of ferrous metals are largely the result of the amount and type of carbon alloyed with the iron. This paper describes experimental work starting with a base iron to which carbon and other alloying elements are added. Engineering properties of the various steels produced are compared and related to their alloy content. Carburization and decarburization are also discussed. Detrimental effects of carbon in both straight carbon and stainless steels are pointed out. Effect of both the amount and the form of carbon in cast irons is shown experimentally in this paper. The rôle of carbon in the production of the



Finned cast iron heating coils in the tanker Merchant Baron nodular cast irons, which are new engineering materials with unique properties, is also shown. In conclusion, the paper states that carbon increases the strength and hardness in steels, particularly after heat-treatment. The ultimate possible strength and hardness is largely determined by the amount of carbon present. Tensile strengths of 50,000-350,000lb. per sq. in. can be obtained in steels with a carbon variation of less than 1 per cent. No other alloying element gives results comparable to carbon in such small amounts. In cast irons the type of carbon largely determines the properties of the metal. Nodular iron with its carbon in spherular form has the best properties of any of the cast irons. The important rôle of carbon in ferrous metals should be considered when designing machines or structures.—*Paper by H. K. Ihrig and J. T. Jarman, read at the 1952 A.S.M.E. Annual Meeting; Paper No. 52-A-81.*

Single Pass Separator for Heavy Fuels

The rapidity with which the practice of burning heavy residual grades of fuel in motorships has spread, is one of the most significant features of postwar marine engineering. Due to their higher specific gravity, the purification of heavy boiler grades is a more difficult proposition than that of distillate fuels. Standard separators can be used for these duties, but it is generally found necessary to operate at the highest permissible temperature, with a through-put considerably less than that when centrifuging fuel of lighter specific gravity. Furthermore, the treatment is generally divided into two separate stages, viz., centrifuging to remove water and the bulk of the solids, and a subsequent clarifying stage to extract the finest solids. Used thus, separator equipment gives satisfactory performance, but running costs are naturally raised by the necessity for duplicate machines. Sharples Centrifuges, Ltd., have developed a one-pass bowl within which the centrifugal treatment of heavy boiler oils is completed in one pass through a single machine. Extensive tests in the purification of heavy oils with viscosities of up to 3,500 seconds Redwood I at 100 deg. F. have been carried out with complete success. The particular feature of the new bowl is that it can be fitted into many existing machines and requires only modified covers with trapped discharge spouts to accommodate the somewhat larger bowl diameter. The centrifuging process is completed in one stage. There is a small annular space, having a greater diameter than the main body, at the upper end of the bowl; a water layer accumulates in this space, thus leaving the main body virtually filled with oil. The normal type of bowl has a substantial water layer throughout its length, which naturally reduces its effective volume and efficiency as a clarifier. The other features of the Sharples separator remain. In the new type 23-B bowl, the discharge screws which regulate the height of the water layer, and the undrilled screws of the clarifier are replaced by a ring dam which can be adjusted to suit the gravity of the fuel to be treated. If oil is found to come over with the water, the effective ring dam must be raised by fitting one of a lower number and conversely.—*The Marine Engineer and Naval Architect, Vol. 78, April 1953; pp. 142-144.*



Sketch of 23-B bowl in operation showing water entrained in the annulus by the ring dam

Recent Developments in Ship Plate

The author points out that the time is not yet ripe for the general application of an acceptance test for the notch ductility of ship plates. Apart from the difficulties of an agreed form of test and interpretation, there is the uncertainty about the level of notch ductility which would be adequate for the purpose, without being extravagant, for it must always be borne in mind that notch ductility, like any other property, has to be paid for. However, there are structural mild steels available which provide exceptional notch ductility and which are supplied to a specification including a notch bar acceptance test. One of these steels, known as Coltuf "28", which was developed by the author, has been used in the construction of large all-welded tankers by Kockums of Sweden. The particulars of Coltuf "28" are as follows:—

CHEMICAL ANALYSIS

	C per cent	Si per cent	S per cent	P per cent	Mn per cent	Mn/C ratio
Specification	0·16 max.	0·30 max.	0·04 max.	0·04 max.	1·3 max.	—
Typical plate	0·40	0·15	0·03	0·025	1·0	7

MECHANICAL PROPERTIES

U.T.S.	28/32 tons per sq. in.
Yield point	17 tons per sq. in. minimum
Elongation	25 per cent on 8in. gauge length
Charpy V-notch	45ft./lb. minimum at − 10 deg. C.

The steel was developed primarily to meet the demands of chemical engineers for a fully weldable structural steel for use at sub-zero temperatures. In spite of its relatively high cost, however, the above mentioned application to shipbuilding in Sweden is evidently an economical proposition, since its use is confined to the more vulnerable parts of the vessel amidships, including the sheerstrake, the bilge strake, and the deck stringer. —*Paper by W. Barr, read at a meeting of the Belfast Association of Engineers, 25th March 1953.*

Hardfacing by Welding

Hardfacing is carried out by welding and all the available methods of welding are used. In addition, a special method termed "braze-welding" has been developed to enable a deposit of alloy to be laid down with little, if any, mixing with the base metal. *Metallic arc deposition:* To ensure maximum hardness in the deposit the penetration into the base metal should be kept to a minimum by using a short arc and low current. The first run is softened most by dilution with the base metal and the third layer is usually pure and undiluted. This is particularly important when depositing stainless and non-ferrous alloys. A cleaner and smoother deposit results from connecting the rods to the positive pole when using direct current. All rods used for arc depositing are flux-coated to give deposits free from flaws. Rapid rates of deposition can be obtained by hand and hardfacing can be carried out with automatic welding heads in the same way as ordinary welding. *Carbon arc deposition:* There is less penetration by this process than with the metallic arc and in some respects, therefore, the process is better. *Atomic hydrogen and argon arc deposition:* Deposits are laid down with good adhesion and only little more mixing with the base metal than with the oxy-acetylene brazewelding process.—*M. Riddohough, Journal of the Junior Institution of Engineers, Vol. 63, April 1953; pp. 213-216.*

Progress in Marine Electrical Installations

Owing to the rigid electrical requirements of Lloyd's Register and other classification societies the use of electric power in tankers has in the past been chiefly confined to lighting, ventilation, galleys and pantry gear, domestic services and supplies for wireless and other essential navigational circuits. The installations for the smaller tankers are all somewhat similar and very simple in their layout. They comprise generally, three generators—two steam engine-driven and one Diesel engine-driven, one being standby, the size of plant varying between 40, 75 and 100 kW. capacity each according to connected load requirements. The dynamos are arranged for parallel operation, each feeding into a common set of bus bars and controlled by triple pole circuit breaker (d.p. and equalizer) with overload and reverse current trips. The main switchboard is of the open type, comprising polished sindanyo panels and angle iron frame, with d.p. circuit breakers for amidship masterboard and shore supply and d.p. switches and fuses for supply to the various services, i.e. engine room and boiler room fans, deck ventilation, galley equipment, refrigerating equipment, domestic machines, laundry, lighting, wireless and radar, etc. In recent years, however, a considerable expansion has taken place in the connected electrical load for tanker installations; this particularly applies to larger vessels which are now being built—18,000-32,000 tons d.w., and the change is primarily due both to the advent of the larger vessel and the adoption of electric power for engine room auxiliaries, and to recent changes in Classification Societies' Rules permitting increase in supply voltage for these installations to the following limits:—

D.C. systems: Power, heating and cooking 230 volts
Lighting, cabin fans, etc.... 110 volts
A.C. systems: Power—250 volts single-phase or 440 volts three-phase
Heating and cooking circuits 250 volts
Lighting, cabin fans, etc.... 115 volts

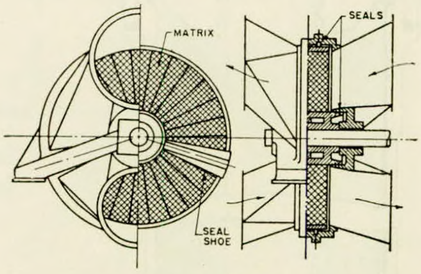
These increased voltage limits have brought about a change in the electrical supply systems for many of the larger tankers, and A.C. supply or A.C. plus D.C. generators for 110 volt D.C. lighting is now favoured by many shipowners, and has been adopted in a number of vessels recently completed or at present in the course of construction. The increased load for these installations is due to the adoption of electric power for all engine room auxiliaries, viz. main circulating pumps, forced and induced draught fans, forced lubrication oil pumps, fuel oil service pumps, fire, bilge and sanitary pumps, extraction pump, general service pumps, feed pumps, steering gear and turning gear, etc., which are in addition to the services referred to above. The supply systems generally accepted for these installations are:—A.C. for power; D.C. for lighting. Electrical energy is supplied normally by two turbine-driven alternators arranged for parallel running, one working and one standby, of 600 kW. or 550 kW. capacity as decided, 0·8 power factor, supply voltage 440 volts, 3-phase, 60 cycles. As an alternative means of supply a Diesel-driven alternator of 150 kW. capacity 440 volts 3-phase is also installed, primarily for harbour duties. For the 110 volt D.C. supply for lighting, etc., one steam driven direct current dynamo 50 kW. is installed, and, in addition, two motor generators, 440 volts, A.C. input and each designed to give an output of 50 kW. 110 volts D.C. These three equipments are standby to each other. Two 75 k.v.a. 440 volt 3-phase A.C. transformers arranged to give an output pressure of 230 volts are also fitted for supply to galley and pantry equipment, one transformer being a standby.—*Presidential address by E. Salthouse, M.B.E., read at the opening meeting of the 62nd Session of the Belfast Association of Engineers.*

Industrial Fires and Explosions from Electrostatic Origin

The author gives a survey of the mechanism of combustion and explosion and of the generation of electrostatic charges by the friction between non-conducting materials. When the charged areas are separated, the voltage increases and sparks may occur. In this manner, conditions often exist in industry for the production of electrostatic sparks. The risks are particularly great in the petroleum industry, where fuels of high flash-point may become exceedingly dangerous owing to electrostatic action. This may happen during the loading or unloading of oil products in barges and tankers. Fine dust of substances like copra, even in small concentrations, may be explosive, as the fine particles may become charged by motion through the air and the mixture ignited. Explosions have been known to take place in the anæsthetic apparatus used in hospitals owing to the generation of electrostatic charges by the flow of gas. The author believes that electrostatic risks can be practically eliminated if proper precautions are taken.—*Paper by R. Beach, read at the 1952 annual meeting of the American Society of Mechanical Engineers; Paper No. 52-A-156.*

Rotary Regenerator Performance

The effect of leakage and pressure drop in a rotary regenerator on gas-turbine plant performance is investigated in this paper. Effect of the regenerator dimensions—length, crosssection area, and rotative speed—on the regenerator and plant performance is illustrated. The efficiency of a simple gasturbine plant can be increased, the paper states, by using a regenerative cycle in which a heat exchanger is used to transfer heat from the hot turbine exhaust gas to the cooler compressor discharge air. In general, the type of heat exchanger considered for gas-turbine applications has embodied a stationary wall separating the hot and cold fluids. These have commonly taken the shell-and-tube geometry, arranged for either counterflow or crossflow passes. Principal advantage of the rotary regenerator is the possibility of designing a more compact unit



ROTARY REGENERATOR

than is possible with the conventional shell-and-tube or plane-fin stationary-surface type. Calculations show that as passage diameter is reduced to very fine dimensions, heat-exchanger volume, for a given effectiveness and pressure drop, can be reduced appreciably. The matrix of the rotary regenerator can be finely divided, without structural difficulty, and the resulting matrix volume may be one-tenth or less than the corresponding tube-bundle volume of a conventional heat exchanger. The sealing and ducting arrangements, and the like, especially for multi-disc regenerators of larger capacities, would occupy a considerable volume but indications are that a regenerator could be built to occupy perhaps one-third or one-quarter of the volume of a conventional heat exchanger. Inherent in the operation of a rotary regenerator is a leakage of some of the high-pressure air into the lower-pressure turbine exhaust-gas side. Leakage will occur at the seals due to constructional difficulties and thermal distortion. As these seals are present on each side of the rotor, cold air will leak into the cooled gas stream and will not pass through the matrix; and heated air will leak into the hot gas stream and return through the exhaust side of the regenerator. In addition to seal leakage there is the positive displacement or "let-down" loss of compressed air trapped in the matrix compartments as they pass the seals into the gas side. The let-down loss will be proportional to matrix r.p.m., and to the pressure ratio across the regenerator. The seal leakage may be considered essentially independent of rotational speed but will depend on pressure ratio.—*Paper by D. B. Harper and W. M. Rohsenow, read at the 1952 A.S.M.E. Annual Meeting; Paper No. 52-A-149.*

Back-to-back Testing of Marine Reduction Gears

In this paper is discussed the principle of testing gears by the back-to-back or power-circulation method. Results obtained from tests by this method on full-size marine gearboxes are described. The coefficient of friction for the teeth was found to be in the region of 0·047 for hobbed pinions of 3½ per cent nickel steel meshing with forged-steel hobbed wheels, and 0·03 with a case-hardened and ground mesh. The bearing losses in a gearbox, which amount to 65-85 per cent of the total losses, depending on the particular design, can be analysed with tolerable accuracy from simple theory. The oil flow in the bearings can also be analysed, and is found to consist of two components which can be separated from one another, one dependent upon inlet pressure and the other upon speed. The second component (the more important of the two) can be calculated. A considerable amount of work on the pitting of gear teeth is described. In a test-to-destruction of a gear, the crack which caused ultimate tooth failure started along a line of pitting near to the pitchline, and not nearer the root of the tooth where the maximum bending stress would normally be expected. The information which can be obtained from a normal shop test of a gearbox—a light run—is discussed. The Pametrada torque loader is described.—*Paper by A. Cameron and A. D. Newman, read at the Conference on Steam Turbine Research and Development, The Institution of Mechanical Engineers, 6th March 1953.*

Development of Marine Steam Turbine Design

The history of the development of steam-turbine designs at the Pametrada research station is outlined in this paper, and an attempt is made to set down the functions aimed at in a design and the reasons for adopting particular configurations and methods of construction. The first of a series of turbine installations is illustrated and described, and the further development that has been carried out during the six succeeding years is illustrated by recent drawings. Figs. 21 and 22 indicate the style of turbine put forward at the present time (1953) for installations where the highest possible efficiency is desired. The high-pressure turbine shown in Fig. 21 is of the double-casing type. The outer casing is fabricated from plate, and the inner casing is a steel casting supported on brackets provided within the outer casing, and suitably located in both the athwart-ship and fore-and-aft directions. The side members of the outer casing are extended forward and aft to form a girder, which additionally supports the casing of a high-pressure astern-turbine consisting of a two-row impulse wheel, whose blading is carried on an extension of the ahead rotor overhung beyond the forward bearing. The bottom half of the ahead turbine outer casing is of rectangular form, stiffened by internal ribs. As the working pressure for this casing is in the neighbourhood of only 30-40lb. per sq. in. above atmosphere, the rectangular construction can be adopted without difficulty and, to some extent, it simplifies fabrication. The top half of the outer casing is of semi-cylindrical form, with much thinner plating. In the design illustrated, steam is admitted to the turbine from below through an inlet branch cast integral with the inner barrel (not visible in the drawing). The fore and aft location of the barrel within the outer casing is in the same vertical plane as the centre-line of this branch, and the branch itself emerges radially on the bottom centre-line. Consequently, provision for radial relative expansion needs to be made only at the junction between the branch and the bottom

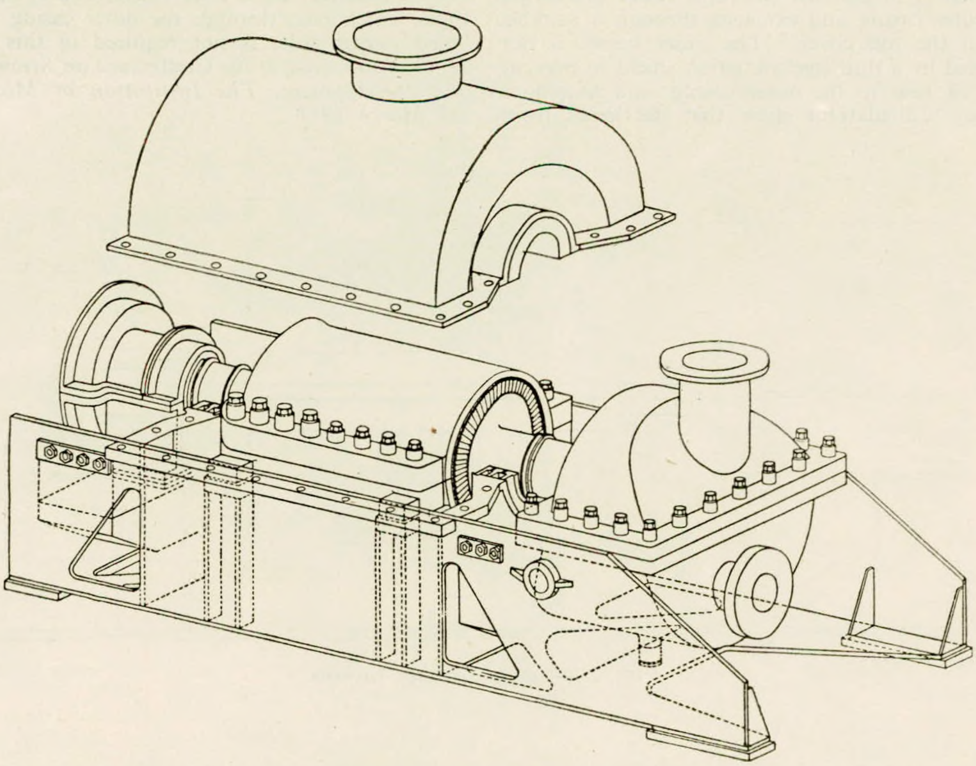


FIG. 21—*Design for 1953 high-pressure turbine*

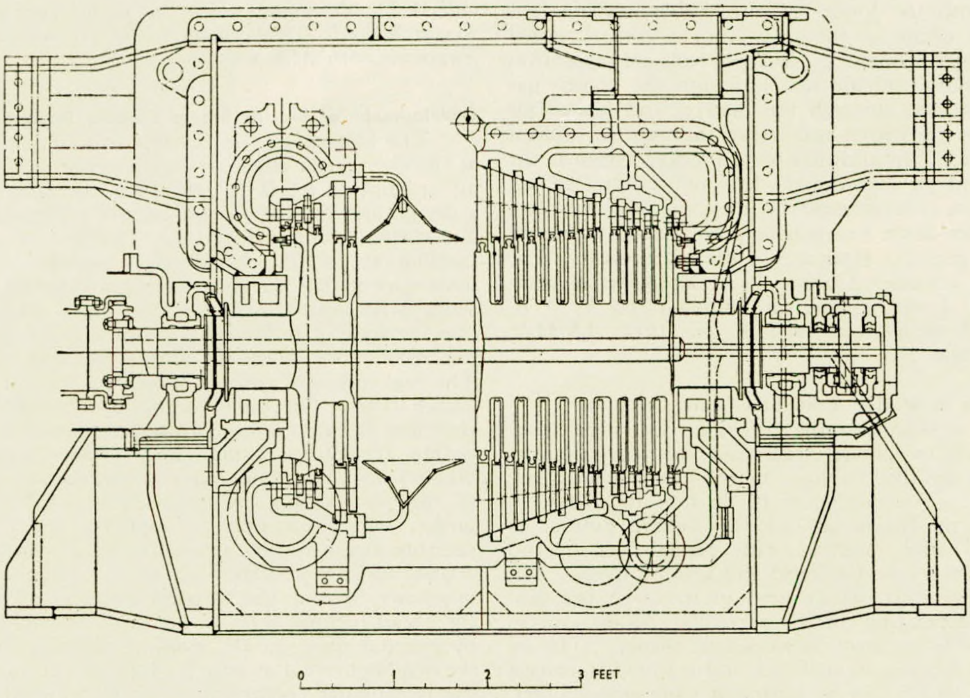


FIG. 22—*Design for 1953 low-pressure turbine*

wall of the outer casing. This connexion is by means of a multiple annulus plate bolted at its outer periphery to the bottom wall and at its inner periphery to a flange carried on a thin steel sleeve ("top hat"), which is welded to the flange of the high-pressure inlet facing. This is shown in more detail in the single-cylinder turbine illustrated in Fig. 23, in which the construction is used. The inner barrel of the high-pressure ahead turbine is open at its exhaust end, and steam discharges straight into the outer casing and exhausts through a suitable branch provided in the top cover. The inner barrel is not lagged, but is jacketed by a thin steel radiation shield to prevent excessive radiation of heat to the outer casing, and to reduce the convection loss. Calculations show that the losses from these two sources are very small. The design illustrated is for an installation under construction, in which the steam inlet temperature is 950 deg. F. In this instance only one group of nozzles is required, in conjunction with a small overload by-pass that is provided by a pipe of 1 inch bore. If two or more groups of nozzles should be required, the preferred way of arranging for this would be to fit steam-operated control valves, attached to the inner barrel and operated by small steam pipes which pass through the outer casing. The provision of "bled" steam belts is not required in this design.—*Paper by H. G. Yates, read at the Conference on Steam Turbine Research and Development, The Institution of Mechanical Engineers, 6th March 1953.*

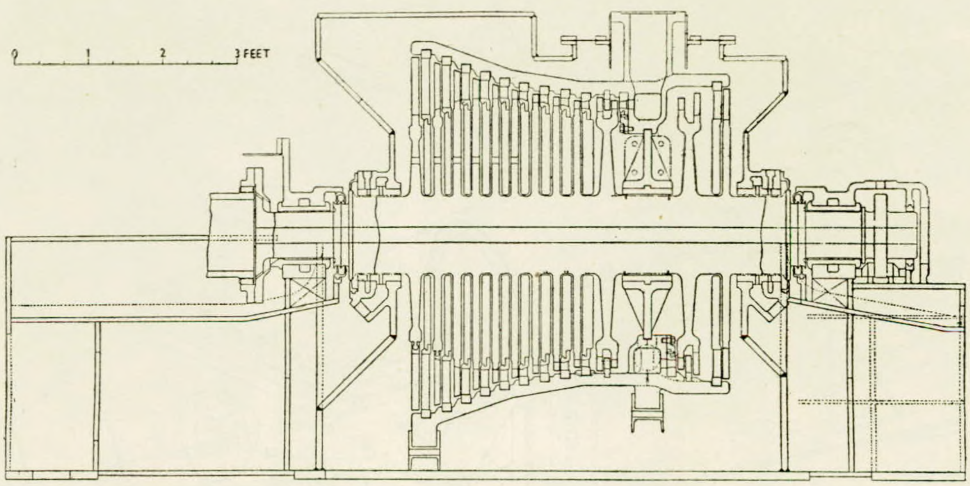


FIG. 23—*Single-cylinder turbine*

ENGINEERING ABSTRACTS

Section 3. Shipbuilding and Marine Engineering

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Cavitation-erosion of Ships' Propellers

At present, it is possible to distinguish between three different types of cavitation, viz. *laminar, burbling* and *cloud* cavitation. Until recently, cloud cavitation was regarded as a particular form of burbling cavitation. Cavitation, i.e. the formation of vapour-filled pockets in the liquid, occurs particularly at the blade tip, where the peripheral velocity is highest. The tip cross-sections are usually very thin, with s/l ratios (thickness/length ratios) of 0·03-0·05. For these cross-sections profiles with a shock-free entry are used, that is, a stagnation point at the leading edge, and a pressure distribution as constant as possible on the suction side, without a negative pressure peak. Owing to their small s/l ratio, these profiles have a fairly sharp leading edge. During the propeller rotation, the angle of incidence varies, so that shock-free entry is not maintained, and the stagnation point drops then to a position on the pressure side (see Fig. 3a). With a fairly sharp contour, the velocity on the suction side immediately behind the leading edge becomes very high and the pressure decreases considerably. The fluid breaks away at this position and forms a vortex as a result of viscosity effects. As the vortex core, where pressure is minimum, is not at a position on the blade surface, it is not very likely that this type of vortex will be the cause of erosion. The vortex core is filled with vapour, but it is separated from the blade surface by a thin layer of liquid. This type of cavitation is known as laminar cavitation. If the leading edge has a rounded contour (Fig. 3b), the flow will remain along the contour, instead of breaking away, when the stagnation point and the leading edge do not coincide. A liquid particle of mass Δm, moving with a velocity c along a circular path of radius R, will remain on this path only if the pressure gradient in the normal direction $\delta p/\delta \eta$ is at least equal to the centrifugal force $(w/g)\ c^2/R$. Hence, this gradient is inversely proportional to the radius. For aerofoil type sections with a blunt leading edge, the value of R is large, so that the liquid can provide enough pressure to avoid a breaking away of the flow. The blunt nose section, however, even under shock-free conditions, has the effect of increasing velocity and reducing the pressure on the suction side behind the leading edge, so that cavitation may again occur. However, the conditions differ from those previously considered for sharp sections, since in this instance the minimum-pressure positions are situated right on the blade surface. Bubble formation occurs and the bubbles are not stationary but burst close to the blade surface, so that it is possible that they may cause destruction of the blade material This type of cavitation is known as burbling cavitation. Cloud cavitation was only recently distinguished as a separate type of cavitation. It consists of hardly discernible small bubbles

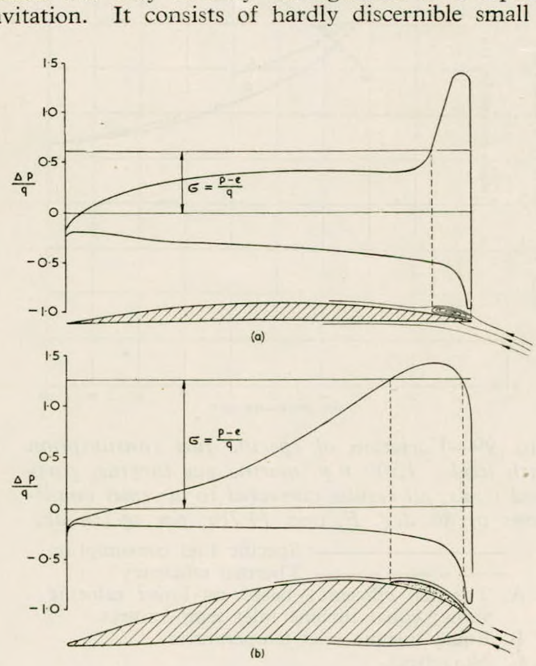


Fig. 3—*Pressure distribution of aerofoils with sharp and well rounded leading edges.*

in a compact cluster which gives them a cloud-like appearance. Hitherto, it was considered that burbling cavitation with fairly large bubbles was the cause of erosion and that the danger of erosion increases with large-size bubbles. To investigate this theory, a bronze propeller was covered with a material having a low erosion resistance and was rotated continuously in the cavitation tunnel for several days. At the end of this test, no trace was found of cavitation erosion on the blades. Repeated attempts were made to obtain erosion by means of ordinary burbling cavitation, but none gave the desired result. During a tunnel test with a non-uniform velocity field (produced by means of a board with strips of gauze-type netting), cloud-like cavitation phenomena were observed on the propeller, and after some time there was clear evidence of erosion. The cloud cavitation is not stationary, so that photographs cannot be taken with normal exposures. Having found the conditions under which cloud cavitation is generated, erosion can be produced in the cavitation tunnel in a period of a few days or even a few hours.—*Article by J. Balham, De Ingenieur, abstracted in The Engineers' Digest, Vol. 14, May 1953; pp. 163-167; 170.*

Application of Research to Marine Turbine Development

The 3,500 s.h.p. marine gas turbine set at the Pametrada station has now been running for some years and its behaviour under heating-up and load conditions has been determined. The results of a complete series of tests to determine full- and part-load efficiencies with and without reheat are set out in Figs. 99 and 100. Fig. 99 shows the measured variation of specific fuel consumption with load. These test results are corrected to ambient conditions of 80 deg. F. and 14·7lb. per sq. in. abs., and the speed of the low-pressure (output) turbine was adjusted to follow the propeller law. The fuel supplied to the primary and reheat combustion chambers can be varied independently, giving in theory an infinite number of possible running conditions at each power. The results of four different methods of control are plotted:—(1) Maintaining equal turbine inlet temperatures; (2) Maintaining constant reheat temperature; (3) Maintaining constant high-pressure-line speed. This involves a slow falling-off in high-pressure-turbine inlet temperature with power, and is the nearest approach possible to constant high-pressure-turbine inlet temperature; (4) Using no reheat. The maximum power obtainable with this condition is approximately 2,000 h.p. Method (1) gives the highest efficiency at part loads and is the method that would be adopted in service. Methods (2) and (3) give almost identical results over the range 70-100 per cent power, the efficiency being slightly lower than by method (1). The elimination of reheat, as in case (4), causes a large drop in efficiency, but the relative loss becomes less as the power is reduced, and at 30 per cent power the fuel rate becomes almost the same as that given by method (1). Fig. 100 shows the variation of cycle conditions with power for the case of equal turbine inlet temperatures. This series is given because the final fuel consumption at full load was the same as when the test started. The lowest consumption recorded in other tests was 0·491lb. per s.h.p. per hr., the fuel being a light Diesel fuel oil. It is hoped that this figure will be bettered when

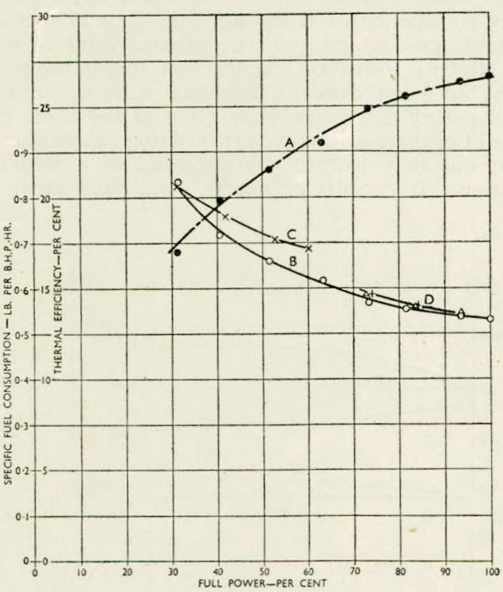


FIG. 99—*Variation of specific fuel consumption with load. 3,500 h.p. marine gas turbine, part-load trials; all results corrected to air inlet conditions of 80 deg. F. and 14·7lb. per sq. in abs.*

——————— Specific fuel consumption.
— — — — — Thermal efficiency.

A. Thermal efficiency based on lower calorific value; equal turbine inlet temperatures.
B. Equal turbine inlet temperatures.
C. No reheat.
D. + Maximum high-pressure-line speed; △ Maximum reheat temperature.

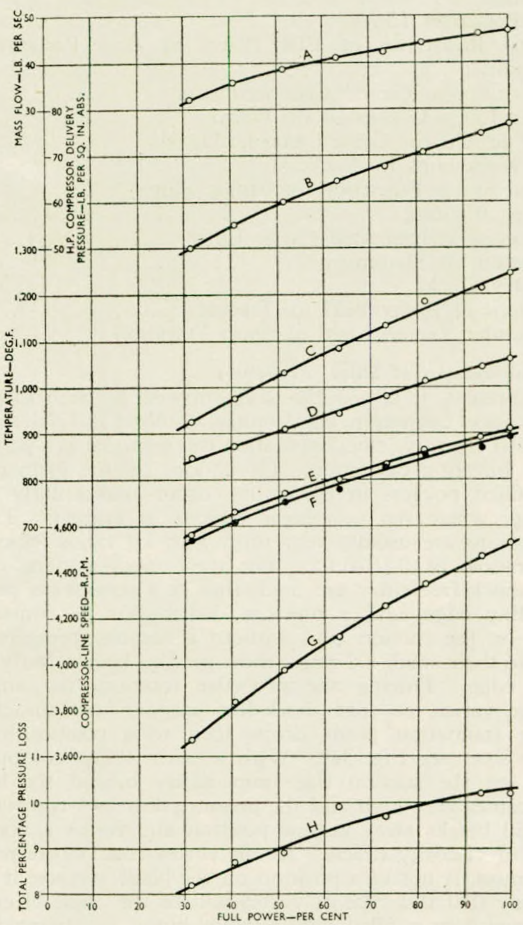


FIG. 100—*Variation of cycle conditions with power. 3,500 h.p. marine gas turbine, part-load trials; all results corrected to air inlet conditions of 80 deg. F. at 14·7lb. per sq. in. abs.*

A. Air mass flow (low-pressure compressor inlet).
B. High-pressure compressor delivery pressure.
C. High-pressure and low-pressure turbine inlet temperatures.
D. Low-pressure turbine exhaust temperature.
E. High-pressure turbine exhaust temperature.
F. Heat exchanger air outlet temperature.
G. Compressor-line speed.
H. Total percentage pressure loss.

some further modifications are made. The set completed a 100-hr. endurance run early in 1952 under Lloyd's supervision and has recently been operated on heavy residual fuel oil containing vanadium in the ash to the extent of 0·05 per cent. Laboratory tests have been carried out on the corrosion of high-temperature alloys by vanadium pentoxide at temperatures from 1,200 deg. F. upwards. These were followed by rig tests with combustion chambers operating at atmospheric pressures firing on targets made from various high-temperature alloys and fitted with thermocouples to record the metal temperatures. Various inhibitors were employed, in most of which metallic oxides were used. Later, water-soluble compounds such as magnesium sulphate (Epsom salts) and magnesium acetate were used; they were injected by a chemical pump in measured quantities into the fuel entering the burners.—*Paper by T. W. F. Brown, read at the Conference on Steam Turbine Research and Development, The Institution of Mechanical Engineers, 6th March 1953.*

Torsional Vibration in Diesel Engines

Whilst it is axiomatic that an engine should be finalized in the drawing office in such a way that critical speeds will cause no trouble in service, it frequently happens that an engine designed for one speed is installed to run at some other speed, or that an engine may be ultimately coupled to a generator in some unsuitable fashion. Such being the case, the engineer is faced with a dual problem of ensuring that his engine is right in the first place and at the same time having in the background a remedy should unpredictable conditions necessitate a change in the engine system. The basic calculations described in this paper make it quite clear that natural frequency control is governed solely by shaft stiffness and the magnitude of the associated masses. Further, it will have been fairly clear that, to change the natural frequency, a shaft stiffness change must be made near the node or, alternatively, a mass alteration must be made at some point remote from the node. But alterations are not always desirable or practicable. In certain cases, engines may have to operate over a speed range in which there is a critical speed which cannot be sufficiently moved by mass or stiffness alterations. In such cases the usual practice is to fit a torsional vibration damper. The introduction during relatively recent years of some of the high viscosity silicones has made possible the construction of a damper which has few working parts and which gives every promise of being the most reliable of all types so far made. This damper consists of an inertia member in the form of a ring of rectangular cross section hermetically sealed within a housing consisting of machine welded pressings with clearances between the inertia member and the housing, the side clearance space being filled with high viscosity silicone. The damper housing is securely fixed to the engine crankshaft and during normal operation, i.e. at non-resonant speeds, both the housing and inertia member revolve at the same speed. At a critical speed, the crankshaft vibrates and the housing takes up the vibratory motion of the crankshaft. The inertia member, however, tends to revolve uniformly and there occurs a relative motion between the housing and inertia member causing the film of viscous fluid in the clearances to be sheared, thereby creating opposing shear forces, which damp the vibration.—*Paper by C. H. Bradbury, Diesel Engine Users Association, March 1953.*

Nomad Diesel-gas Turbine Unit

The Napier Nomad Diesel-gas turbine unit consists of a simple valveless port-scavenged two-stroke Diesel engine to which is added a turbine compressor set. The axial-flow compressor is on a common shaft with a multi-stage exhaust turbine, and the turbine-compressor set thus formed is coupled mechanically to the compression-ignition engine through suitable gearing. The power of the compression-ignition engine and the surplus power available from the exhaust gas turbine are transmitted to a common, single-rotation propeller shaft, by a reduction gear in the nose of the engine assembly. A compound engine of this type has great possibilities for certain marine applications, including motor torpedo boats. The weight of the complete engine is 3,850lb. and on the static rating the equivalent horse power is 3,135, giving the remarkable specific weight figure of 1·14lb. per h.p. If the marine continuous-service rating is reduced to 2,000 h.p., we still have a very low specific weight—1·79lb. per h.p. Remembering that the overall length of the engine is 1 inch less than 10 feet, and that the full load specific fuel consumption on Diesel oil might be of the order of 0·33/0·34lb. per h.p. per hour at that output, it will be seen that a particularly interesting power unit for such vessels—and one or two other special craft—would be available. There is no complication about engine control, no bulky exchangers would have to be accommodated, and it might even be possible to make the engine direct-reversing. In the event of the latter feature being used, the engine-turbine gearing would have to be disconnected when manœuvring or running astern.—*Gas and Oil Power, Vol. 48, April 1953; p. 79; pp. 85-86; 88.*

Napier Deltic Diesel Engine

The Napier Deltic engine has been designed and developed for the Royal Navy by D. Napier and Son, Ltd., Acton, London, on behalf of their parent firm, the English Electric Co., Ltd., Rugby. The name Deltic was derived from the triangular form in which the engine cylinders are disposed. The engine has 18 cylinders, and it operates on the two-stroke cycle, utilizing the opposed-piston principle. The cylinder diameter is $5\frac{1}{8}$ inches and the stroke is $7\frac{1}{4}$ inches, giving an effective capacity of 5,300 cu. in. With the crankshaft rotating at 2,000 r.p.m., a maximum of 2,500 b.h.p. is available, the continuous rating being 1,875 b.h.p. at 1,700 r.p.m. The components of the engine are small enough to permit the use of the most modern aero-engine materials and manufacturing techniques, and this has made possible the use of such items as fully-hardened crankshafts, thin-wall lead-bronze bearings, and case-hardened and ground gears, which, at the designed ratings, give extremely long life. The Deltic engine is produced as a self-contained power plant, complete with all pumps and filters,

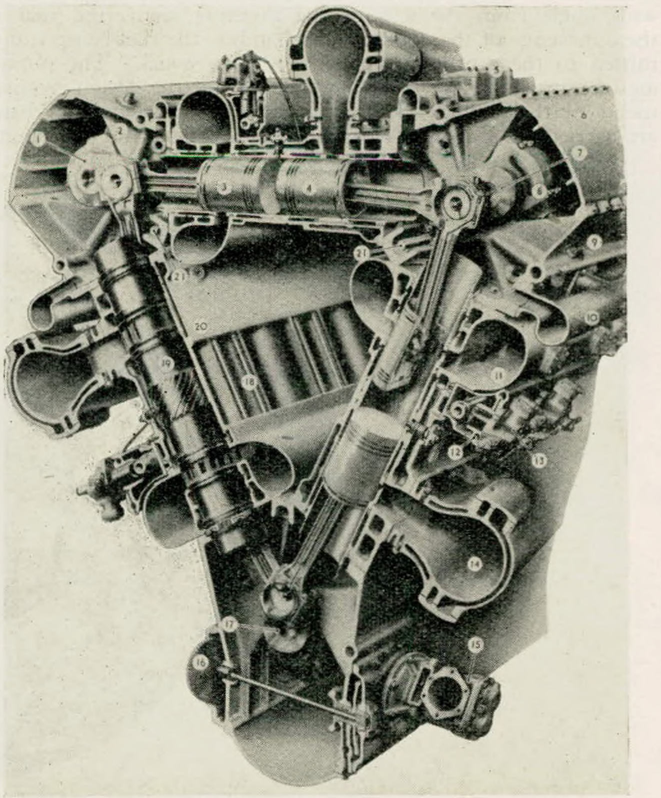


FIG. 3—*Section through No. 5 cylinder of the Napier Deltic Diesel engine, viewed from driving end*

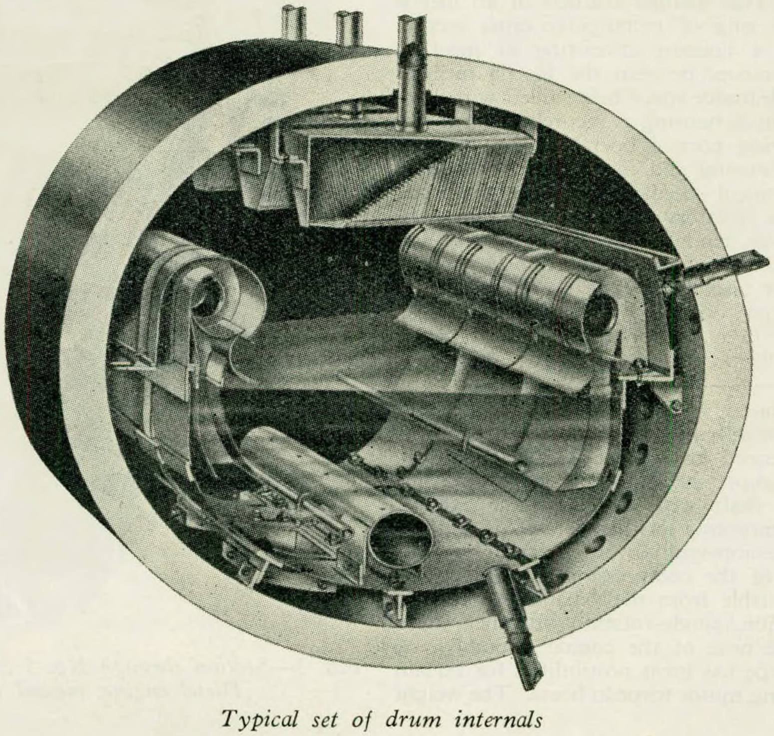
and control of both the engine and reverse gear is effected by a single lever. The outstanding feature of the engine is, undoubtedly, the disposition of the cylinders, which, in cross-section, form an equilateral triangle. With this arrangement, one crankshaft (in practice, the bottom one) must rotate in the opposite sense from the other two. From this geometrical layout, two important technical advantages arise. Firstly, there is a phase-angle difference of 20 degrees between exhaust and inlet crankshafts, which leads to an extremely efficient porting layout for scavenging and cylinder charging. The second advantage is that each crankpin carries one inlet and one exhaust piston, so that the loading on each crankpin and the power transmitted through each crankshaft are identical. When this triangular arrangement is extended through the length of the six-throw crankshafts, the resultant 18-cylinder engine has equal firing intervals of 20 degrees, which gives extreme smoothness in running. The triangulated unit is built up from three identical cast-aluminium six-cylinder blocks, which form the sides, and there are three cast-aluminium crankcases at the corners. The resulting structure is held together by high-tensile steel through-bolts, which extend from each crankcase through the cylinder block to the other crankcase. These bolts carry all the combustion loads, the cylinder blocks remaining in compression. The cylinder liners are of the wet type and are machined from hollow steel forgings. The bores of the liners, in addition to being chromium-plated and lap-finished, are so treated in the plating that they have a comparatively close-grained, dimpled surface arranged in a definite pattern. The depressions caused by this arrrangement will, in time, fill with carbon, which will absorb enough lubricant to ensure, even when starting, that the piston will not operate in a dry condition. Exhaust ports are machined around approximately half the circumference at one end; while, at the end, inlet ports are provided around the whole circumference, and are machined with a partially tangential direction of entry, thus imparting a "swirl" to the inlet air. Two injectors of the outward-opening valve type are provided for each cylinder and both are fed with fuel from a single-injection pump. The nozzles inject at a wide angle from the axis of the injectors, delivering fuel at the outer edge of the combustion chamber, the fuel being transmitted to the centre by virtue of the air swirl. The pumps are of normal C.A.V. type, modified to suit the design requirements of the engine. The six pumps for each cylinder bank are carried on a camshaft casing, which extends for the length of the cylinder block. Rack control is by a rotating shaft, the pumps being connected together by small couplings which are torsionally stiff but axially flexible, thus eliminating differential expansion effects. The design allows any one pump to be removed and replaced by another, correct timing and matching being automatically obtained.—*The Shipbuilder and Marine Engine-Builder, Vol. 60, June 1953; pp. 384-386.*

Steam Purifying Equipment

High steam purity, which is essential for trouble-free operation of the modern high pressure and high temperature steam power plant, can be attained only through the use of efficient steam drum internals. Foster Wheeler drum internals now incorporate a system of horizontal centrifugal type separators and integral chevron driers, recently developed, that works in two stages. The first stage of separation is effected by the use of the separators, while the second is performed by the integral driers, which are used to extract the last traces of water remaining after the first stage of separation. The drawing shows a typical set of these drum internals with the horizontal separators located on each side of the drum, and the integral chevron driers at the top. In this arrangement, the primary separation is completed after the steam-water mixture has made less than a full turn around the periphery of the separators. The water is removed by the primary drain and discharged horizontally at the normal water level. The separated steam then leaves through orifices in the side plates and flows to the chevron driers, where the purification is completed. The resulting steam-free water and high purity steam have the apparent advantages of maintaining maximum circulation in the first instance, and preventing superheater tube failures and eliminating soluble deposits that reduce turbine efficiency in the second. Such results are particularly important in modern power plant installations that require constantly increasing pressures and temperatures. To meet these requirements, Foster Wheeler steam generating installations are now being equipped with this new steam purifying system. The simplicity of its design also allows ready access for tube and drum inspection.—*Heat Engineering, Vol. 28, January 1953; p. 16.*

Service Performance of Boiler Brickwork

The investigation described in this paper was undertaken by the British Shipbuilding Research Association and the British Ceramic Research Association with the following objects: to



Typical set of drum internals

determine the extent of deterioration of refractory materials in water-tube boilers in the merchant service; to formulate the main causes of wastage; to consider ways of improving the life of furnace brickwork; and to provide a background for any experimental work that may be desirable. Following an introductory section which gives brief details of furnace linings commonly used, the information gained from the examination of over forty boilers of eight different types is summarized. It was found that thermal spalling or fracture is undoubtedly the most common and serious type of damage. Cracking of bolted blocks and oxidation of the bolts is probably the second most serious form of failure. Damage directly attributable to slag action is negligible, but it is possible that slag penetration of the firebrick may play some part in promoting spalling. More serious damage of all types occurs where the lining is swept by flame. The aluminous firebricks now generally used have adequate refractoriness, but it appears that rammed plastic refractory materials could be substituted with advantage in certain positions, notably in those sections containing the burner openings. The influence of service conditions on the life of refractories is examined in a further section and it is concluded that the forcing rate (lb. oil/hr./sq. ft. projected radiant heating surface) may be taken as a general indication of the severity of the operating conditions and also the extent of slag deposition. Information relating to the useful life of sections of the linings of various types of boilers is given and in a concluding section suggestions are put forward for improving the performance of boiler brickwork.—*Paper by Bryan Taylor and H. Booth, read before the Institute of Marine Engineers, 12th May 1953.*

Caustic Cracking in Marine Scotch Boilers

"Caustic cracking" is the term used to describe the intercrystalline fractures which occur occasionally in boiler components under the influence of static stress and in contact with fairly concentrated solutions of caustic soda. During the past thirty years a great amount of work has been done in an endeavour to obtain basic information on the phenomenon known as caustic cracking. It has been established that the simultaneous action of four factors is necessary. These are: (a) A boiler water which contains a caustic alkali; (b) high and unhomogeneous stresses in the metal; (c) very slow leakage or other processes causing concentration of caustic alkali at highly stressed points; (d) contact of the concentrated solution with the highly stressed boiler metal. Much of the work done has been done on small specimens in laboratories and practical experience has not produced conclusive evidence of the value of these laboratory results. At present, however, a few recommendations can be made:—(a) Caustic alkalinity of boiler-feed water should be avoided whenever possible. (b) The production of high and unhomogeneous stresses by bad workmanship, such as faulty alignment of rivet holes or the use of excessive riveting pressure, greatly increases the risk of cracking. (c) As concentration occurs in seams and crevices, internal caulking should be as effective as possible. It should be noted that very slight leakage is more dangerous than heavy leakage. The specification for the control of the sodium-sulphate/sodium-hydroxide ratio included in British Standard 1170:1947 should be maintained. (e) Current investigations into the use of lignins, tannins, and nitrates as inhibitors should be watched closely as these show promise.—*W. McClimont, Report No. 79, The British Shipbuilding Research Association. (Released 1953.)*

Investigation of Slamming

Experimental results are presented for impact tests on the model of a ship's hull. The impact was produced by dropping the model on to the water surface of a test basin. The vertical distance of free fall before striking the water was 0·6 feet. The tests were performed both for zero forward speed and for a forward speed of three knots. The hull was 11 feet in length and, with an additional mass attached near its centre, weighed about 293lb. Accelerations were measured with a Statham accelerometer attached near midship on to a strut fastened to the bottom of the hull. Three acceleration-time curves are shown for the experimental data, and these are compared with theoretical results based on a simplified assumption of two-dimensional fluid flow treated in a previous paper. Maximum deceleration measured was 4·4g., while the corresponding theoretical deceleration is 4·7g. Approximate time to reach peak acceleration is 3 millisec from experiment and about the same from theory. The authors state that more tests are necessary and that these should include a larger number of such parameters as hull shape. This further experimentation should provide more realistic tests to represent conditions actually encountered in service.—*V. G. Szebehely and S. H. Brooks; David W. Taylor Mod. Basin Rep. 812, 12 pp., July 1952. Abstracted in Applied Mechanics Review, Vol. 6, March 1953; Abstract No. 760.*

Use of Hydrofoils to Reduce Wave Making Resistance

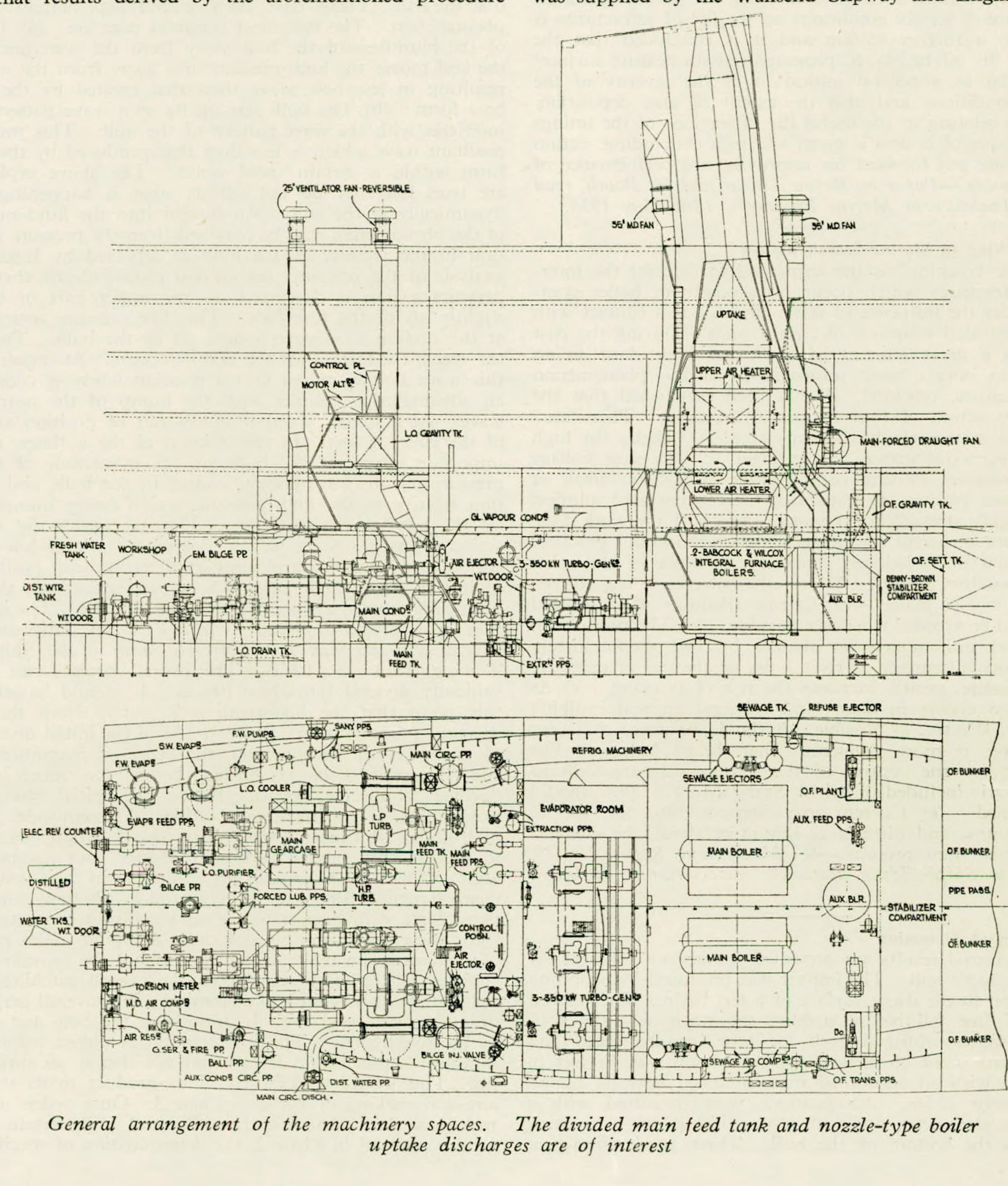
The bulbous bow has an advantageous effect on the resistance characteristics of ships in the range of $V/\sqrt{L}$ of 0·8 to 1·0. There have been several explanations for this phenomenon. The two most common ones are: (a) The shift of the bluntness of the bow away from the waterline toward the keel moves the high-pressure area away from the waterline, resulting in less bow wave than that created by the normal bow form. (b) The bulb sets up its own wave pattern which interferes with the wave pattern of the hull. This produces a resultant wave which is less than that produced by the normal form within a certain speed range. The above explanations are true, but they do not tell us what is happening hydrodynamically at the bulb. An insight into the fluid mechanics of the phenomenon may be obtained from the pressure distribution in the vicinity of the bulb as reported by Eggert. An analysis of the pressure distribution plots indicate that a low-pressure region is developed on the upper part of the bulb slightly aft of the fore foot. This low-pressure region is felt at the surface as a wave hollow aft of the bulb. The higher the speed, the farther aft the hollow occurs. At certain speeds, this wave hollow moves into a position where it combines in an advantageous manner with the hump of the normal bow wave. At a higher speed it moves out of position and some of the effect is lost. In the opinion of the authors, the most important effects of the bulb are the magnitude of the low-pressure region (wave hollow) caused by the bulb, and its location relative to the high-pressure region (wave hump) caused by the bow at the waterline. This is considering the bulb action by itself. Since we are looking for a large low-pressure region and some control over its longitudinal position, the hydrofoil at an angle of attack, as an appendage, should be promising from a hydrodynamic point of view. The investigation of a properly placed hydrofoil, as an anti-wavemaking device, is under way at the present time in the Ship Model Towing Tank at M.I.T. The overall project can be conveniently divided into three phases. It should be stressed at this point that the investigation is purely from the hydrodynamic point of view. Phase 1. As in the initial investigation of any physical phenomenon, the order of magnitude of the effect must first be established before too much effort is devoted to specific details. Hence, the initial phase of the programme is to investigate orders of magnitude. Several hydrofoils, arbitrary in size, shape and location, are attached to specific ship models (arbitrary shapes) and resistance tests with and without hydrofoils are compared in their wavemaking characteristics. Phase 2. A hydrofoil at the bow can produce two effects with respect to the seaworthiness properties of the ship; these are slamming and pitch damping. The possibility of slamming is a major disadvantage; but the fin may produce sufficient damping to cut down the overall pitching, thereby reducing the probability of slamming. The overall performance of a ship equipped with hydrofoils at the bow and stern are to be investigated. The investigation is aimed toward determining whether the hydrofoil may act also as an anti-pitching fin. This would be a beneficial by-product to its still water, anti-wavemaking properties. Phase 3. Once orders of magnitude have been determined in Phase 1 and certain practical aspects checked in Phase 2, the determination of specific design information by means of theoretical calculation, using potential theory, will be attempted.—*Part 2 of paper by M. A. Abkowitz and J. R. Paulling, entitled "The Ship Model Towing Tank at M.I.T.", read at the Spring Meeting of The Society of Naval Architects and Marine Engineers, New York, 7th-8th May 1953.*

Frictional Resistance of Flat Plates in Zero Pressure Gradient

The laws of the turbulent boundary layer on a smooth flat plate in zero pressure gradient are briefly reviewed and evaluated. On the basis of the "law of the wall" and the "velocity defect law" of the boundary layer, the displacement and momentum thickness, the form parameter H and the shear stress at the wall are obtained as functions of a local Reynolds' number, and the coefficient of total frictional resistance is derived as a function of Reynolds' number based on distance along the plate. For the numerical calculation of these quantities, formulations of these laws are chosen which are based on boundary layer profile data from various sources. It is shown that results derived by the aforementioned procedure are valid only beyond a certain critical Reynolds' number. A corollary of this result is that formulations of shear stress and total frictional resistance of flat plates, which have been derived by assuming that the laws of a turbulent boundary layer could be extrapolated to zero Reynolds' number, are fundamentally incorrect. The results of the present analysis are compared with values obtained from boundary layer measurements and also with Schoenherr's formula as representative of a mean of total drag measurements of smooth flat plates.—*Paper by L. Landweber, read at the Spring Meeting of The Society of Naval Architects and Marine Engineers, New York, 7th-8th May 1953.*

North Sea Passenger Steamship

The new twin-screw passenger, mail and cargo steamship *Leda* is the fastest vessel in regular service across the North Sea. She was built by Swan, Hunter and Wigham Richardson, Ltd., for the Bergen Steamship Co., Ltd. The machinery was supplied by the Wallsend Slipway and Engineering Co.,



General arrangement of the machinery spaces. The divided main feed tank and nozzle-type boiler uptake discharges are of interest

Ltd., and a Denny-Brown ship stabilizer is installed. The leading particulars are as follows:

Length between perpendiculars ...	410ft. 0in.
Moulded breadth	57ft. 0in.
Moulded depth to upper deck ...	30ft. 0in.
Gross tonnage	6,670 tons
Deadweight capacity on 20ft. 0in. draught	1,970 tons
Cargo capacity	75,630 cu. ft. (approximately)
Service speed	21 knots

The hull form is based upon tank tests carried out at the National Physical Laboratory. With the exception of the frames and beams which have been riveted, the structure is welded. The steel castings incorporated in it were supplied by A/S Strömmens Vaerksted and include the stern and rudder frames, upper and lower rudder stocks and bearings. There are three continuous steel decks, a lower deck forward and aft of the machinery spaces, two superstructure decks amidships and seven watertight bulkheads. The deckhouse on the boat deck, 160 feet in length and weighing 34 tons, is constructed of aluminium alloy, erected on the ship and riveted throughout. Special bulb angle sections were used for the deck beams and stiffeners, by which a maximum saving of weight is obtained and the structural efficiency improved. The *Leda* is the first Scandinavian vessel and also the first North Sea passenger ship to be fitted with a Denny-Brown ship stabilizer. The equipment is installed in a compartment, four frame spaces long, forward of the boiler room and separated from it by a screen bulkhead. The compartment contains the controlling gyroscope, V.S.G. hydraulic pumps and their driving mechanism which has the actuating cylinders arranged vertically, thus economizing in fore-and-aft space. The *Leda* is propelled by twin-screw Wallsend-Pametrada geared turbines designed to develop a total of 13,000 s.h.p. at 150 r.p.m. in service. A two-casing articulated double-reduction design has been adopted, the h.p. turbine being of impulse pattern and running at about 6,000 r.p.m. while the impulse-reaction l.p. turbines run at about 3,500 r.p.m. An astern element designed to develop 70 per cent of the ahead power, and consisting of three two-row impulse wheels, is incorporated at the after end of each l.p. turbine casing. The nickel steel primary pinions are connected to their respective turbines through flexible claw couplings and engage with built-up primary gearwheels having cast-steel centres and shrunk-on forged steel rims. There are two Babcock and Wilcox integral furnace type boilers, built by the Wallsend Slipway and Engineering Co., Ltd. They deliver steam at 450lb. per sq. in. and 800 deg. F. with a feed temperature of 310 deg. F. and an air temperature to the oil fuel burners of 339 deg. F. Each boiler has a heating surface of 7,436 sq. ft., a superheater surface of 868 sq. ft. and a total airheater surface of 8,040 sq. ft. The airheater is of vertical tubular construction arranged in primary and secondary sections. Manual control of the final steam temperature is obtained by means of a surface-type attemperator, the coils of which are arranged in the lower boiler drum. The upper drum is provided with Babcock cyclone steam separators, which ensure dry steam delivery under all conditions and, by keeping steam bubbles out of the main body of water in the drum, prevent wide variations in water level when manœuvring.—*The Marine Engineer and Naval Architect, Vol. 76, May 1953; pp. 183-196.*

Estimating Towing Resistance of Ships from Small Models

The resolution into components of the total resistance to motion of models of ships is discussed. In particular, the influence of the form on the frictional resistance and the separation phenomena at the stern is dealt with on the basis of previous investigations. Experiments at low values of the Reynolds' number ($5\cdot10^5$-2, $5\cdot10^6$) have been made with 1-metre and 2-metre models. In these experiments, the extent and the shape of those parts of the wetted surface where laminar flow is present have been determined by means of the soluble film technique. A description is given of a method for routine applications of this technique. Comparison is made with earlier tests with models of about 6 metres in length. The results seem to indicate a possibility of using such small model sizes for transforming test data to full scale conditions, if corrections are made for the laminar flow portions of the hull's surface.—*C. Falkend, Transactions of the Royal Institute of Technology, Stockholm, No. 64, 1953.*

Hobby Workshops in Tankers

A recent issue of "J.L. News", house magazine of J. Lauritzen, of Copenhagen, contains a report on the development of hobby workshops in several vessels of the fleet. The report reads: It may be recalled that Mrs. Thorkil Kristensen, wife of the Danish Minister of Finance, once threw out the idea of providing certain ships with a hobby workshop. We found it a brilliant idea, and tried to put it into practice on our tanker *Nerma Dan* in 1950. The experiment proved successful, and we decided to continue on the same lines. The *Petra Dan* was also equipped with hobby workshops, with improvements gained in experience on the *Nerma Dan*. Our third tanker, the *Berta Dan,* delivered in April this year, also has a hobby workshop on board as will the three new reefer vessels, though their floor space will not allow such big rooms as on the tankers. Our seafaring staff makes great use of the ship's hobby workshops. As shown in the plan above, the hobby workshops also have a darkroom, which we have found to be of great value in our tanker, because these ships do not stay long enough in port to leave sufficient time for developing and printing. It might be necessary to emphasize that the idea of hobby workshops, and welfare work in general, is not that spare time must be spent on work of some sort. The idea is that no person need ever be bored on our ships.—*The Shipping World, Vol. 128, 13th May 1953; p. 447.*

Sea Trials on a Victory Ship

During the years 1951 and 1952 an extensive series of trials was undertaken for the Centre Belge de Recherches Navales on a Victory ship AP3 of the Compagnie Maritime Belge. The basic purpose was the determination of the efficiency and economy of the ship and machinery in different conditions of weather and fouling. By collecting reliable and accurate records of fuel and steam consumption and engine output, it was possible to ascertain efficiency of boiler and engine under varying service conditions. As a predominant part of the programme, a comparison between service data in smooth water and tank results was to be carried out. During one of the voyages a progressive measured-mile trial was undertaken which gave basic information for the comparison with model experiments. Dr. Allan, Superintendent of the Ship Division, N.P.L., undertook to run a model and to make the comparison as part of the investigations. Calibration of the pitometer log was achieved, too, on the measured-mile trial. In different conditions of draught, fouling and weather, numerous records were collected of power, thrust, revolutions, speed through the water, ship motions, wind, and waves. The service data are analysed and the results given in a series of tables and diagrams. A similar series of trials will shortly be carried out on a Diesel-driven cargo ship.—*Paper by Professor G. Aertssen, read before The Institution of Naval Architects and The Institute of Marine Engineers, 27th March, 1953.*

Large Dry Docks

In relation to its mercantile shipowning interests, the declining position of the United Kingdom in failing to keep pace with increasing size in private and public docks is significant. There is a probable deficiency in the number of dry docks for merchant ships in this country, not only in the larger ranges but certainly for all sizes of ships over 65 feet beam, which need a working width of about 75 feet or more for repairs, and the growth in numbers of these vessels in the last thirty years with an almost unchanged United Kingdom dry dock position is very considerable, i.e., probably about ten times. Two or three years at least are required for the construction of a large dock at present, so that this is a matter that cannot be suddenly righted. This country is not a large user of floating docks, and one of the chief reasons is that apart from life and maintenance and vulnerability, the depth of water available is not generally sufficient on the rivers and estuaries where our major repairing facilities and man-power are available.—*Paper by E. L. Champness, read at a meeting of The Institution of Naval Architects, 26th March 1953.*

New Whaling Vessel

A new mother ship for whaling, which is to replace the present mother ship *Willem Barendsz,* will be built at a yard at Schiedam (N.V. Dok- en Werfmaatschappij Wilton Fijenoord). The vessel will have a displacement of 44,000 tons and a deadweight of 26,500 tons. Two 5,250 h.p. Diesel engines will be installed, giving the vessel a service speed of 14 knots. The vessel, which will be the largest floating whale oil factory in the world, will be able to reach a production of about 30,000 tons of whale oil and by-products per season. She will be provided with an evaporation plant with which about 1,000 tons of fresh water per day can be produced. The crew will number 500.—*Netherlands Economic Bulletin, No. 126, March (II) 1953.*

French Passenger Liner

A recent issue of the Journal de la Marine Marchande gives a description of the steamship *Bretagne* which is now engaged in the Marseilles to South America service of the Société Générale de Transports Maritimes. She is a sister ship in silhouette, general arrangement and subdivision, to the *Provence,* completed some eighteen months ago on the Tyne by Swan, Hunter and Wigham Richardson, Ltd., and has the following principal particulars:

Average loaded draught	26ft. 0in.
Length overall	580ft. 0in.
Length between perpendiculars	540ft. 0in.
Moulded beam	73ft. 0in.
Depth to D deck	42ft. 6in.
Gross tonnage	15,500 tons

There is accommodation for 131 first-class, 99 tourist and 1,062 third-class passengers. The crew number 258. Approximately 353,150 cu. ft. of cargo space is provided in six holds, of which Nos. 1, 2 and 4, totalling 122,000 cu. ft. are insulated. The main machinery consists of two sets of three-casing Parsons-Penhoet single-reduction geared turbines having a total output of 15,000 s.h.p. at 150 r.p.m. in service and 17,000 s.h.p. for trials. The h.p. turbines run at 3,720 r.p.m., the i.p. turbines at 3,170 r.p.m., and the l.p. turbines at 2,400 r.p.m. for a normal shaft speed of 150 r.p.m. The h.p. turbines are bladed throughout with Hecla ATV-1 steel. The turbines take steam at 570lb. per sq. in. and 850 deg. F. from three Penhoet P 41 watertube boilers. In this respect, the machinery differs somewhat from the *Provence,* which has three Babcock and Wilcox integral-furnace boilers delivering steam at 450lb. per sq. in. and 750 deg. F. It is claimed that the *Bretagne* has an advantage in this respect, and that machinery has a higher overall thermal efficiency. The output of each boiler is 30 tons per hour, and the full load efficiency is 87·8 per cent, based on the gross calorific value of the fuel. The superheaters are tubed in chrome-molybdenum steel and Penhoet economizers are fitted in the uptakes, followed by two-pass air-heaters with bypasses for lighting-up and when operating at low speeds. Each boiler has ten soot blowers and a Linderoth grit extractor is fitted in each uptake. Bailey automatic feed regulators are employed. As with the *Provence,* a Lascroux funnel has been installed to reduce the nuisance of furnace fumes reaching the decks. All the auxiliaries are driven by electric motors with the exception of the turbo feed pumps. Auxiliary power supplies are maintained at 220 volts D.C. by four 500 kW. Diesel generator sets and two 750 kW. turbo-generators. The Diesel generator sets are driven by six-cylinder four-stroke turbo-charged S.G.C.M., G6VU 55 engines. Most of the pumps in the machinery spaces are of Guinard type, while the refrigerating installation has been supplied by Paul Duclos of Marseilles. A speed of 20½ knots was attained with 15,000 s.h.p. at a fuel consumption of 0·594lb. per s.h.p. for all purposes. It is stated that this was partly due to a special streamlined design of shaft bracket and fairing.—*The Marine Engineer and Naval Architect, Vol. 76, May 1953; p. 217.*

Metals under Tensile Load of Short Duration

The object of the tests described in this paper was to determine the dynamic yield strengths of various materials when subjected to a certain type of dynamic loading. Such tests have been carried out on fifteen metals, using constant loads of 5 and 10 milli-seconds duration. These loads were produced by the longitudinal impact of a brass bar 58 feet long. Measurements of stress and strain were made during the tests, and a comparison of static and dynamic strengths was made on the basis of a 0·1 per cent proof stress when possible. It was found that for a material with a yield point there is a delay period before yielding (under constant applied stress), and for the mild steel a further investigation was made using longer load durations, which were produced by an arrangement of buckled struts.—*Paper by J. Gibson, submitted to The Institution of Mechanical Engineers for written discussion, 1953.*

Inert Arc Welding

The sketch shown in Fig. 6 is a schematic representation of the inert-gas-shielded tungsten-arc welding process showing the arc end of the torch. This process is very similar to the carbon-arc process in that the electric arc established between the electrode and the work provides only a concentrated source of intense heat which melts the abutting edges of the parts being joined. When filler metal is required, it must be added from an external source. In this process, inert gas is conducted around the electrode and through a nozzle forming a gas envelope that protects the terminal end of the electrode and the molten weld pool. Since the electrode is not consumed to any appreciable extent, it is held by the welding current conductor in the upper part of the torch. This process has found wide popularity because of its simplicity of operation, its ability to weld practically all metals and the freedom from slag. The success of this process soon led to the development of an inert-gas-shielded metal-arc welding process in which the electrode was consumed and deposited on the work as filler

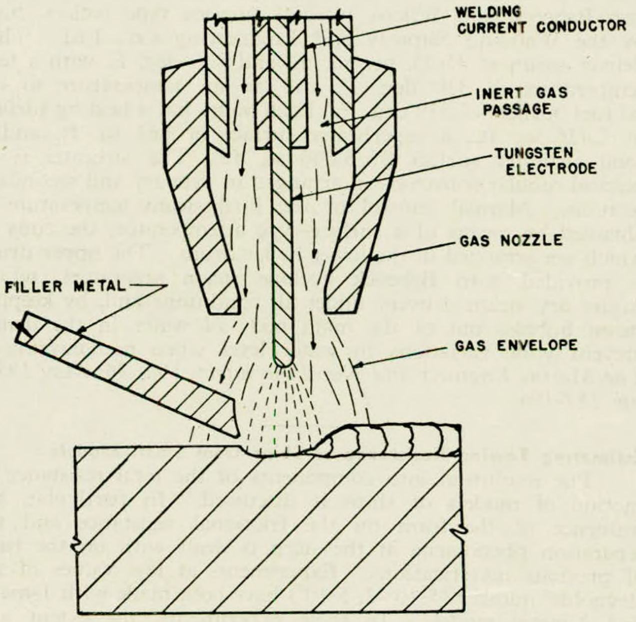


FIG. 6—*Inert-gas-shielded tungsten arc welding (non-consuming type electrode)*

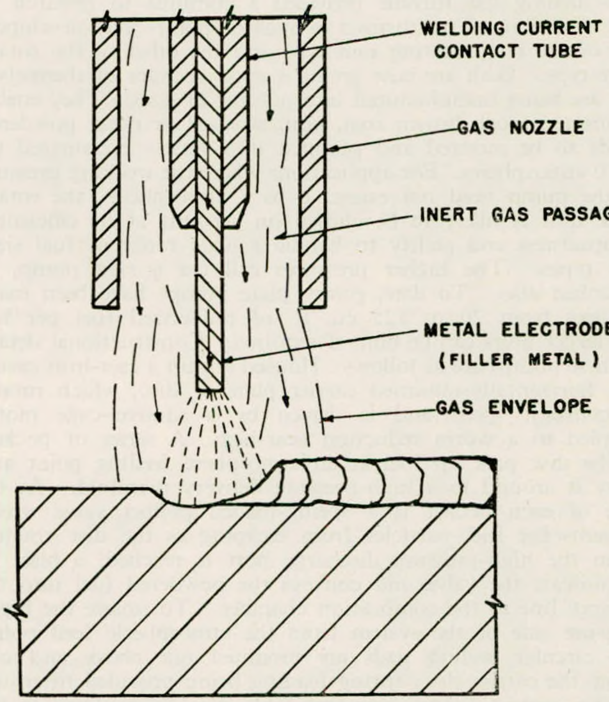


FIG. 7—*Inert-gas-shielded metal arc welding (consuming type electrode)*

metal. A schematic sketch of this process is shown in Fig. 7. The similarity of the two processes is quite apparent. In place of the tungsten electrode a metal electrode has been substituted which maintains the arc between its terminal end and the work. However, in this process, the electrode is consumed by the arc and deposited as weld metal. Consequently, the electrode must be fed into the arc at a constant rate to maintain the proper conditions for continued welding. Therefore, instead of a connector, the welding current must be conducted to the electrode by a contactor along which the wire must slide. Since, in this process, this portion of the equipment not only provides the heat for welding but also the filler wire as well, it is not generally referred to as a torch but rather as the welding head or gun. As in the tungsten arc torch, the outer portion or barrel of the metal arc gun forms the channel through which the inert gas flows to form the gas envelope for the protection of the arc stream and molten weld metal.—*The Welding Journal, Vol. 32, April 1953; pp. 299-312.*

Photo-elastic Technique for Three-dimensional Problems

For the application of photo-elasticity to the stress analysis of three-dimensional problems, the existing methods rely on the "stress-freezing" property of thermosetting resins. This again requires the model material to have certain characteristics that are not always present. The lack of these leads to experimental difficulties that may be overcome only by skilful technique. In the "sandwich" method described in the paper, the difficulties associated with stress-freezing—as, for example, large distortion of the model and rind effect—are avoided by loading at room temperature. In addition a series of fringe patterns is obtained for the specific plane selected for investigation, as distinct from one pattern only for a frozen slice. These patterns are repeatable for the same loading. The difficulties that arise in the production of a sandwich in "Perspex", with "Catalin 800" as the birefringent layer, are discussed in detail. It is emphasized that, before it is made up into the sandwich, the Catalin 800 must be subjected to additional curing to bring its elastic properties into line with those of Perspex; this also improves its behaviour at room temperature. Semicircular notches on round bars subjected to axial tension have been selected to demonstrate the correctness of the solutions obtained, and the results are in good agreement with those predicted by Neuber. The method has been applied also to the case of a thick cylinder under internal pressure, to investigate the increased stresses arising at a diametral port, and also to the estimation of the stress distribution in nut and bolt assemblies. Despite certain limitations met with in each particular application, it is evident that the "composite model technique" described in the paper provides a very useful tool for stress analysis in three-dimensional problems.—*Paper by J. H. Lamble, read at a meeting of The Institution of Mechanical Engineers, 20th March 1953.*

Photo-elastic Study of Stresses in Screw Threads

The several components of the deformation of a screw subjected to tension through a nut have been examined theoretically by Sopwith. He showed that the distribution of load along the thread was far from uniform and that it reached its maximum intensity at the plane of the bearing face of the nut. Sopwith's conclusions have now been investigated photo-elastically with a Fosterite model of a stud and nut, using the "frozen-stress" method. An additional investigation was made of the stress distribution in a tension nut with a recessed stud, suggested by Sopwith as a means of rendering the load distribution more uniform. The experimental results confirm the theory in both cases, showing a large rise in stress in the stud near the bearing face of the ordinary nut and a much more uniform distribution of stress when the tension nut is used. The stress concentration factors determined for the normal nut are higher than those found by Hetényi, who used Bakelite B.T. 61-893 as the model material. This difference may be accounted for partly by improvements in technique and partly by the greater accuracy which becomes possible with the use of Fosterite.—*Paper by A. F. C. Brown, submitted to The Institution of Mechanical Engineers for written discussion, 1953.*

Diesel-electric Pilot Vessel

The Diesel-electric pilot vessel *Wyuna*, built by Ferguson Brothers, Port Glasgow, for the Port Phillip Sea Pilots' Association of Melbourne, is 185 feet long, 39 feet broad and 22½ feet deep, with a gross tonnage of 1,400. There is complete living accommodation for twenty pilots, consisting of ten double-berth cabins, with dining saloon and lounge. Crew's accommodation consists of single cabins on the main deck, with lounges and mess-room. Some of the upperworks, including a specially designed mast and radar support, are of aluminium alloy construction. The Diesel-electric propulsion machinery was supplied by the English Electric Company, and installed by the shipbuilders. The two motors are capable of developing a total of 1,400 s.h.p. An interesting feature of the installation is that all main machines are fitted with resilient mountings to isolate machinery noise from the pilots' and crew's accommodation. The British Shipbuilding Research Association have chosen the *Wyuna* to carry out special tests on her trials.—*Shipbuilding and Shipping Record, Vol. 81, 23rd April 1953; p. 544.*

Compound Beam Experiments

The introduction of aluminium to ships' structures inevitably raises the problem of compound beams, and, indeed, if an aluminium superstructure of stressed design is used, the entire ship itself constitutes a compound beam. As part of a programme investigating the stresses produced in such a composite ship by external loadings and differential thermal expansion, a series of experiments has been carried out under the direction of Professor L. C. Burrill at King's College, Newcastle-upon-Tyne. Tests were carried out on compound beams fabricated from B.S. steel and aluminium sections. These researches covered part of a long-term programme and were sponsored by The Aluminium Development Association. Known loads were applied and the resulting strains were measured, in all cases in detail about a central transverse section of the beam, and in some cases longitudinally along the flanges as well. The basic intentions were to check the validity of the compound beam theory and also to get some preliminary idea of the effect of length on the contribution of the alloy component to the compound beam. Some fourteen composite sections were tested, the ratio between aluminium area and steel area ranging between 3·95 and 0·46. As suitable facilities were not continuously available at the University, a special testing frame had to be constructed. This consisted of a triangular framework on which the load bearing rollers rested. The test beam rested on these rollers and was loaded by a strongback. This strongback was, in turn, acted upon by a tie bar which entered a worm gear at its lower end. The strongback was so arranged that it was possible to apply a constant bending moment to the beam over the length between the load application points. In design this machine was given a high factor of safety and the scantlings of the main frame were such as to minimize deflexion. It was capable of an axial load of 20 tons or 10 tons on each load point. This gave a maximum constant bending moment over the central portion of the test beam of either 180 tons per inch for 3 feet or 270 tons per inch for 2 feet, there being two positions for the load pins. A full description of the tests is given in a report entitled "Some Investigations into the Behaviour of Compound Beams in Pure Bending" by E. C. B. Corlett, published by the Aluminium Development Association.—*The Shipping World, Vol. 128, 1st April 1953, p. 328.*

Insulation of Refrigerated Cargo Liner

The authors review the various ways in which heat enters the cargo spaces of a refrigerated ship. This is discussed in relation to the total demand on the refrigerating machinery. The effect of various depths of insulation on the ship as a carrier of refrigerated cargo is examined. In particular, the interdependence of the power of the fans used to circulate air, and the standard of insulation, is illustrated for a particular example. The effect of insulation and fan power on cargo capacity and deadweight is shown. There is an optimum depth of insulation which will give a minimum of capacity lost to insulation and air ducts and a minimal penalty on deadweight. The necessity, in some cases, to use insulation as a means of minimizing condensation may mean a heavier insulation in places than would be required on grounds of temperature control. The effect on heat leakage of position in the ship and type of space is shown. The disposition of insulation as between overheading and deck in a locker is discussed. While no suitable insulating material of lower thermal conductivity than those in common use is at present available, the possible savings in capacity, should one become available, are indicated. The calculated figures for heat leakage are minimal. In practice, margins for deficiencies in construction of insulation and performance of refrigerating machinery have to be allowed. The most common causes of excessive heat leakage are by exchange of air between inside and outside of the space and by air movement through the body of the insulation. The methods of measuring the performance of the refrigerating equipment and insulation are discussed and results of measurements on a number of ships are quoted. These suggest that, in some cases, heat leakage can be far in excess of any estimate, but that in some ships at least agreement between measured and estimated heat leakage is reasonably good. —*Joint paper by K. C. Hales and J. D. Farmer, read before The Institute of Marine Engineers and The Institution of Naval Architects, 27th March 1953.*

Pumps for Solid Fuels

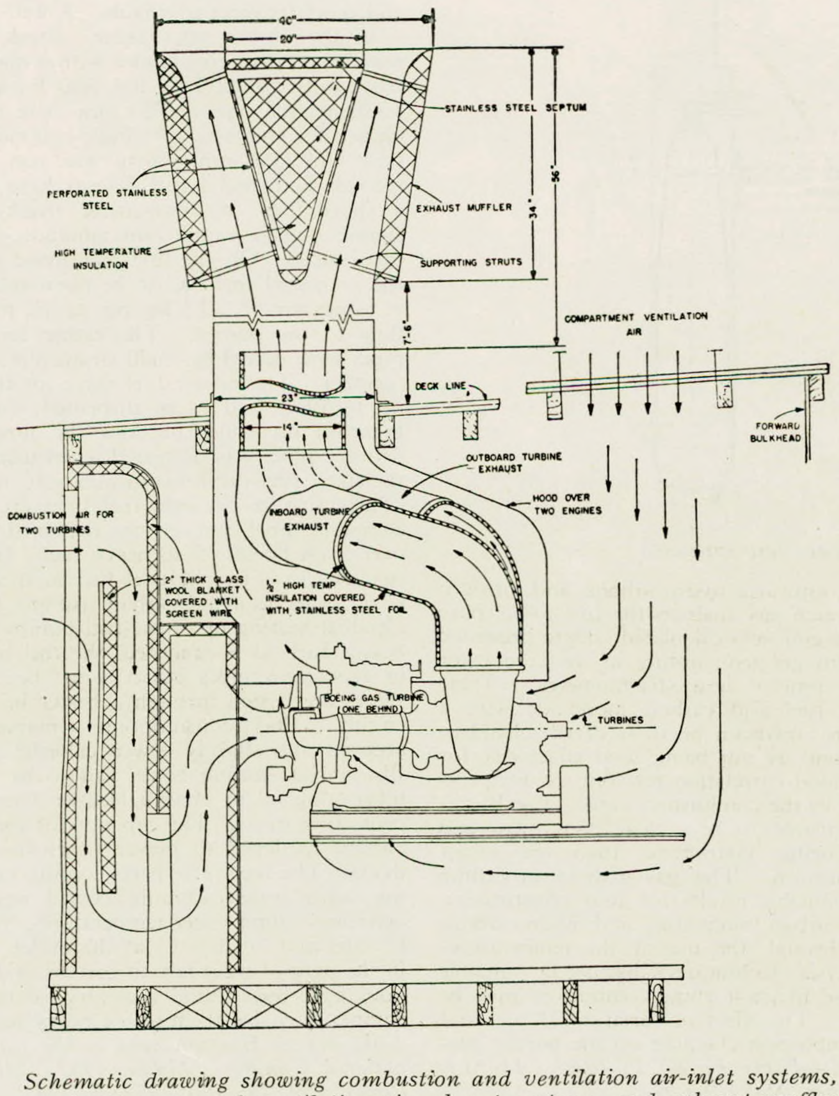
One of the first problems which faces the designer of a solid-fuel system for an open-cycle gas turbine is how to introduce the pulverized or granulated particles into the combustion space against the cycle pressure. By far the most convenient method of achieving this is to employ a solid-fuel pump designed to do much the same duty as a liquid pump. The prime need is to pressurize the fuel particles or, more accurately, to pressurize the air in which they are entrained. Until recent years, no such device had been required and very little work had been done on the pumping solids, but the advent of the coal-burning gas turbine provided a stimulus to research in that direction. Two distinct designs of pump were developed, one of the reciprocating ram type and the other of the rotary plate type. Both are now giving a good account of themselves and are being manufactured in a number of sizes. They enable bituminous coal, brown coal, peat, sawdust or other powdered solids to be metered and pumped into vessels pressurized up to 10 atmospheres. For applications where the working pressure of the pump need not exceed 6 to 7 atmospheres, the rotary plate unit is likely to be chosen on account of its efficiency, compactness and ability to handle a wide range of fuel sizes and types. The higher pressures call for a ram pump, as described also. To date, rotary plate pumps have been made in sizes from 20 to 125 cu. ft. of pulverized fuel per hr., but larger units can be built if required. Constructional details of these pumps are as follows: Housed within a cast-iron casing is a horizontally-mounted carrier plate or disc, which rotates at constant speed and is driven by a squirrel-cage motor coupled to a worm reduction gear-box. A series of pockets in the disc pick up fuel at an atmosphere feeding point and carry it around to a high-pressure delivery manifold. At the base of each pocket is a spring-loaded poppet valve which prevents the fuel particles from escaping as the disc rotates; when the high-pressure discharge port is reached, a blast of air unseats the valve and conveys the powdered fuel into the delivery line to the combustion chamber. To isolate the high-pressure side of the system from the atmospheric feed point, two circular sealing pads are mounted one above and one below the carrier disc, spring loading being provided to ensure a close contact. Any wear that takes place is confined to the sealing pads, which are relatively soft compared with the hard chromium plating of the carrier disc surface. The abrasive action of solid fuels is liable to cause differential wear on the seals, so the two pads are slowly rotated by means of a gear drive from the main shaft of the carrier plate; this keeps the surface of the pads lapped in and thus prevents air or fuel from blowing back into the feeding chute. In the sealing pads are ports of a size and section to match those of the carrier plate, sufficient lap and lead being available to ensure efficient and rapid evacuation of the pockets during the fuel discharge period. A continuous stream of airborne fuel at a uniform density is thereby provided. High-pressure air trapped in the empty pockets is allowed to escape through an exhaust pipe before the pockets return to the atmospheric filling point. Pumps of this type have behaved encouragingly in service, the first prototype having completed several hundred hours of running without any measurable wear on the sealing pads or carrier plate. Ready access to the sealing pads is a feature of the design, so that periodic inspections can be made without difficulty and replacements fitted if necessary. Important advantages of these units are the modest power consumption and the range of particle size which can be handled; performance is equally good on ⅛-inch lumps of coal or on solids pulverized so that 60 per cent passes a 200-mesh screen.—*The Oil Engine and Gas Turbine, Vol. 21, May 1953; pp. 28-30.*

Ash Deposition in Gas Turbines

The loss of power due to the deposition of fuel-oil ash on the turbine blades at present limits the use of boiler fuels in open-cycle gas turbines, and therefore prevents the more widespread application of this form of prime mover in the marine and industrial fields. The nature and occurrence of the ash-forming constituents are discussed, followed by consideration of the possibilities of removal of these from the oil. There appears to be no solution along these lines nor by removal of the ash from the gas stream. Although the factors controlling the deposition of ash on turbine blades are not yet fully known, it has been possible to throw some light on the chemistry and mechanism of deposition by making comparative analyses of ash composition of fuels and deposits, and by systematic variation of the nature of the ash in the fuels used. It seems highly probable that certain sulphates of low melting point play a decisive part in blade deposition at the temperatures of the Parsons experimental gas-turbine. Two distinct methods of reducing ash deposition on turbine blades have been established. It has been found possible, by controlling the particle size of the fuel droplets, to limit combustion in such a way that sufficient carbon remains in each particle after it leaves the combustion chamber to hold the ash in an innocuous form. In this manner deposition of the otherwise sticky ash on the blade surfaces is very largely reduced. Combustion loss by this method is considered to be of the order of 1 per cent. The second method involves the addition of certain materials to the flame, either by the medium of the fuel or of the combustion air. The most promising materials appear to be the oxides of silicon, zinc, magnesium, and possibly aluminium. It is considered that these materials function by preventing formation of the compounds of low melting point responsible for sticking. Although all these tests have been carried out at the normal turbine inlet temperature of 565 deg. C. for the experimental plant, this is equivalent to the part-load operation of gas turbines operating with the higher inlet temperatures more current today. It is also considered that the results will be applicable to gas turbines using somewhat higher inlet temperatures. In the tests the beneficial effects of combustion control and the use of certain additives have been confirmed over a relatively short period of time only. It is not possible to carry out further lengthy tests on the experimental gas-turbine and confirmation of their effectiveness over long periods of time will be obtained only when gas turbine plant burning residual fuel comes into operation. The authors consider it likely that neither of the above methods may be completely satisfactory on its own, but that a combination of factors such as turbine design, combustion control, and the use of additives, will eventually enable gas turbines to operate successfully for reasonably long periods on residual fuel without intolerable loss of performance.—*Paper by A. T. Bowden, P. Draper and H. Rowling, read at a meeting of The Institution of Mechanical Engineers, 24th April 1953.*

Gas Turbines in Closed Compartments

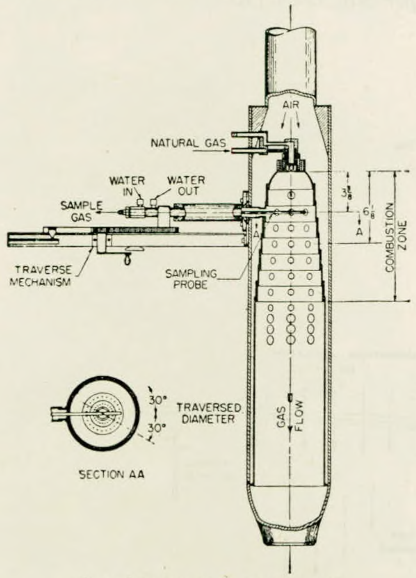
The U.S. Navy is presently constructing a group of wood-hulled vessels. To meet size and weight limitations, gas-turbine engines are being installed for generator drive. Problems of design which arose in connexion with the installation of gas-turbine engines in a closed compartment are discussed and the solutions as developed at the U.S. Naval Engineering Experiment Station are given. The problems discussed in this paper are: (1) The maintaining of satisfactory temperature limits in the compartment, (2) the maintaining of satisfactory limits on the exhaust ducting and stack, (3) a combustion air-inlet system that will prevent sea-water droplets entering the compressor, partially silence the compressor noise, and give a low intake pressure loss, and (4) an exhaust muffler that will partially silence the turbine exhaust noise and give a low-exhaust system pressure loss.—*Paper by L. W. Shallenberg, J. O. White and K. E. Schlachter. A.S.M.E. 1952 Annual Meeting; Paper No. 52-A-130.*



Schematic drawing showing combustion and ventilation air-inlet systems, combustion gas and ventilation air-exhaust systems, and exhaust muffler

Temperature and Gas Analysis of Gas Turbine Combustors

Problems encountered in the design of gas-turbine combustors are not essentially different from those associated with any other industrial combustion equipment except for the complications introduced by the extreme space limitations and the wide range of fuel-air ratios under which the former must function. In a satisfactory gas-turbine combustor, optimum conditions with respect to preparing the fuel, performing the mixing of fuel and oxidant, and piloting the combustion must be accomplished without the aid of adjustable dampers or other controls which function as operating conditions change. To achieve expeditiously such a design, detailed information as to the admission of the fuel and oxidant and their subsequent mixing and reacting is highly desirable. This paper presents a report of explorations of the possibility of obtaining such information by temperature and gas-analysis surveys in the combustion zone of a gas-fired combustor. A radiant target pyrometer, developed especially for gas-temperature measurements, and a sonic-flow orifice probe similar in design to that proposed by Blackshear have been successfully used in making temperature explorations in the combustion zone of a gas-turbine combustor. A continuous gas-sampling procedure employing a Beckman oxygen analyser was used to make the gas-analysis surveys in the combustor. From measurements of oxygen concentration in the sample at specified points in the analysis train, the percentage by volume of carbon dioxide,



Combustor test apparatus

oxygen, carbon monoxide, saturated hydrocarbons, and nitrogen were determined. From such gas analyses the fuel-to-air ratio throughout the traversed region was calculated. Data presented indicate that the maximum gas temperature in the combustor occurs where the fuel-air ratio is near stoichiometric. There are regions of unburned fuel and carbon monoxide with a deficiency of oxygen in the upstream portions of the chamber, indicating that these regions are not being used effectively for combustion. There is a good correlation between the temperature and the fuel-air ratio in the combustion zone. The Blackshear temperature probe proved to be a more convenient and reliable temperature-measuring instrument than the target pyrometer for this application. The gas-analysis procedure gave consistent and reproducible results for four constituents, oxygen, carbon dioxide, carbon monoxide, and hydrocarbons within ± 2 per cent. Through the use of the temperature-measuring and gas-analysis techniques discussed, valuable design information for use in gas-turbine combustors may be obtained, the paper states. The effect of variation of fuel and air distribution in the combustion chamber on the burner performance may be studied.—*Paper by K. L. Rieke, A.S.M.E 1952 Annual Meeting Paper, No. 52-A-97.*

Corrosion in Condenser Systems

The main effort of long-standing research into this subject by the British Non-Ferrous Metals Research Association is now devoted to a study of intercrystalline attack of certain materials which are widely used because of their high resistance to impingement attack. Intercrystalline attack, similar in form to that occasionally encountered in practice, has been reproduced in the laboratory, and the factors responsible for such failure are being studied. One of the alloys developed for marine water services in the course of this work is extensively hot- and cold-worked in installation, and some troubles have been encountered in hot-working the semi-finished products. During the year it has been shown that these are largely attributable to the effects of minor impurities and recent experience suggests methods of controlling their effects.—*The Marine Engineer and Naval Architect, Vol. 76, May 1953; p. 218.*

Tests with Stork Hesselman Engines Running on Heavy Fuel

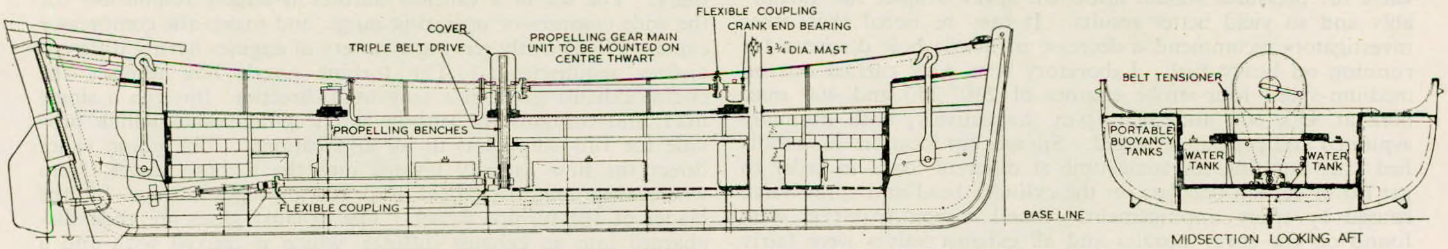
After having pointed out that the fuels indicated by the term "boiler oil" comprise a wide range of viscosities and constituents, making the task of the engine builder a rather difficult one, the authors deal with the various aspects of adapting the large engine in question to the use of such fuels. It is stated that the wear of cylinder liners, pistons and piston rings will undoubtedly be considerably higher as compared with operations on Diesel oil, involving higher costs of maintenance and more frequent overhauls. A well-trained engine room staff is of the utmost importance. Stork have confined their test mainly to heavy fuel grades with a maximum viscosity of 1,500 seconds Redwood I at 100 deg. F., and a double-acting two-stroke engine type of 720 mm. bore by 1,200 mm. stroke was chosen for the tests. A single-cylinder test engine of 540 mm. bore and 900 mm. stroke was run successfully on a 3,500 seconds Redwood I fuel. Tests have also been carried out for a short time on four-stroke trunk-piston marine auxiliary engines, where rapid contamination of the crankcase oil was experienced. On the first-mentioned engine it was found that the preheated fuel had to be circulated to the h.p. fuel pumps at a pressure of 2-2·5 kg. per sq. in. to ensure an uninterrupted flow to the pumps. The rather long fuel pressure delivery pipes were heated by small steam pipes, both pipes being lagged together. The master fuel valve for the bottom cylinders, used up to 1952, had to be dispensed with in order to get good injection on boiler oil and the inner diameter of the fuel delivery pipes was decreased when using hot fuel so as to bring the static pressure between injections to an acceptable low level. (These smaller-diameter fuel lines are now standard on Stork engines.) Fuel cam settings remained unaltered when changing over from Diesel oil to heavy fuel. Injector valve cooling was switched over from Diesel fuel to fresh water, kept at 15 deg. C. in order to avoid carbon trumpet formation at the nozzles. Gradual heating up of the fuel pumps by gradual circulation of heated fuel oil is practised, the fuel lines are heated by means of the steam pipes adjacent to them, and the fuel injection valves are heated through a heater in the nozzle cooling water circuit. Fuel consumption remained normal, taking into account the slightly lower calorific value of the heavy oil. Piston rod stuffing boxes had to be lubricated with a doped lubricating oil to prevent lacquer formation on the piston rod protecting tubes. The cap nuts of the bottom fuel valves were chrome-hardened to prevent corrosion from combustion products. The fuel valve parts coming into contact with the cooling water were cadmium coated against corrosion. As for cylinder cooling water temperatures, Stork recommends 60 deg. C. inlet and 70 deg. C. at the outlet. As a broad conclusion, in the light of these tests it may be said that no major modifications have been found necessary on the large Stork two-stroke engines to enable them to use heavy fuel.—*Paper by G. Wieberdink, and A. Hootsen, read at the Internal Combustion International Congress, Milan, 1953. Abstract in Gas and Oil Power, Vol. 48, May 1953; p. 124.*

Grinding Large Crankpins on Board

When the crankpins of a large Diesel engine require to be trued up and grinding can be employed, it represents an extremely important saving of time and money if the operations can be effected with the machinery in place. An illustration is given of this procedure being undertaken on a Doxford-type engine, one of two units on board a P. and O. S.N. Company's cargo liner, the aggregate output of the machinery being 12,500 b.h.p. at 112 r.p.m. The cylinder diameter is 725 mm. and the combined stroke 2,250 mm., the diameter of the centre crankpins being about 560 mm., or approximately 22 inches. The side rod pins require no attention, in this instance. The necessity for this grinding operation was not due to normal wear on the engine but to a mishap in which the ship was involved. The work was undertaken by Nicol and Andrews, Glasgow, who completed the grinding of the pins, five in each engine, in a fraction of the time otherwise needed had the engines been dismantled for the shafts to be taken ashore. Such a process would have meant immobilizing the ship for about three months and the monetary saving can scarcely be estimated. The machine is driven by an electric motor and the tool is traversed along the pin by hand. However, an automatic machine will shortly be in production.—*The Motor Ship, Vol. 34, June 1953; p. 107.*

New Hand Propelling Gear for Lifeboats

Ministry of Transport approval has now been given for the hand-operated propulsion gear developed and manufactured by Viking Marine Co., Ltd., builders of corrosion-resisting aluminium-alloy lifeboats. It consists of a main pulley on the centre thwart by three vee-belts connecting the pulley to the propeller shafting. A crank is fitted forward and aft of the pulley, each being set at 90 degrees to the other. To propel the boat, the persons operating the gear sit facing inboard and each rotates the respective crank. The gear shown is fitted on a 24-ft. boat having a beam of 8ft. 6in. and a depth of 3ft. 6in. It was tested by the M.o.T. with a load corresponding to forty persons, and when propelled by ten men gave a speed of 3·73 knots. On larger boats of, say, 30 feet, it is likely that an extra pulley and cranks will be fitted. Installation is a relatively simple matter and the gear could be fitted to all types and sizes of boats. Although still in the early stages of production, it is stated that the cost of the gear for a 24-ft. 40-person boat will be almost £70 less than that for the lowest-priced 8 h.p. motor installation and cheaper than existing forms of hand-propelling gear. Current regulations can be interpreted to allow such a mechanical arrangement to be used instead of a motor for lifeboat propulsion duties. The ten tankers which have been ordered by the Lowland Tanker Co., Ltd., will each have one of the four lifeboats fitted with the new Viking hand gear, which is based upon an original design by Burness, Kendall and Partners, Ltd., naval architects.—*The Motor Ship, Vol. 34, June 1953; p. 97.*



General arrangement plans of Viking lifeboat with hand-propelling gear

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Turbo Supercharging of Large Two-cycle Diesels

The essential advantage of supercharging is that one burns more fuel by having a greater weight of air in the cylinder and thus the output is increased. At the same time the increase of the heat load should, by sufficient residual air, cooling or charging air, etc., be kept as small as possible. So it is just a matter of getting as much energy as possible from the exhaust gases in order to obtain the highest possible charging density in the cylinder, thus to reduce the mechanical work required for scavenging and charging, and with it the fuel consumption. This can on the one hand be achieved by high efficiencies of exhaust gas turbine and blower, which can be obtained by systematic development of normal units or by special designs, like the Tosi unit described in the paper by Scirocco-Casci, or the M.A.N. turbo unit, mentioned two years ago in Paris. On the other hand the energy available for the turbine can within certain limits be artificially increased by the use of the exhaust pulse. The system described, however, only partly uses dynamic impulse waves and their reflections and is mainly based on the static loading of the exhaust volume between cylinder and turbine, working, therefore, with a definite storage capacity to the turbine. In a four-stroke engine the factors are favourable in that the cylinder on starting up acts as its own scavenge pump and that, as a result of the higher combustion temperatures permissible, the exhaust temperatures and thus the exhaust energy are higher. Here turbo-charging has become general since its introduction about thirty years ago by Büchi and usually makes use of the exhaust pulse principle. In the two-stroke, for some time it has not been thought possible to omit the separate scavenge pump, which, especially on starting up and at part load, where the exhaust gas temperatures are insufficient for the exhaust turbine, delivers the excess energy required. Now, by the maximum use of the exhaust energy it has been found possible to leave out the scavenge pump. In the large two-stroke engines with common charging pressures of 1·6 atm., sufficient energy can be obtained by using the exhaust impulses if the exhaust cross section is increased sufficiently fast and the volume of the exhaust line is kept sufficiently small. With higher charging pressures on the other hand the constant pressure turbine can become more favourable. The author discusses the advantages and disadvantages of operation with and without scavenge pump, with different arrangements of turbine and scavenge pump (series, parallel, etc.). After a general discussion of the turbo-charging of both four-stroke and two-stroke engines, the author passed on to a consideration of troubles which may occur in the turbo-charger when using heavy oils. These include fouling by coke-like residues or ash deposits of both engine parts and the turbo-charger turbine. The deposits are sometimes corrosive at temperatures over 500 deg. C., the corrosion being vanadium attack. Experience shows that acid condensation, due to the temperature falling below the dew points, very rarely occurs.—*Paper by Dr. Carletti, read at the Internal Combustion Engine International Congress, Milan, 1953. Abstract in Gas and Oil Power, Vol. 48, May 1953; pp. 121-122.*

Limitations of Boiler Fuel for Diesels

The paper deals with service experience with marine propulsion engines and tests on relatively small medium-speed engines operating on various grades of boiler fuel. It is held that not all such engines, having a bore of from 200 to 500 mm. would be capable of operating satisfactorily on the more viscous fuels, but reasonable performance can be expected from many engines in this category on low or medium viscosity fuel oil, especially when used as generating sets with steady operating conditions. Results on twenty-three tankers owned by the British Tanker Company on fuels of up to 1,500 seconds Redwood I at 100 deg. F. are reported on in the paper and it is mentioned that two ships have operated experimentally on fuels up to 3,500 seconds. Special attention is given to fuel valve water-cooling, with inlet temperature of 46 deg. C. and 52 deg. C. at the outlet, for the purpose of preventing local fuel temperature rise and thermal cracking of the fuel at the nozzle tips. Cylinder liner wear figures are given for both two-stroke and four-stroke engines running on Diesel fuel and, for comparison, on boiler fuel. It is held that ash-forming constituents of the boiler fuel are mainly responsible for the increase of abrasive wear experienced, sulphur only playing a small part because liner skin temperatures do not normally fall below the dew point of sulphurous gases; the thickness of cylinder liners on large marine engines ensures this condition. Regular draining of scavenging belt spaces of two-stroke engines is strongly recommended when using heavy fuels, so as to avoid the accumulation of excessive deposits and the danger of scavenge main fires. Exhaust valve guttering is reported to be a common phenomenon when running four-stroke engines on boiler fuel, and this necessitates more frequent attention to exhaust valves. It appears that fuel valve condition has a major effect upon exhaust valve failure from seat guttering. Guttering, which is illustrated in the paper, can be counteracted by using improved valve lid materials and better cooling of the valve assembly and seats. The phenomenon of valve guttering is explained as being due to the valve seat becoming coated with molten sodium sulphate, which ultimately breaks away locally to allow gas passage and erosion to occur. Increased nozzle hole diameters were adopted for heavy fuel operation for two reasons: (a) to compensate for the reduced volumetric calorific value of the fuel oil as compared with Diesel oil, and (b) to limit the increase in fuel line pressure. Increased nozzle hole size results in increased droplet size and greater penetration of the sprays, with the effect that an increase in spray impingement upon the piston crown becomes noticeable. It is believed by the authors that more holes should be a better way of getting more orifice area, and it is also thought that increased fuel valve lift pressures should affect the spray droplet size favourably and so yield better results. It may be noted that other investigators recommend a decrease in nozzle hole diameter for running on heavy fuel. Laboratory tests were carried out on medium-speed four-stroke engines of 230, 250 and 400 mm. bore at 450, 600 and 333 r.p.m. respectively, both normally aspirated and pressure charged. Sprayer tip location was modified after running for some time at different loads in order to get rid of carbon deposits on the cylinder head and spray valve caused by spray impingement. Small carbon trumpets were found on the spray nozzles and all exhaust valves were fairly heavily pitted, with one slightly guttered. The mean wear rate over 1,310 hours operation on heavy fuel was 2½ thousandths per 1,000 hours. Bearing in mind the severe test schedule, it is considered that such an engine would run for 1,000 hours between piston overhauls at normal service load factors. This period could be extended if a heavy-duty lubricating oil were used; straight mineral lubricants were used during the tests.—*Paper by G. M. Christie and C. L. Bailey, read at the Internal Combustion Engine International Congress, Milan 1953. Abstract in Gas and Oil Power, Vol. 48, May 1953; pp. 124-125.*

New De Laval Pressure Charger

The new De Laval turbocharger has been designed and developed specifically for high-pressure pressure-charging service. The series A model, which has been built in two different sizes, is designed for a pressure ratio of 2 to 1 under full-load engine operating conditions, and slightly higher under engine overload conditions. The later models are being planned to be capable of producing pressure ratios of 3 to 1 under engine full-load conditions. One of the outstanding characteristics of the De Laval pressure-charger is the high efficiency of the turbine and the compressor. The compressor is of the mixed-flow type, so called because the air enters axially and flows through and out of the impellers partly axially and partly radially. This type of compressor was originated by De Laval more than twenty-five years ago, and has gradually been refined aerodynamically and improved structurally, until today efficiencies and performance characteristics are obtained which are substantially better than those which can be achieved with either the axial-flow type or the radial-flow centrifugal type, the latter being commonly used in conventional pressure-chargers. A departure from conventional practice was also made in the design of the turbine, because it was found that the desired turbine characteristics and high turbine efficiency could not be obtained by using the conventional axial-flow type of turbine. The mixed-flow centripetal turbine—which De Laval first developed as long ago as 1928—was found to be much more suitable for service in exhaust-gas-turbine-driven pressure-chargers. One of many attractive features of the centripetal turbine is the fact that its blades are very rugged and that they are few in number. In the axial-flow turbine, many more blades are necessary and, therefore, they must be comparatively narrow and delicate. The construction of the rotor in the new design is also different from that of any other pressure-charger currently available. The mixed-flow impeller and the centripetal turbine wheel are combined in a single rotor structure in which the compressor blading is carried on one side of the rotor hub and the turbine blading on the opposite side. This arrangement eliminates all parasitic losses such as turbine and compressor leakage, windage and friction of non-working surfaces—losses which are unavoidable in a design employing separate compressor and turbine rotors. By eliminating these losses, the turbine and compressor efficiencies are substantially improved over and above the improvements already achieved by the aerodynamically highly improved mixed-flow compressor blading and the refined centripetal turbine blade configuration. This so-called "Monorotor" construction offers the additional advantage of extreme simplicity, eliminating a great number of parts such as air and gas seals and partition walls. The mixed-flow compressor has an axial air inlet and a vaneless diffuser, terminating in a scroll housing from which the air is discharged tangentially. The use of a vaneless diffuser is largely responsible for the wide compressor operating range, and makes the compressor capable of efficiently serving a variety of engines having different airflow requirements. The turbine nozzle box receives the engine exhaust gases in a tangential direction, through a single inlet, and contains the turbine nozzle guide vanes which provide for full admission in all applications. The guide vanes direct the flow radially inward into the turbine blades, from where they are discharged into the passages formed by the blades of the turbine wheel. The exhaust gases are then discharged into an exhaust diffuser, which is coaxial with and a part of the turbine housing. The turbine housing is made from a heat-resisting alloy, and is insulated and lagged instead of being water-cooled.—*Gas and Oil Power, Vol. 48, May 1953; pp. 105-107.*

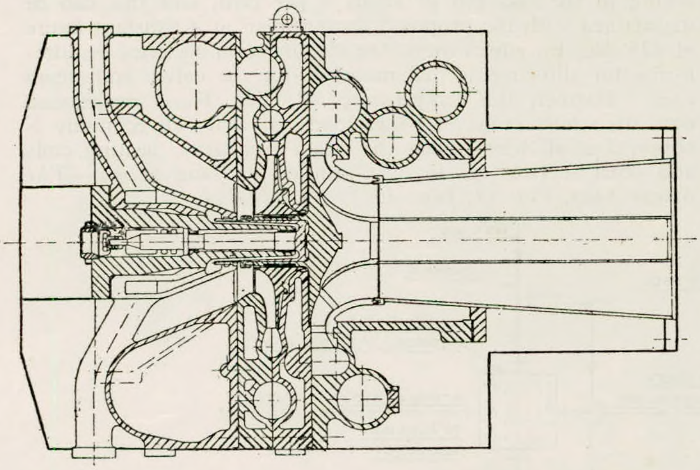
Heavy Fuel Oil for Geared Marine Diesels

Crosshead engines with piston rod stuffing boxes do not suffer from crankcase oil contamination by combustion residues. Even if the ash content of the fuel is reduced to 0·02 per cent, it should be appreciated that this apparently very low figure works out to be about 8 kg. of abrasive products a day passing through a 10,000 b.h.p. engine. Split-type filters in the delivery fuel lines to the injectors are recommended to prevent choking of the nozzles. All normally used analysis figures of the fuels are thoroughly discussed in the paper and their value for judging combustion quality indicated. Concerning the measures to be taken to improve the running of Diesel engines on these heavy fuels the author recommends a thorough preheating of the engine before starting with fresh cooling water to 40 to 50 deg. C. Also, pistons cooled by oil should be brought to an adequate temperature. Ample time should be allowed for this, eight hours being normally required.

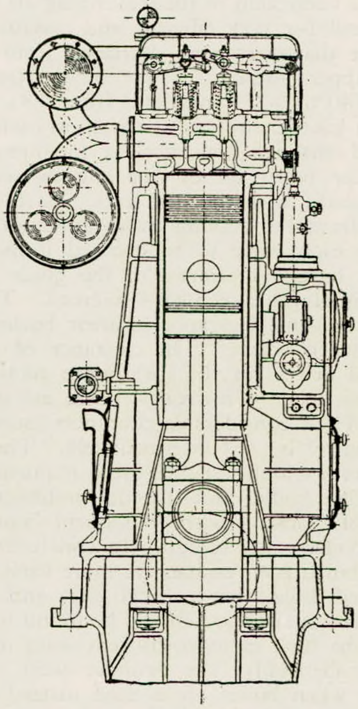
Great importance is attached to the methods of purifying and filtering the fuel before use and also before taking the fuel into the bunkers. A combination of disc type filters with strainers is recommended. Engine room personnel should be well trained in the use of heavy fuels and supplied with such means and instruments as are needed to examine the properties of the fuel bunkered and to judge the engine condition in service. Oil concerns and engine builders together should make systematic efforts to facilitate the running of Diesel engines on heavy fuels for the benefit of engine users. The author stresses the increasing tendency to use geared Diesel engines as a competitor of geared turbine drive and generally examines geared Diesel drive. A favourable point regarding operation on heavy fuels is that geared engines can quickly be disconnected when lower ship speeds are required, enabling the remaining engines to continue to run at high load. Thus the heating of heavy fuels and the cooling of cylinder jackets, cylinder covers, and pistons, and other conditions remain more or less unchanged. The same applies to the fuel injection pumps and valves.—*Paper by W. Brose, read at the Internal Combustion International Congress, Milan, 1953. Abstract in Gas and Oil Power, Vol. 48, May 1953; pp. 126-127.*

Tests on the Use of Heavy Fuels in Tosi Four-stroke Exhaust Gas Supercharged Diesel Engines

The author describes comparative tests with gas oil, Diesel oil and heavy oil which the firm of Franco Tosi carried out on the test bed at Leguano with two medium-size engines. One of them, an unsupercharged engine, had one cylinder of 310 mm. bore by 405 mm. stroke and ran at 500 r.p.m. With heavy oil a mean pressure of 6·5 kg. per sq. cm. was reached against 7·2 kg. per sq. cm. with gas oil, the exhaust temperature being slightly higher than with gas oil at the same load. The second engine was a six-cylinder engine of 310 mm. bore and 450 mm. stroke running at 428 r.p.m. It was pressure-charged on the Büchi system with extensive utilization of the exhaust impulses. The turbo-charging set used was of the latest Büchi design with centripetal turbine and centrifugal blower and was made by Tosi. This arrangement excels by its effective use of the exhaust energy, demonstrated by high charging pressures with low exhaust temperatures. Both with gas oil and with heavy oil mean pressures of 9·5 kg. per sq. cm. were reached, with fuel consumption increased in roughly the inverse proportion of the calorific value of the fuel. In order to determine the extent of fouling of the engine in continuous service, tests were carried out with Diesel oil for 48 hours' duration at 9·8 kg. per sq. cm. mean pressure and with heavy oil for 88 hours at 9·25 kg. per sq. cm. The rate of fouling is shown in diagrams in the paper. The deposit was, as expected, somewhat greater with heavy oil operation than on gas oil; but, although the tests were short, the deposits were not excessive with either fuel. In order to estimate the increase in fouling, the deposits in the individual cylinders were scraped off and weighed several times during the tests. From this it was apparent that, after a rapid increase of deposits during the first hour, the rate slowed down. The conclusion was drawn that in longer service no difficulties need be expected in this respect but the rapporteur thinks this to be a little optimistic. Nothing definite could be said about wear because of the short duration of the tests.—*Paper by Prof. Casci and Dr. Ing. Scirocco, read at the Internal Combustion Engine International Congress, Milan, 1953. Abstract in Gas and Oil Power, Vol. 48, May 1953; p. 120.*



Section through the new Büchi-Tosi centripetal turbo-charger mentioned in the paper by Prof. Casci and Dott. Ing. Scirocco



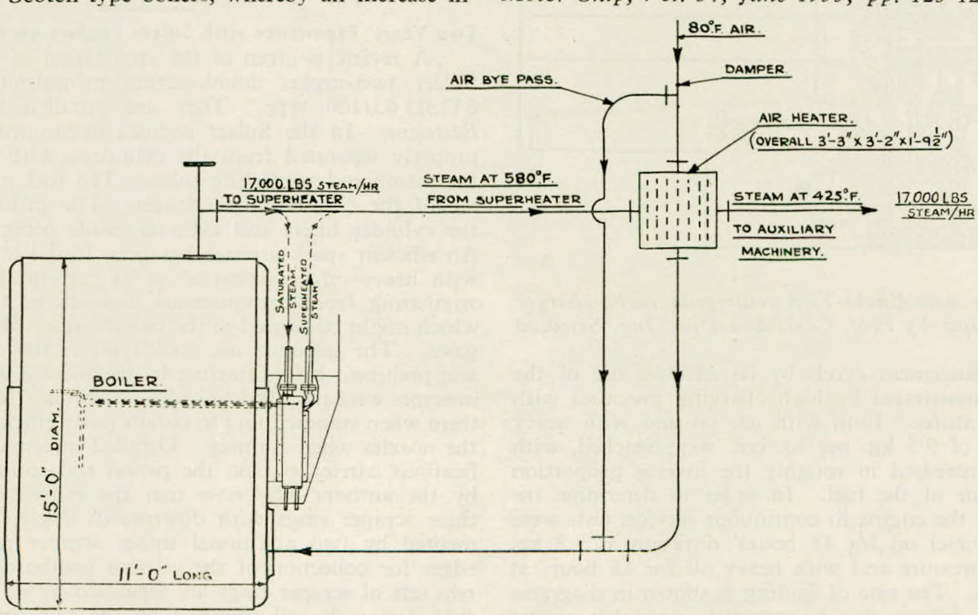
Transverse section through the Tosi Q34-type engine which is one of the test units referred to

Two Years' Experience with Sulzer Engines on Heavy Fuel

A review is given of the application of boiler oil to three Sulzer two-stroke, double-acting propulsion engines of the 6 DSD 60/100 type. They are installed in the motorship *Bastogne.* In the Sulzer engines mentioned the crankcase is properly separated from the cylinders, with stuffing boxes on the piston rod protecting tubes. The fuel pumps and nozzles are of the classic Sulzer design. The pistons are oil-cooled, the cylinder liners and cylinder heads being cooled by water. An efficient spark arrester has been found absolutely necessary with heavy oil operation so as to prevent fires on the bridge originating from carbonaceous deposits in the cylinder ports which might be carried to the outside atmosphere by the exhaust gases. The injectors are cooled when the engine is running and preheated before starting by means of a water circuit. The injectors were modified in order to circulate heated fuel through them when standing and to obtain really efficient cooling around the nozzles when running. Detailed information on the modifications carried out on the piston rod stuffing boxes is given by the authors who note that the normal units, comprising three scraper rings with downwards edges, have been supplemented by two additional upper scraper rings with upward edges for collection of the oil and combustion deposits. The two sets of scraper rings are separated by an intermediate piece that drains the oil collected by the two upper scraper rings, and fitted with a connexion to the scavenging air receiver. The stuffing boxes call for very efficient and careful maintenance. The nozzle hole diameters were decreased from 0·45 to 0·40 mm. for the upper nozzles. Those of the undersides were reduced from 0·40 to 0·375 mm. and from 0·45 to 0·425 mm. This resulted in less penetration of the sprays while no trouble was experienced through the increase in injection pressure. The pipelines for boiler oil on the engines were heated by steam lines insulated with the fuel lines. Each engine has an inspection box from the cooling water outlet of each injector so as to enable each valve to be checked individually. The injector needles have a clearance in the guide of 0·01 mm. instead of the usual value of 3 to 4 microns. The fuel pump plungers have the same clearance in their bushes in order to avoid sticking when warm. The clearance of the eccentric-driven push rod drives for the fuel pump suction valves has been modified to about 6 micron. There are no spill valves in the pumps. These considerable clearances cause rather heavy leakages if Diesel oil is used intermediately. The authors give warning that a rather high test pressure is required for the steam jackets of the filter bodies and the fuel preheaters in order to avoid the risk of cracking with consequent contamination by fuel oil of the boiler water circuit. In conclusion, it is stated that ash and unburnt fuel residues are more likely than sulphur to be the cause of difficulties with cylinders and pistons. The remedy to the present situation should be found in an improved preparation of the fuel, in more efficient ways of centrifuging and filtration. Generally, the cylinder wear of two-stroke engines doubles when boiler oil is used instead of Diesel oil. Trials with chromium-plated cylinder liners are to be made at a later stage.—*Paper by W. Sazanoff and P. Pluys, read at the Internal Combustion Engine International Congress, Milan, 1953. Abstract in Gas and Oil Power, Vol. 48, May 1953; p. 127.*

Increased Steam Efficiency for Motor Ships

With increasing steam requirements on many classes of cargo and oil-carrying motor ships, there is special interest in the proposals put forward by The Superheater Co., Ltd., to improve the superheat conditions for the auxiliary machinery and ancillary steam services. By such provision, the loss is avoided from the appreciable steam condensation which occurs with saturated steam, leading to the cost of feed water replacement by evaporation. Maintenance costs, too, can be high for the auxiliary machinery, owing to the expenditure required to make good the wear and tear on valves. Thus, many British and Continental owners have favoured the superheating arrangements for oil-fired Scotch-type boilers, whereby an increase in temperature of 50 degrees is given. However, it has been found difficult to maintain a constant superheat temperature owing to the wide variation in boiler load encountered in cargo ships, and more especially tankers. It is only possible to design a superheater to give a specific temperature at a given evaporation, and, as in the past it has been customary to base the design on the normal evaporation, the consequence has been that at the lower boiler loads little or no superheat is obtained. Many motor ships have their boilers arranged for forced draught with cold air, this being due to difficulties in fitting a gas/air heater in the limited available space. However, the new scheme evolved by The Superheater Co., Ltd., embodies a smoketube superheater which, at normal evaporation, will increase the steam temperature by 150-180 degrees, with the superheated steam passing through a steam/air heater. Usually, the size of boiler tubes is 2½ inches outside diameter and a superheater giving a 50 deg. F. rise can be fitted in these without restriction of the gas-free area. If the superheater is to have a greater heating surface, as is now proposed, the size of the boiler tube should be at least 3 inches outside diameter. In a typical installation with a boiler evaporating at normal load (17,000lb. of steam per hr.), the steam temperature of 580 deg. F. is reduced at the outlet of the steam/air heater to 430 deg. F.; at the same time the air temperature is increased from 80 deg. F. to 326 deg. F. A damper is fitted in the air ducting to the steam/air heater so that the final temperature leaving this heat exchanger can be maintained at 425 deg. F., irrespective of the boiler load, by controlling the amount of air passing over, or bypassing, the steam/air heater. For this installation, the overall dimensions of the steam/air heater would be 3ft. 3in. by 3ft. 2in. by 1ft. 9½in., and it would comprise a number of gilled tubes attached to inlet and outlet headers. Such a compact unit could be fitted in any position between the fan and the furnace fronts—an advantage in the normally restricted space in motor ships. The pressure loss on the steam side is about 3lb. per sq. in. and the draught loss on the air side about 1·2 inches. It is contended that 50 deg. F. superheat in the steam, maintained constantly, will give a saving in the fuel bill of about 5 per cent, and this can be maintained with the proposed arrangement at a constant figure of 425 deg. F., which meets the classification societies' requirements for allowing normal materials for the valves and steam lines. Further, this temperature, 425 deg. F., is maintained over the whole range of boiler loads, so that full economy is obtained at all times when the steam auxiliaries, heating coils and other services are in use, both in port and at sea.—*The Motor Ship, Vol. 34, June 1953; pp. 123-124.*



Arrangement of smoketube superheater and steam air superheater

ENGINEERING ABSTRACTS

Section 3. SHIPBUILDING AND MARINE ENGINEERING

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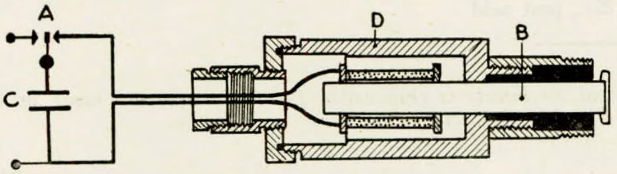
Some Notes and Results Relating to the Use of Heavy Fuel Oils in Marine Diesel Engines

The author states that the interest of the Diesel engine manufacturers in arranging for the use of boiler oil is to give the owners the best possible assistance in reducing operation costs and thereby further increase the advantage of Diesel engines compared with steam turbines. From the point of view of the engine builder it is held that normally boiler oil can be used with propulsion engines provided with suitable scavenging and fuel injection systems. A possible increase of maintenance work when burning boiler oil will generally be due to deposits and greater wear in comparison with operation on Diesel oil. In many cases it seems possible to reduce these difficulties through the cylinder lubrication. Auxiliary medium-speed engines will be more sensitive to boiler oils of poor quality than the main engines. The fuel valves of main and auxiliary engines of the B. and W. type are normally cooled with Diesel oil from a separate circuit when running on boiler fuel. Superior combustion properties and a simple fuel injection system, which is easy to adjust and to overhaul on board, are the first and foremost requirements for running on heavy fuel. Trunk-piston engines carry the risk of lacquer formation and abnormal sludging and coking, as well as contamination of the crankcase oil. These risks are avoided with crosshead engines when complete separation by piston rod stuffing boxes is used. The essential disadvantages of burning boiler fuel are generally the increase of cylinder liner and piston ring wear and breakage of piston rings. Piston overhauls should be easily effected and cylinder liners should be of a simple and not too expensive design. Liners should be easy to replace simultaneously with piston overhauls within an eight-hour working day. Even a doubling of the cylinder liner wear when using boiler fuel will not reduce the replacement period of the liners to less than 15,000 hours and will not decrease to an important extent the economic advantages of the burning of boiler fuels at the present price difference. It is quoted from a chief superintendent's report that the use of boiler fuel in a low-speed B. and W. two-stroke crosshead engine had not necessitated alterations to fuel valves, nozzles or fuel pumps. No signs of damaged exhaust valves were reported, and there was no abnormal contamination of the crankcase oil. Fuel consumption with boiler fuels of from 400-4,000 seconds Redwood I at 100 deg. F. was generally unaltered, but in some cases an increase of 5 per cent was recorded. Cylinder liner wear was from 0·4 to 0·8 mm. with an average of 0·5 mm. per 1,000 hours against 0·2-0·3 mm. per 1,000 hours on Diesel fuel, at an average mean indicated pressure of 6·5 kg. per sq. cm. Piston ring wear was also increased, but no sticking occurred; in some cases piston rings were found to be broken in several pieces. Chromium plating of cylinder liners was not found to work out to the same improving extent on cylinder wear as when running on Diesel oil.—*Paper by M. H. Andresen, read at the Second Internal Combustion Engine International Congress, Milan, 1953. Abstract in Gas and Oil Power, Vol. 48, May 1953; p. 127.*

Ultrasonic Apparatus for Descaling Boilers

In the "Crustex" boiler descaler, ultrasonic vibrations are generated and transmitted through the boiler water in order to remove scale. The descaling action depends upon the difference in the moduli of elasticity of the scale and of the steel beneath it and is effective in all parts of the boiler reached by the pulses of ultrasonic energy. The "Crustex" apparatus, which is an electrical device producing a mechanical effect, and is operated from the main electricity supply, was first developed and made in Switzerland—the British version of the product is made by H. N. Electrical Supplies, Ltd., London. The equipment consists of two parts, a generator and an oscillator for producing pulses of energy at 28,000 c/s, and directing them through the boiler water. The generator is a simple transformer and valve rectifying circuit which is housed in a dustproof metal case, 13 inches by 10 inches by 6 inches, and supplies pulses of direct current to an oscillatory circuit composed of a condenser and a solenoid. The oscillator is fitted to the boiler, and it takes the form of a nickel-plated brass cylinder 1½ inches diameter and 5 inches long, which contains a nickel tube surrounded by the solenoid of the oscillatory circuit. The generator is wall mounted at any convenient position up to 50 yards from the boiler, and is connected to the oscillator by flexible cable. A simplified sketch of the oscillator is shown in the accompanying diagram. The condenser (C) is charged from



Simplified diagram showing the arrangement of the oscillator

the rectifier circuit of the generator and is discharged via the cable through the solenoid surrounding the nickel tube (B). This tube is housed in the oscillator body (D), attached to the boiler in order to give the ultrasonic vibrations free access to the water in the boiler. Alternate charging and discharging of the condenser is effected by the mercury commutator (A). The pulses of current in the solenoid produce magnetostriction (contraction and expansion under the influence of a magnetic field) in the nickel tube (B). Damped pulses of vibration are thereby produced in the nickel tube, the frequency being 28,000 c/s. The tube is brazed at its mid point into the oscillator body (D), which is attached to the boiler by the threaded portion to some convenient point of the boiler. The damped oscillations are then transmitted from the free end of the nickel tube, which is closed by a metal disc, into the water. The point of attachment to the boiler varies with the type of boiler, and must be a point on the shell, end plates, or any point where the ultrasonic vibrations have free access to those parts of the boiler where scale forms. The ultrasonic vibrations dislodge the scale from the heating surfaces of the boiler when it reaches egg-shell thickness. Thicker layers of scale formed at inaccessible parts of the boiler prior to the installation of "Crustex" will also be dislodged, and will collect at the lower parts of the boiler. The heating surfaces are thus kept free from a thick layer of scale, but are given some measure of protection from corrosion by a thin layer of scale some few thousandths of an inch thick.—*Engineering and Boiler House Review, Vol. 68, June 1953; p. 188.*

French Fire Floats

The fire floats *Pythéas* and *Commandant-Filleau* were built by the Chantiers Navals Franco-Belges, one to be employed at Marseilles, the other at Bordeaux. The main machinery of each boat consists of a pair of non-reversible Diesel engines, each developing 180 h.p. at 1,500 r.p.m. These engines are arranged to drive either twin variable-pitch propellers through reduction gearing, or a pair of pumps, or both. The main centrifugal pumps can be operated either in parallel or in series; their combined output can be varied from 3,500 gal. per min. at 85lb. per sq. in. to 770 gal. per min. at 310lb. per sq. in. Four monitors are fitted on deck, one forward, two on top of the wheelhouse, and the fourth on top of a hinged mast with the nozzle 28 feet above water level when in the operating position. Hydrants to supply twenty-six hoses of different sizes are fitted. Two foam-generating installations are provided, one fixed on board to fight fires on the surface of the water, while the second comprises two portable foam generators. A battery of CO_2 cylinders with a total capacity of 2,000lb. of liquid CO_2 is provided. There are four portable submersible salvage pumps; these are driven by hydraulic turbines supplied with water under pressure by the main fire-fighting pumps. When the water supply consists of 185 gal. per min. at 85lb. per sq. in., each pump has a capacity of 370 gal. per min. against a head of 30 feet or 220 gal. per min. against a head of 60 feet. The principal dimensions of the floats are: length, o.a., 69 feet; breadth moulded, 15·4 feet; depth, 6·7 feet; draught, maximum, 6 feet; speed, 11 knots.—*Journal de la Marine Marchande, Vol. 35, 5th March 1953; p. 519. Journal, The British Shipbuilding Research Association, Vol. 8, April 1953; Abstract No. 7,374.*

Scientific Research in Shipbuilding in Holland

The author reviews shipbuilding research in Holland, which is divided between the Research Centre for Shipbuilding and Navigation, and the Dutch Shipbuilding Experiment Station. The results achieved include the work on the Wageningen propeller series, while the resistance tests being carried out on a series of geosims of a Victory ship, ranging in size from 1/60 to full-scale, are likely to produce important results. The subjects on which research is either in progress or envisaged in the near future include an investigation into the upper and lower limits desirable for the metacentric height; an investigation into the motion of ships at sea; the analysis of voyage reports; the provision of design recommendations for welded construction; investigations on corrugated bulkheads and bottom deformations; stress measurements in decks; the determination of the damage caused by vibration; and an investigation into heat-insulating materials. Future research programmes may include such subjects as light alloys as shipbuilding materials; the determination of the stress distribution in structures stressed beyond the elastic limit; the magnitude of the dynamic loads on a ship at sea; more accurate measurements of the frictional resistance and the surface roughness of ships; manœuvrability; gas-turbine propulsion; atomic propulsion; cavitation; and tests on full-scale ships to determine the useful thrust, the loading on the superstructure, and the motions at sea.—*H. E. Jaeger, Schip en Werf (1953), Vol. 20, 10th April; p. 180. Journal, The British Shipbuilding Research Association, Vol. 8, No. 5, May 1953; Abstract No. 7,500.*

Performance of Mariner Class Vessels

Sea trials held on the Mariners to date have shown them to be seaworthy and well designed. Extensive trials are being conducted on the first vessel out of each yard to test the ship and her machinery thoroughly. On all but the *Old Colony* the trials have been conducted at light draught, averaging about 12ft. 6in. forward and 23ft. aft (11,600 tons displacement). Fuel consumption was measured during a 6-hour economy run at normal power and a 2-hour run at 22,000 s.h.p. In addition, fuel meaurements of shorter duration were made on the *Old Colony Mariner* at three other points. The results of the tests to date are shown in Fig. 2. In order further to check efficiency, the water rate of both main and auxiliary turbines was measured at normal rating. The main unit was run non-extracting for one hour, and flow determined by either disc meters or by calibrated orifices. Flow meters were used for the auxiliary turbines. The water rates of both main and auxiliary units in each case met the specification requirements within the

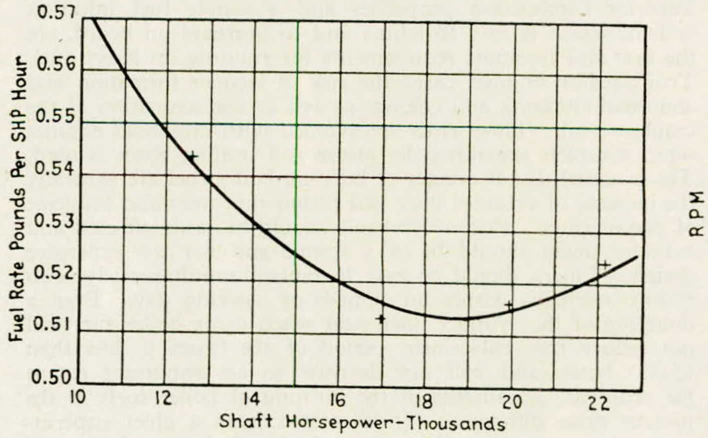


FIG. 2

limits of accuracy to be found during a test of this nature. The capacities of the evaporators, contaminated evaporator and boilers were measured to ensure that they meet design requirements. All successfully passed with ample margin. Severe tests were applied to the steering gear and anchor windlass to prove their adequacy for all service conditions. Steering tests were conducted at 22,000 s.h.p. on each ship, and the speed at which most of the units responded was remarkable; the average time from hard-over to 30 degrees was only 22 seconds. Anchor tests were held in 60 fathoms of water on the first ship out of each yard, including a free fall between 45 and 60 fathoms in order to check the ability of the brake to operate properly under extreme load. After proving the design and installation on the first ship, 30 fathoms anchor tests were felt sufficient on the remaining vessels. Several methods of determining tactical diameter were tried: cross bearings, range readings, and radar trace. The results so far have been disappointing as only three right and left circles have plotted at all and the accuracy of even these diameters is felt to be somewhat open to question. The average tactical diameters measured 965 and 900 yards respectively. The ships seem to close a little faster in a left circle, averaging 5 minutes 11 seconds as compared with 5 minutes 29 seconds for a right circle. Z-manœuvres were carried out in the standard manner. There was little difference in the total manœuvring time among the various ships, the average being 3 minutes 37 seconds. Considerable differences were found in the crash stop manœuvres, however. The time to "crash" a ship and the reach obtained is not only a function of the characteristics of the astern turbine and the general wind and sea conditions, but it also will vary in accordance with the practices of the operating engineer.—*Marine Engineering, Vol. 58, May 1953; pp. 111-113.*

Napier Engine with Exhaust Gas Turbo and Mechanically-driven Blowers

In Fig. 1 is illustrated a blower system for an engine of the Deltic type, with three crankshafts (A) in casings (B), connected by blocks of water-cooled cylinders (C). Scavenging air is supplied by mechanically-driven and exhaust gas turbo-blowers, a hydraulic clutch being provided between the two machines. When the engine is in operation, the exhaust gas passes through the manifolds (D) to the turbine (K), which drives the shaft (N) and the impeller (F) of the first stage of a centrifugal compressor through the clutch members (G, J). Assuming the hydraulic coupling (O) to be filled, any power developed by the turbine (K) in excess of that required to drive the impeller (F) of the first stage will be transmitted to the shaft (M) of the impeller (H) forming the second stage. This power is either sufficient only to assist in the driving of the impeller, or it may also transmit the torque through the gearing (L) to the crankshaft of the engine, so as to add to the power delivered to the propeller shaft (S). In this manner, the impeller (H) is driven from the crankshafts through two parallel transmission lines, each including torsional flexible shafts (R). These shafts ensure an even distribution of the load and prevent over-stressing of the gearing (P) during rapid acceleration or deceleration of the engine. One part (T) of the coupling (O) is connected to the tubular shaft (M), while the other part (U) has a tubular extension (V) to which is connected the splined member (W). A similar coupling (J) is mounted on the shaft (N) and the splined rims of the two couplings engage corresponding internal splines in the tubular coupling (G) secured to the turbine rotor (E).—*Patent No. 689,795. H. Sammons. D. Napier and Son, Ltd., London. The Motor Ship, Vol. 34, June 1953; p. 132.*

Tank Cleaning Vessel

A recent arrival on the River Tyne is the converted light escort craft *Tulipdale,* of about 459 gross tons, which is equipped for the British Wheeler process of tank cleaning. This vessel will be based on the Tyne and will cover the whole of the north-east coast. Self-propelled, the ship can proceed to any port in the United Kingdom or near Continent and, being of a handy size, can manœuvre comfortably in congested areas. The *Tulipdale* is equipped to deal with any oil, including crude, having a flash point above 150 deg. F. and is available for the cleaning and gas-freeing of oil fuel tanks in the double bottoms and elsewhere, and deep tanks which have been used for the carriage of edible oils. The vessel, equipment and precautionary methods have been fully approved by Lloyd's Register and the harbour authorities, and it is emphasized that

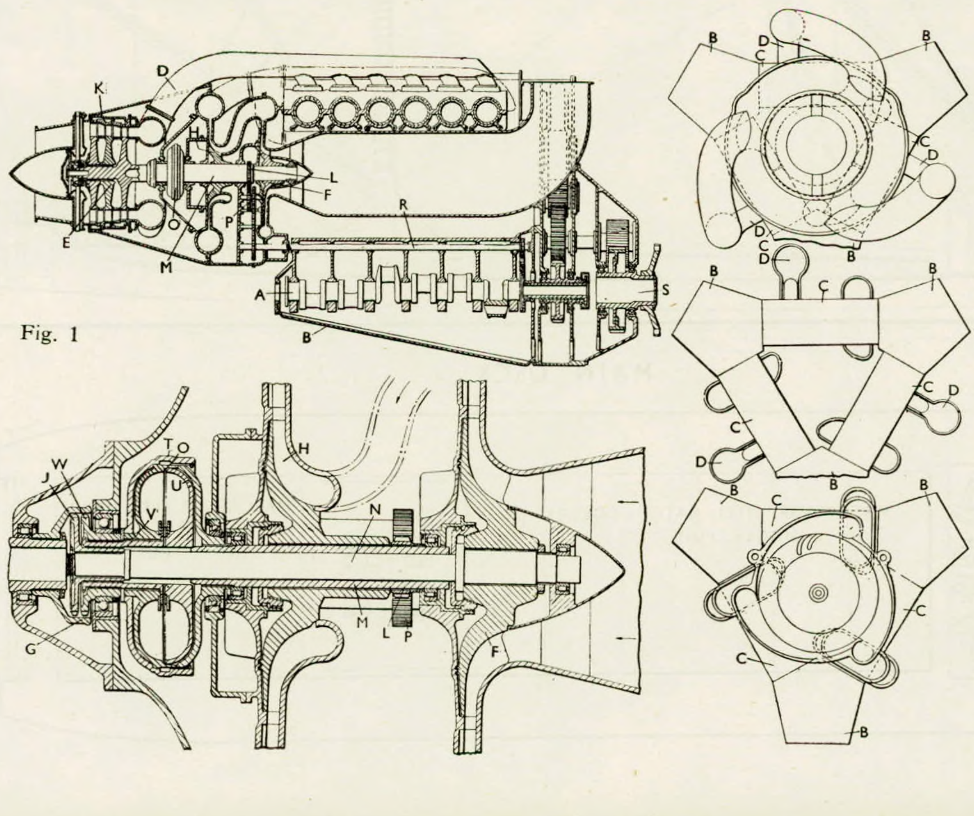


Fig. 1

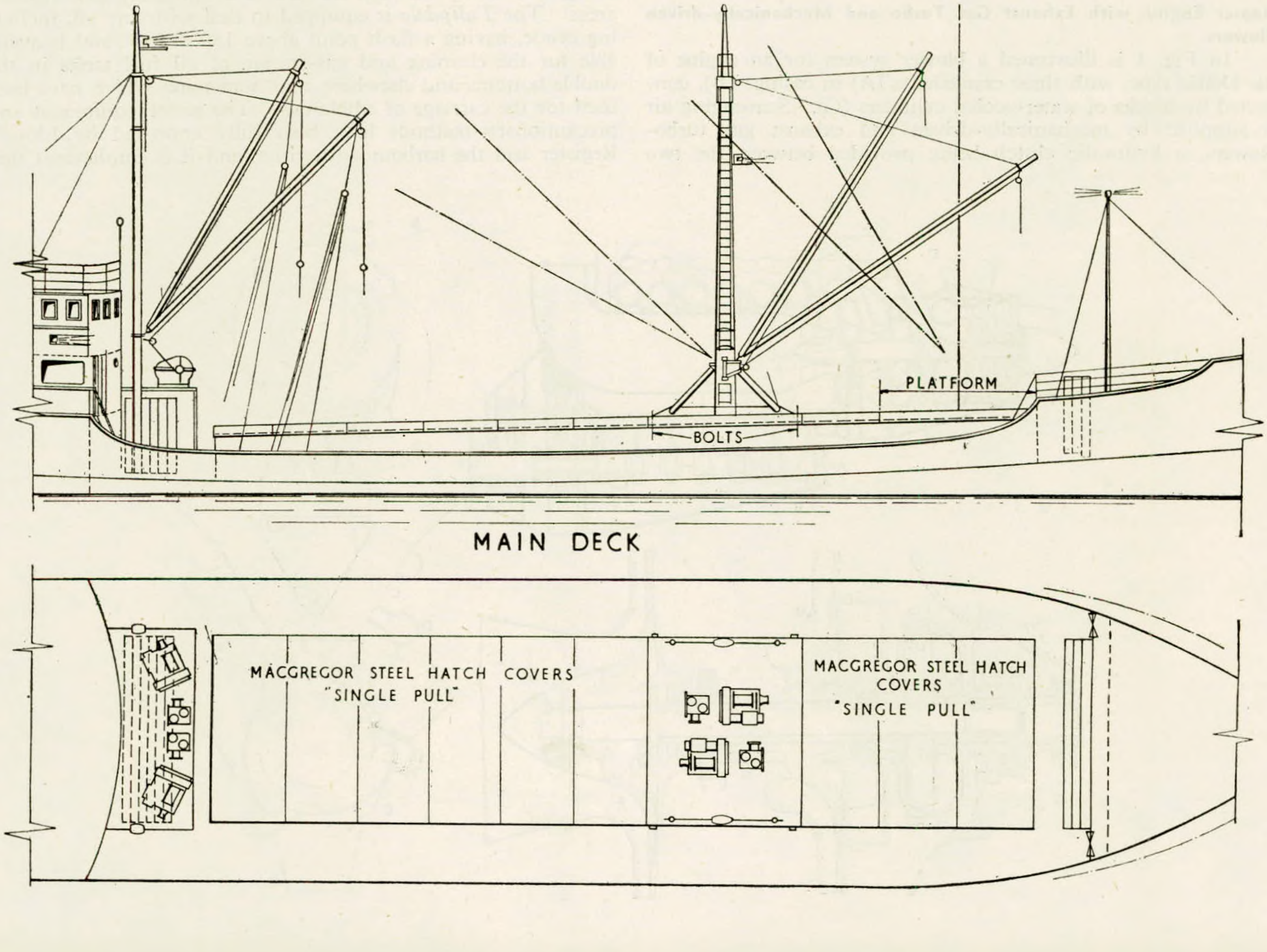
in dealing with passenger liners, the fire hazard is reduced to the minimum. With the suction method employed in this process, the tank cleaning work can be carried out much more quickly than by normal methods and is, furthermore, a cleaner operation. The twin vacuum pumps fitted on this vessel are capable of pumping through a pipe line up to a quarter-of-a-mile in length. The steadily increasing number of motor ships being equipped to burn boiler oil has contributed largely to the demand for this service, and it is contended that there is a saving in time of some 70 to 80 per cent over the usual method of hand cleaning. The *Tulipdale* has a service speed of 11 knots and carries a crew of twelve, excluding the tank cleaners, and has six sludge tanks forward having a total capacity of 200 tons.—*The Motor Ship, Vol. 34, June 1953; p. 97.*

Bipod on Travelling Platform

A novel arrangement of cargo-handling gear is fitted in the motor vessel *Muphrid N,* owned by Van Nievelt, Goudriaan and Company's Stoomvaart Mij., of Rotterdam. This ship, a vessel of about 500 tons gross, has a single hatch 90ft. 2in. in length, with a breadth of 16ft. 5in. The hatch is covered by MacGregor "single pull" hatch covers operated from either end, and when closed these link up in the centre with a platform which moves along the hatch coaming on wheels. On this platform is mounted a bipod mast carrying two derricks, together with the two associated winches. Four wheels on the platform can be lowered on to the hatch coamings by means of jack screws, and the platform can then be warped along the hatch by means of the winches. These winches are of a self-contained Diesel-driven type, the use of which avoids the necessity of trailing electric leads or other connexions between the platform and the ship. When cargo is being handled the platform is lowered on to the coamings, and secured in position by means of rigging screws. In operation the platform can be moved to the position most convenient for handling any particular part of the cargo. Presumably it will normally be kept in the centre of the hatch, but can be moved right to one end or the other when long loads are handled. It is supplemented by two other derricks mounted on samson posts abaft the hold, just forward of the bridge. This arrangement of a moving platform is dependent on the use of a bipod mast, as a mast of the normal stayed type could obviously not be accommodated on the platform.—*The Shipping World, Vol. 128, 27th May 1953; p. 491.*

Experiments with Tanker Models

This paper describes some experiments with models of a single-screw tanker with a displacement of 22,000 m.^{3} and a designed fully loaded trial speed of about 15 knots. The tests were intended to form a systematic investigation into the effect of the shape of fore-end sections on the performance qualities of the hull. Originally, the experiments were planned bearing in mind the different prevailing opinions about suitable fore-body lines for this type of ship, but account was also taken of the further knowledge which has been obtained in recent years concerning the character of the boundary layer on models with full entrances. Thus, it has become apparent that there is a risk, particularly with a full model, of laminar flow occurring along the forward part. The usual methods of calculation assume turbulent flow over the *whole* model and it is, therefore, with full models that there is the greatest risk of error in computing the ship resistance. It has also been discovered that laminar flow is more stable when there is a pronounced negative pressure gradient, i.e. pronounced acceleration in the flow. These facts, when applied to the relative qualities of models with different types of entrance, can be taken to imply



that full models with extreme V-form forward sections are more liable to laminar flow than corresponding models with more U-form sections. In view of the above statements, it seems probable that in earlier model experiments, before the effect of the boundary layer on large models had been fully appreciated, the qualities of V-form sections for full entrances were overestimated in comparison with those of more U-form sections. At first hand, this applied, of course, in such experiments where artificial turbulence stimulation was not provided. In the tests in question, a systematic comparison was made on the basis of both resistance and propulsive qualities between forward sections of V- and U-form, all the models being fitted with turbulence stimulators. Four model versions were investigated; one with pronounced V-form forward sections, one with pronounced U-form forward sections and two intermediate forms. In addition to the original programme, however, tests were made with the *Extreme* U model fitted with a bulb and the *Extreme* V model with a somewhat modified stem contour. The experiments were carried out at the Swedish State Shipbuilding Experimental Tank in Göteborg.—*H. Edstrand, E. Freimanis, and H. Lindgren, Publications of the Swedish State Shipbuilding Experimental Tank, No. 23, 1953.*

Bolts Under Shock Loading

An investigation into the properties of bolts subjected to the types of shock likely to be met with on combat-type naval vessels is being carried out by the Naval Research Laboratory, Washington at the request of the Bureau of Ships. The preliminary results given in this paper are for bolts of S.A.E. 1020 cold-rolled steel. It was found that the physical properties of these bolts under conditions of shock were better than might have been expected from the results of static tests. Owing to the work-hardening that occurs under dynamic conditions, there is more general strain before necking begins, which gives rise to greater elongation in long bolts; this indicates that long bolts will absorb about twice as much energy under dynamic conditions. For short and straight-shank bolts, little difference was observed between the elongation under dynamic and under static conditions. The total elongation to fracture is roughly constant for a given type of bolt, regardless of the number of shock cycles to fracture. In straight-shank bolts, necking and considerable strain always occur in the threads, and the resulting distortion makes it difficult to retighten the bolts after relatively small strain; with reduced-shank bolts retightening is simple, since necking always occurs in the shank. The initial tension of holding-down bolts should be enough to prevent motion under steady-state vibrations; but where the storage of elastic strain energy is small compared with the energy absorption for plastic flow, the bolts should be tightened to nearly the value of yield. A minimum of tightening is best for a high-tensile material of low ductility. Stress raisers, such as scratches, threads, and fillets, are not as important for shock as for the endurance limit associated with fatigue. This conclusion might not apply to materials liable to brittle fracture under impact.—*H. M. Forkois, R. W. Conrad, and I. Vigness, Proc.Soc.Exp. Stress Anal. 10, No. 1, 1952; p. 165. Journal, The British Shipbuilding Research Association, Vol. 8, No. 5, May 1953; Abstract No. 7,555.*

Effective Flange

In the analysis of the bending of beams with flanges large in comparison with the span, a knowledge of the effective width of the flange is of great importance. The stress in the flange will be greatest over the web, and decrease towards the edges. For simplicity in design, it may be considered that a constant stress equal to this maximum value acts over a width of flange reduced in such a manner that the total force remains the same in the section under consideration. This reduced flange is known as the effective flange, and the concept is of particular use in the design of stiffened panels and sections with asymmetrical flanges. The author gives a theoretical treatment of the problem, and presents formulæ for calculating the effective flange of beams with both symmetrical and asymmetrical flanges. In the case of asymmetrical flanges, the forces from the web will act outside the neutral axis of the flange, and bending in the plane of the flange occurs. But for the stiffness of the web, the effectiveness of the flange (especially in the extreme cases of channel and Z sections) could be reduced to a value of about 0·25. The usual sections have values a little higher than this. A treatment is given of stiffened panels, and particular attention is paid to the effect of buckling and initial deflexion upon the effective flange.—*J. R. Getz, Norwegian Inst.Ship.Res.; Report No. 1, November 1951. Journal, The British Shipbuilding Research Association, Vol. 8, May 1953; Abstract No. 7,485.*

Explosion Bulge Test

In the N.R.L. Explosion Bulge test for investigating the structural performance of welds in ships' plates, only simple equipment is required. Controlled stress and strain conditions are developed in a model structure of the simplest possible initial geometry—a plate which is loaded by the application of the uniform gas pressure resulting from the explosion of a fixed charge of pentolite, while the plate is supported in a die. The bulging produces a biaxial stress, which is balanced in the case of a circular die and unbalanced when an elliptical die is used. The charge is offset from the test piece to ensure uniform loading and true bulges, of controlled geometry, while reducing the "slap effects" (brisance) to a minimum. The size of the bulge is adjusted to include the weld, the heat-affected zone, and an equal width of base plate. Susceptibility to the development of brittle cracking in welds is tested by deposition on the plate of a short bead of brittle surfacing weld metal, which is then notched to half its thickness by means of an abrasive wheel. Thus, when the plate is bulged by the explosion, the metal is presented with a sharp material crack at the instant it is loaded to the yield point. Brittle cracks propagate by release of elastic strain in the material. In the bulge test, the rapid transmission of load by the gas pressure causes the elastic strain to be replaced as rapidly as it is lost in feeding the advancing crack, so that the high potential elastic energy of large structures, such as ships, is simulated. Surface shear and the nature of the cleavage fracture affect the rate of crack propagation. In ship fractures, the maximum surface shear is 0·01-0·02 inch, but in the bulge test, it reaches 0·05 inch, owing to the rapid replacement of strain energy. The temperature at which this critical surface shear (0·01-0·02 inch) is developed is defined as the "fracture transition" temperature. Above this temperature, the amount of shear is too great for free-running cracks to occur in ship structures; below this temperature, crack running occurs with little absorption of energy since the surface shear approaches zero with decreasing temperature. At a lower temperature, the "ductility transition" temperature, plastic deformation is not necessary to initiate the running of the crack. These transitions are of fundamental importance in structural design. Structures operating entirely in elastic loading do not develop the amount of deformation needed to trigger a brittle crack at temperatures above the ductility transition, and would, therefore, remain safe, despite serious flaws, so long as the service temperatures remained above the ductility transition temperature. If the service conditions could give rise to plastic deformation, the structure would not be safe at temperatures below the maximum temperature of crack propagation.—*W. S. Pellini, U.S. Naval Research Laboratory, Report No. 4,034, 4th September 1952. Journal, The British Shipbuilding Research Association, Vol. 8, April 1953; Abstract No. 7,387.*

Unusual Tank Vessel Corrosion

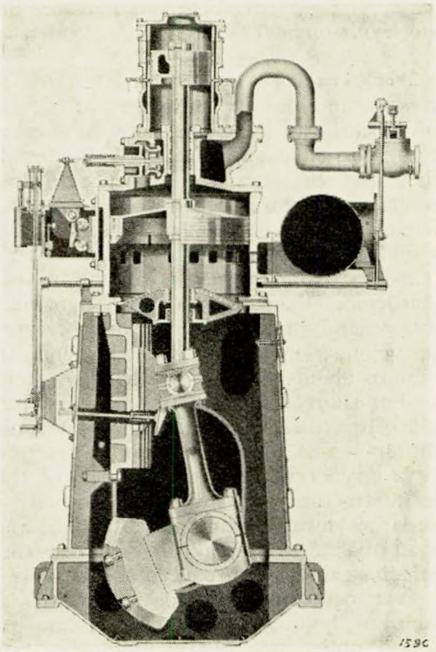
An unusual type of localized corrosion attack has recently made its appearance on relatively new tank vessels in exclusive crude oil service, both coastwise and transoceanic. The fact that this occurs with crude oil service is noteworthy in that crude oil, even sour crudes, is not normally considered to be of serious corrosivity. It is of interest also that attacks of equivalent degree are being reported both by foreign and domestic tanker operators. The writer has observed this corrosion on several vessels and finds that it appears as discrete pits of continuous chains of pits in the lower reaches of the cargo tanks and in every case on the top of exposed surfaces of the piping and structural members, e.g. heating coils, cargo lines, transverse and longitudinal members, etc. In an attempt to find the cause of this corrosion, investigation was made of the many possible contributing factors, such as type of cargo, ballast operations, relative times in cargo and in ballast, cleaning procedures, etc. Several of these factors appeared to be significant. 1. Serious pitting occurred in tanks alternately carrying cargo and ballast and not in tanks carrying cargo only. This indicated that the attack was associated with the sea water ballast phase. 2. A close inspection showed that pitting occurred not only on the top surfaces of the various members described above but actually was oriented—attack was slightly to one side or the other of centre, the nearest side—with respect to the location of the cleaning nozzles (these nozzles deliver revolving streams of high pressure, high temperature salt water which is used to clean the tanks between cargoes). This established that the cleaning operation was tied in with the corrosive attack. Most severe attack occurred on vessels most frequently cleaned. In way of explanation, independent carriers because of change in cargo or change in ownership of cargo find it necessary to clean cargo tanks after practically every trip. By contrast, vessels on long term charter to a single oil company may find it necessary to clean only two or three times a year and then primarily to avoid build-up of heavy residues. Maximum penetration of the pitting attack was $\frac{5}{16}$-inch and this was observed in just under three years. In this respect, it should be noted that the ship's log showed that the service life was divided roughly into two-thirds time in cargo and one-third time in sea water ballast. The writer considers the provable mechanism of attack to be as follows: Steel in tanker service is normally covered with scale—this can be mill scale at the outset but after varying intervals of service life may well be a combination of mill scale and/or rust scale which develops as a result of alternate exposure to cargo and ballast. Characteristic of such scaled surfaces are "breaks" in the scale which may be present at the outset or develop as a result of "working" of the vessel. When exposed to crude oil only or to alternate exposure of crude oil and sea water ballast without cleaning, such surfaces suffer no unusual corrosion. However, when these same surfaces are freed from protective oil or residues by cleaning, thereby exposing small areas of bare steel and large expanses of cathodic scale, and the tanks are then ballasted with sea water, galvanic attack occurs at the "breaks" in the scale as influenced by the surrounding areas of cathodic scale. In this latter respect, measurements showed substantial current flow between pits and surrounding areas of scale, which bears out the galvanic nature of the attack. In way of summary, it is suggested that the observed pitting attack occurs during the clean ballast period only and is accelerated by a galvanic couple between scaled and unscaled steel with the attack concentrated at the small anodic areas of bare steel.—*L. M. Mosher, Corrosion, Vol. 9, May 1953; pp. 1-2.*

Dynamometer for Admiralty

One of the most important pieces of equipment installed in the new Admiralty test house at the National Gas Turbine Establishment, Farnborough, Hants., is a highly specialized form of hydraulic dynamometer, manufactured by Heenan and Froude Ltd., of Worcester, for development testing of marine gas turbines. In view of the unusual combinations of powers and speeds provided by gas turbines, the machine had to meet an exacting specification. The requirements were for a maximum of 10,000 s.h.p. at a speed of between 575 and 1,100 r.p.m. in each direction of rotation. This double-ended, reversible, hydraulic dynamomenter is capable of being sited, as required, on the test-bed. Remote control of the back-pressure valves, which regulate the load, is arranged from the control room, and hydraulic weighing gear also enables the torque to be measured from the control room. A feature of the weighing gear is an oscillator, which prevents inaccuracies in readings caused by sluggishness in the transmitting and receiving pistons and in the long lengths of connecting piping. Means are provided whereby test loads can be applied to the dynamometer casing to check the accuracy of the torque readings. Speed readings are given electrically. The dynamometer can operate with its shaft at any angle up to 11 degrees from the horizontal. With this arrangement, it is possible to test engines, intended for the propulsion of small ships, with their shafts raked to the angle at which they will be installed.—*The Shipbuilder and Marine Engine-Builder, Vol. 60, July 1953, p. 459.*

New Unaflow Steam Engine

The Skinner Unaflow steam reciprocating engine, developed and manufactured in the United States by the Skinner Engine Company, has proved a popular engine in North America despite the competition of the steam turbine. It is manufactured under licence in Canada by Canadian Vickers, Ltd., in addition to being built by the Skinner Company. In recent months licences to build the Skinner engine have been taken



Cross section through the new steeple-compounded Skinner engine

out by two European shipyards, one in France and one in Holland. In this new engine, steam can be employed at the very high conditions for steam reciprocating machinery of 440lb. per sq. in. and 740 deg. F. It has been employed to propel two train ferries which have recently been completed for service on the American Great Lakes. These two ships, *Spartan* and *Badger,* were built by the Christy Corporation, of Sturgon Bay, Wisconsin, for the Chesapeake and Ohio Railway Company. They are vessels of 7,920 tons displacement each, designed to accommodate thirty-two 40-ft. freight cars with an average load of 80 tons each. They have a length overall of 410ft. 6in., breadth moulded of 59ft. 6in. and depth to main deck of 24 feet. Each ship is propelled at a speed of 15½ knots by an 8,000 s.h.p. steam reciprocating power plant, consisting of two Skinner Unaflow engines fed from four Foster-Wheeler D-type coal-fired boilers. The engines operate on the Wolff cycle, in which there is no steam receiver between the cylinders, the exhaust valve of the h.p. acting as the inlet valve of the l.p. cylinder. The transverse section through one pair of cylinders shows the general construction and design features of these engines. The base and frame, which constitute the enclosed crankcase, are rigid weldings patterned after the design normally used on the more familiar Unaflow engines. The unit assemblies of the h.p. and l.p. cylinders are mounted individually on the frame, after alignment, and are free to expand separately without imposing any thermal stresses on any of the engine parts. The two cylinders are arranged on each crank in steeple formation with the h.p. above and the l.p. below. A single cylinder head serves the two cylinders and is fitted with three Skinner expansion-compensating steam-tight poppet valves, the transfer valve being the one shown in the illustration. The combination of both cylinders on the one crank results in a double-acting cycle which is equivalent to the simple "Unaflow" engine. Therefore the steeple compound may be built with any number of cranks all having equal reciprocating weight and optimum balance of the inertia forces. The engines in the *Spartan* and *Badger*, for example, each have four cranks and may be termed "four-cylinder" although each crank has a high-pressure and a low-pressure cylinder. The h.p. cylinder liner is made of steel which is electro-plated with porous-chrome to provide a hard, oil-retaining surface. Both the h.p. and l.p. piston rings are of the sectional type for a medium of lubrication. As an additional precaution, copper-lead bearing metal rings are permanently fitted to the pistons to burnish the cylinder walls and to prevent contact with the piston body. As the steeple compound is designed to run at high temperatures, low-pressure steam is used as a cooling medium for the critical parts. This is the secondary purpose of the annular space round the outside of the h.p. cylinder liner, the l.p. steam reducing the liner wall temperature to a point at which lubrication is readily accomplished. The heat extracted from the liner in this manner is not wasted, as it is recovered in the l.p. cylinder. The engine is therefore, in effect, a type of reheat engine.—*The Shipping World, Vol. 128, 24th June 1953; pp. 581-582.*

Roller Bearings

Preliminary investigation of the possible advantages of antifriction bearings for marine lineshafting was conducted in Germany in 1930. A simple and apparently fairly reliable torquemeter was developed, and power losses of from 5 to 6 per cent in lineshaft plain bearings were found. Following this, the Hamburg-American Line, in 1933, applied one spherical roller bearing to the motorship *Milwaukee*, on the 15½-in. diameter propeller shaft. Satisfactory experience with this and following complete installations has led to the building of 192 ships in Europe, between 1934 and 1947, using spherical roller bearings on the lineshafting. These ships had shafts ranging in size from 3½ inches to 18½ inches in diameter, the greater part being in the size range from 9½ inches to 15½ inches The spherical roller bearing, being inherently self-aligning, seems especially well suited for marine use, where alignment cannot be maintained. The very high load capacity of these bearings is also advantageous, not only for the extremely long life which should be realized, but also to cope with transient heavy load conditions resulting from the weaving of the hull. From the standpoint of fatigue strength, spherical roller bearings have capacity to show anticipated lives in marine service measured in hundreds of years. Concerning stern tube bearings, antifriction bearings were first used in Europe in 1936. In 1936, the motorship *Wuppertal* was built using spherical roller bearings in the stern tubes. The shafts have ranged in size from 7¼ inches to 15½ inches. The use of antifriction bearings in the stern tube is attended by several desirable consequences. The conventional bronze sleeve may be eliminated, since the surface of the shaft no longer is a wearing surface, and since the shaft, now being surrounded by oil instead of sea water, is no longer exposed to corrosion. Bearing clearance is quite small to start, and does not increase during operation, thus reducing or eliminating tailshaft vibration. Maintenance should be decreased, since the periodic renewal of plain stern bearings is no longer required. The stern tube itself no longer need be a strength member, since the shaft and propeller loads are, with antifriction bearings, concentrated in the stern eye and at the forward end of the stern tube. Consequently, the stern tube need be only an oil-tight pipe.—*Paper by W. L. Aiken, abstracted in S.N.A.M.E. Bulletin, Vol. 8, June 1953; pp. 42-43.*

New Swedish Diesel Engine

The official workshop tests of a new type of Diesel engine for the Swedish Navy have recently been completed at the Götaverken shipyard and the engine is now being installed in a ship. The engine operates on the two-stroke cycle and is of the opposed-piston type with two crankshafts. The lower crankshaft is directly connected to the propeller shaft by an hydraulically operated friction clutch and the upper crankshaft is geared to the lower crankshaft by a spur gear-train. Scavenging and pressure-charging of the engine is accomplished by an exhaust gas driven turbo pressure-charger. The new Götaverken design is known as the T.O.P. (turbo-charged opposed piston) engine and is directly reversible with a normal continuous full load rating of 2,500 s.h.p. at 920 r.p.m. The maximum rating is 3,000 s.h.p. at 975 r.p.m. and during the official tests the engine was run for a period at a rating of 3,100 s.h.p. The design is entirely self-contained, with direct-driven pumps for sea water and fresh water, engine lubricating oil, lubricating oil for the propeller shaft clutch and fuel oil feed supply. Starting and reversing of the engine is accomplished by means of compressed air. The engine is a ten-cylinder in-line unit with a cylinder bore of 7·087 inches and a stroke of 9·055 inches for both top and bottom pistons; the total swept volume is 7,138 cu. in. At the maximum rating the piston speed is 1,472ft. per min. and the mean effective pressure is 168·3lb. per sq. in. The upper pistons control the exhaust ports and the lower pistons control the scavenging ports. At low engine load the power developed in the turbine is insufficient for compressing the necessary amount of scavenging air and additional power must be supplied to the turbo-compressor unit. This power is transmitted from the upper crankshaft of the engine by an auxiliary drive consisting of bevel gears with free wheel arrangements so that additional power is transmitted when running astern as well as when running ahead, with the turbo-compressor running in its normal direction of rotation. At the normal rating of 2,500 s.h.p., and at higher loads, the power developed by the turbine is sufficient for driving the turbo-compressor and the unit consequently runs free without taking any power from the Diesel engine. At the maximum rating the turbo-compressor unit runs at about 16,000 r.p.m. and the pressure of the scavenging air is 23lb. per sq. in. gauge. The exhaust gas pressure before the turbine is then 15lb. per sq. in. gauge. Between the compressor and the Diesel engine an intermediate air cooler is fitted cooling the compressed air to about 105 deg. F. at the normal rating and to about 130 deg. F. at the maximum rating.—*Gas and Oil Power, Vol. 48, June 1953; pp. 145-146.*

Corrosion of Fuel Injection Nozzles

The British Internal Combustion Engine Research Association at Slough recently received a request for assistance from a member company which was confronted with a serious problem in connexion with engines of a modified design. A considerable number of these engines had been sold mainly to overseas customers. After they had been in service a short time, reports were received of rapid nozzle failure due to loss of metal from the wall of the nozzle body resulting in many cases in discharge of fuel through the nozzle wall. Most of the affected engines were operating on class A fuels. It was later learned that the same trouble has been experienced in varying degrees of severity in many engines of other makes. It was ascertained that the temperature of the surface exposed to combustion products is a factor of first importance in this type of corrosion problem. Consequently, when two examples of the affected fuel injection nozzles were received at the laboratory, attention was given to the possibility of the surface being overcooled. The surfaces of the nozzles had many interesting features, but the most significant was judged to be the complete absence, in all cases, of attack at the discharge end of the nozzles. This end, being subjected directly to the heat produced in the combustion chamber, is at a high temperature while the end adjacent to the cap-nut is shielded from the heat and cooled via a short heat-flow path. It was considered very probable that the attack was due to the temperature of the nozzle surface remote from the combustion chamber being too low and it was therefore suspected that the nozzles were being cooled more effectively than usual. A brief investigation sufficed to show that this type of nozzle corrosion can be eliminated simply by a suitable increase in nozzle temperature. Raising the cooling water temperature is normally an effective method of achieving this. Where the use of cold water cannot be avoided, the cylinder head should be designed to ensure that a moderate nozzle temperature is attained shortly after starting the engine. On the other hand, it is well known that excessively high temperatures lead to other nozzle troubles. —*W. P. Mansfield, Gas and Oil Power, Vol. 48, June 1953; pp. 147-149; 153.*

Cylinder Wear in Diesel Engines

This lengthy paper is based on a study of the available literature on the subject of cylinder wear in Diesel engines. The author begins with a general section in which he discusses methods of investigating and measuring cylinder wear, and the nature and magnitude of normal cylinder wear. He then gives a survey of the various causes of wear, which he classifies under the following headings:—*Rubbing or Scuffing Abrasion.* This is due either to abrasive particles introduced into the cylinder in the fuel oil, lubricating oil, or air; or to particles formed in the cylinder, as, for example, carbon or recrystallized aluminium oxide. *Corrosion* is due to various organic and inorganic substances present in the fuel. *Erosion and Surface Disintegration.* The author then discusses the various structural factors that influence wear, namely the effect of the type of engine, fuel injection, scavenging systems, loading, bore and stroke, revolutions, compression ratio, type of material used and methods of working, and the form of liners and piston rings. The influence of conditions of operation are then considered, particularly fuels and conditions of combustion (with mention of the importance of correct injection for oils of different viscosities), methods of lubrication and lubricating oils used, and temperature of operation. It is pointed out that the use of a large stroke-diameter ratio may cause increased wear, since, if the stroke is increased beyond a certain limit, the lubrication in the cylinder will be affected. It is also stated that the duration of burning increases with increasing cetane value of the fuel, with the result that the lubrication is affected over a greater proportion of the stroke, causing more wear. For this reason, it is no disadvantage to use a low cetane value fuel. The paper concludes with remarks on the possibility of research to reduce wear in Diesel engines by improved fuels and other methods.—*J. B. Gulbrandsen, Norwegian Inst. Ship Res. Report No. 3, January 1953. Journal, The British Shipbuilding Research Association, Vol. 8, June 1953; Abstract No. 7641.*

Combustion in Marine Boilers

In the discussion of furnace design factors, the author presents data for comparative purposes only, on furnace temperatures, heat-liberation, and heat-absorption rates. These should not be regarded as either upper or lower limits. Each set of operating conditions must be carefully analyzed to determine what the permissible limits are for the design under consideration. Operators are aware of the increase in fireside deposits which have made operation at high rates, over long periods of time, a matter of considerable difficulty. Some of these are due to improper burner adjustment. In the majority of cases, however, the cause is traceable to place of origin of the fuel oil, although variations in analyses and blending are also contributing factors. Considerable research is now under way to establish effective and low-cost means for avoiding such fireside deposits. Some suggested remedies seem to have merit but require time-tested operating experiences, under many differing conditions, to establish their real worth. The performance of oil burners available for marine applications is good. Modifications should, however, be developed to further increase their flexibility. The wide-range nozzle is of recent origin and has served to improve somewhat flexibility of operation. The next step indicated is a change in register design. At present these are virtually strait-jackets in that a possibility for regulating or changing flame contour is practically non-existent. The register should actually be of such design as to permit narrowing or spreading of flame, at will, by the operator. This is a matter of varying longitudinal component of air flow to increase or decrease its spin. With this sort of built-in flexibility the furnace could be more effectively used over wider ranges in capacity.—*Paper by O. de Lorenzi, abstracted in S.N.A.M.E. Bulletin, Vol. 8, June 1953; p. 37.*

Modern Ships' Propulsion

From the moment the Diesel motor had definitely established its position, the maximum power developed by this type of engine obviously became an important factor. For the single-screw slow-speed Diesel engine this maximum power normally lay at 6,500 b.h.p., for the reciprocating steam engine at about 2,500 s.h.p., and for the steam turbine at about 10,000 s.h.p. When, after the war, the speed of ships was considerably increased, this affected the required engine power and the maximum output reached about 12-14,000 b.h.p. per shaft for Diesel propulsion, about 4-5,000 s.h.p. for the reciprocating engine, and about 15,000 s.h.p. for the steam turbine. Where the ship is driven by more than one propeller, the power for the steam turbine per shaft may even attain 50,000 s.h.p. and more. This more or less spectacular development has, so far as the Diesel engine is concerned, resulted from a considerable increase in mean pressure, and an improvement in the scavenging system, etc., while for steam reciprocating engines and steam turbines, the improvements have been mainly achieved by increase of steam pressure and temperature, as well as heat economy produced by other refinements, such as heating feed water by bled steam and other smaller details. Coupled with this improvement in efficiency was an endeavour to simplify the construction of the engine, and cut down its weight. Here, too, marked results have been obtained, more particularly with the Diesel engine, and for equal or even higher powers savings in weight have been reached of 20 per cent and more. This result is principally due to the application of welded instead of cast iron construction. The results for steam engines are less striking, although here too, savings in weight have been obtained, i.e., in high-pressure boiler construction, rotors, etc. It may be expected that also for this type of machinery further reductions in weight are possible. The following table outlines some important characteristics of four modern propulsion engines:

Type of installation	Reciprocating steam engine	Steam turbine	Slow-running 2-stroke single-acting Diesel	Fast-running 4-stroke highly supercharged Diesel
Steam pressure	440lb. per sq. in.	600	—	—
Steam temperature ...	740 deg. F.	850 deg. F.	—	—
Feedwater temperature ...	225 deg. F.	240 deg. F.	—	—
Fuel consumption, lb./s.h.p./hr.	0·67	0·53	0·372	0·348
Total efficiency per cent of installation...	20·5	26·0	37·0	39·4

The owner will not be guided in his choice of a certain type of propulsion installation by its technical merits only, but also by factors bearing on the exigencies of the trade. The turbine installation commends itself for instance, as being low in height and by the possibility to use heat produced in the boilers for purposes other than propulsion alone, which, more especially on passenger liners and big tankers, is a factor of importance. Further, there are the low cost of repair and overhaul, and the problem of personnel. This latter consideration has become of foremost importance since the training of responsible engineroom personnel has failed to keep pace with the rapid expansion of the world fleet. At present, the Diesel motor demands more personal supervision and attention and calls for a longer training of the staff than the steam-driven engine.—*Paper by S. van West read before the Baltic and International Maritime Conference at the Hague on 11th June 1953; The Shipping World, Vol. 128, 24th June 1953; pp. 571-572.*

High Powered Diesel-electric Icebreaker

In order to keep the entrances to Finnish harbours free from ice during the winter, the Finnish Government, in 1950, ordered the highest-powered Diesel-electric icebreaker that had hitherto been constructed. This ship, the *Voima*, has been completed by the Sandvikens Skeppsdocka. She is about 274 feet long with a breadth of 63 feet and a draught of 20·3 feet, the displacement being 4,415 tons. In two enginerooms six Polar engines of 2,000 b.h.p. are installed, each driving a d.c. generator of 1,370kW. There are four propelling motors, two forward and two aft, each with an output of 3,500 b.h.p.—*The Motor Ship, Vol. 34, July 1953, p. 172.*

Electrolytic Tank Development

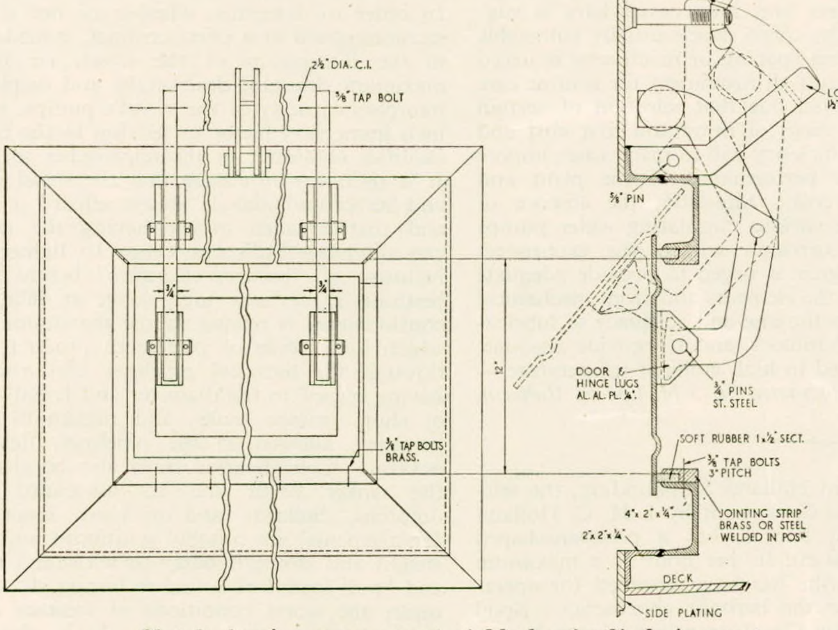
A recent report describes the design and development of an electrolytic tank for the study of hydrodynamic flow about torpedoes and other bodies. In this tank it is possible to survey velocity distributions of the potential flow about two-dimensional bodies of revolution with an accuracy comparable to that obtained by direct measurement in the towing basin or water tunnel. The method may eventually be extended to study more general types of flow on three-dimensional bodies, e.g., pressure distributions on a ship model. Electrolytic tanks, of course, have been built at many laboratories in various countries, but there still remains much scope in the refining of testing techniques, as, for instance, for obtaining pressure distributions. Also, considerable care is necessary in designing and constructing an electrolytic tank. The lining of the tank described in the report was cast of a wax compound and the surfaces were made smooth and plane by hand scraping. This type of lining was adopted after considerable trouble was encountered with a glass lining. The dielectric models investigated were machined out of the same hard wax compound used for ship models, and sprayed with a hydrophobic waterproofing plastic compound to eliminate the meniscus of the water surface. Polarization of the electrolyte adversely influences accuracy of readings, but polarization may be reduced by increasing the frequency of the power supply and by coating the probe and electrodes with Aquadag.—*The Engineers' Digest, Vol. 14, June 1953; pp. 199-200.*

Challenge in Ship Design

In his paper, the author acknowledges the contributions to the science and practice of shipbuilding and marine engineering by those who have gone before but asserts that great fields for endeavour remain, new methods bring problems and fresh demands for their challenge. After making reference to Canadian shipbuilding achievements in past and present days, he points out that present-day ships are either wholly or largely electrically welded. Attention is directed upon the merits of longitudinal framing of electrically welded bottom structures. The case of the welded *Ocean Vulcan* and the riveted *Clan Alpine*, upon which investigations were carried out under the direction of the British Admiralty Welding Committee, is cited and it is pointed out that corrugation on the bottoms, wholly or partially welded, of transversely framed ships causes considerable concern. Different methods of transverse and longitudinal framing of the bottom structure are discussed, supported by tables giving steel weights and electric welding quantities, with a view to nullifying believed reasons for reluctance of builders to incorporate longitudinal framing in seagoing dry cargo vessels. Brief reference is made to the strength of framing required by the Load Line Rules and Classification Societies, and of opportunities sometimes available to designers. The question of deck supports is considered, alternative pillar and girder arrangements are indicated and their features discussed. Various types of pillars are shown and attention drawn to savings in weight made possible by the better disposition and balance of material permitted by designs for electric welding. In conclusion, the author mentions several necessary guides toward good structural design.—*Paper by H. L. Walker, abstracted in S.N.A.M.E. Bulletin, Vol. 8, June 1953; p. 34.*

Ship's Freeing Port

The accompanying sketch illustrates a form of freeing port recently designed by Mechans, Ltd., of Glasgow, for fitting in covered spaces above the freeboard deck and in well deck bulwarks. The port is fitted inside the line of the ship's plating so as to avoid any interference from the wash of the sea running along the side of the ship, and to prevent damage from contact with quay walls or other vessels. It is so designed



Sketch showing arrangement of Mechans' self closing port that as soon as the internal head of water is 6 inches above the sill, it opens and allows the water to drain, after which it closes automatically and silently against the watertight rubber seating, preventing the entry of sea water. A locking bar, which may be operated locally at the freeing port or from the deck above by link bar and operating screw being coupled to the locking bar, can be fitted to prevent pilfering of cargo. All parts are readily accessible from inboard to facilitate inspection and maintenance. It is claimed that the port will remain in the closed position in a ship which may be rolling heavily, and that its ability to do so ensures that there is no diminution of the ship's righting arm. The sill of the port is sufficiently high above the deck or waterway to prevent its being fouled by refuse.—*The Shipping World, Vol. 129, 1st July 1953; p. 21.*

Ship Maintenance and Future Design

The decisions made during the design stage will ultimately be of utmost importance in the maintenance expense of the vessel. Accordingly, and within the limitations imposed by owner's requirements, laws, class rules, and normal economy, the designer is urged to pay the utmost attention to those details which may result in excessive maintenance. In attempting to combat shell corrosion, the use of a minimum number of scupper connexions is strongly indicated, and attention directed to consideration of sewage sumps eliminating all sanitary discharges above load line. Other suggestions are advanced in the same vein, attempting to allow maximum shell protection from corrosion and maximum performance from standard preservative coatings. Examples of recurring defects observable in the various mass produced vessels are described, pointing to faulty design or material choices. Consideration is urged for the use of alloy materials in a variety of applications. In general, suggested applications for alloys are confined to situations wherein (a) the cost of materials in the completed article is minor compared with the cost of fabrication and (b) the weight of material involved in the finished article is relatively negligible. Obvious examples are those deck pipe line systems subjected to corrosion from within and without, and ventilation louvres and similar light sheet metal assemblies exposed to weather. In lieu of the use of alloys, careful surface treatment is urged for metal joiner work at deck level and the interiors of hollow metal joiner items such as doors and airport boxings. The designer is urged to restrict the use of comfort insulation and sheathing to those areas specifically requiring such treatment, both as a construction economy and to relieve the necessity of expensive removals and replacements in the event of future repairs. Increases in shell plate weights over rule requirements and use of additional stringers and intercoastal bars is suggested in the bottom areas of dry cargo vessels usually vulnerable to pounding damage. Uniform spacing of machinery is urged to provide adequate space around all machinery for routine care and maintenance. It is pointed out that selection of certain vital auxiliaries solely on the basis of minimum first cost and maximum initial mechanical efficiency will in many cases impose overall restrictions upon the performance of the plant and result in high maintenance cost. Similarly, the absence of adequate speed control in the various circulating water pumps may result in accelerated corrosion within the exchangers served. Electrically, the designer is urged to provide adequate protection for wireways from the elements and from mechanical damage, to look particularly to the ease and adequacy of lubrication arrangements for electric motors, and to provide adequate ventilation to all motors located in high ambient temperature.—*Paper by W. K. B. Potts, abstracted in S.N.A.M.E. Bulletin, Vol. 8, June 1953; p. 35.*

Dredging Equipment for Brazil

According to an article in Holland Shipbuilding, the self-propelled bucket dredger *Vera Cruz,* built by I. H. C. Holland for work in the harbour at Santos, has a pontoon-shaped hull and a capacity of 17,500 cu. ft. per hour at a maximum dredging depth of 50 feet. She has been designed for operation in the open sea to clear the harbour approaches. Spoil is transferred to a fleet of four Diesel-propelled hopper barges, which dispose of it into the South Atlantic some ten miles out. These hopper barges made the voyage across the Atlantic under their own power. They have a carrying capacity of 800 tons of spoil at a hopper content of 17,500 cu. ft. An outstanding feature is the absence of any obstruction above the hopper compartment, where cross members have been dispensed with by incorporating all necessary transverse strength in the bottom construction. The principal dimensions of the hoppers are:—

Length	164 feet
Breadth	29·6 feet
Depth	14·7 feet
Draught, loaded ...	12·9 feet (at a speed of 10 knots)

The propelling unit is a 600-h.p. M.A.N. Diesel engine. The overall measurements of the hopper compartment are 65 × 22 × 14 feet. In the bottom, twelve doors, arranged in two rows of six, which are operated by hydraulic rams on the foredeck, ensure efficient discharge of the spoil.—*Journal, The British Shipbuilding Research Association, Vol. 8, June 1953; Abstract No. 7598.*

Terminal Facilities for Large Ocean Tanker

The high initial and subsequent operational costs of large ocean tankers necessitates that these vessels should load and discharge their bulk oil cargoes at the maximum loading and discharging rates commensurate with safety, with resultant reduction in port turn-round and increased overall carrying capacity. The modern 26,800 d.w.t type tanker, it must be pointed out, is a vessel capable of discharging cargo at average pumping rates of 16,000 to 18,000—42 gallon barrels per hour, i.e. 2,100 to 2,400 long tons per hour (crude oil basis) using centrifugal type cargo pumps. Experienced tanker personnel and terminal staff supervise all cargo loading and discharging operations in port, having regard to the requirements of the local harbour authorities' by-laws and Governmental regulations relating to the loading and discharging of "high flash" and "low flash" petroleum products, in addition to which tanker owners' and oil companies' safety regulations have to be observed at all times. "In port" turn-round time is an important factor in the economical operation of tankers and this means that marine terminal facilities used by vessels discharging and loading bulk oil cargo at the oil refineries and bulk storage terminals at the various ports are designed to eliminate unnecessary delays to tankers. This may entail improvements in the navigational approaches to the port and at the various tanker terminal berths located therein, having regard to the dimensional characteristics of tankers using such facilities. In order to determine whether or not a tanker can be safely accommodated at a port/terminal, consideration must be given to the dimensions of the vessel, i.e. overall length, beam, maximum draught, deadweight and displacement tonnage, the pumping capacity of the vessel's pumps, and also the tons-per-inch immersion factor in relation to the navigational and berth facilities obtaining in the approaches to and within the port. It is necessary to ensure that the vessel can safely proceed to and be accommodated "always afloat" at the prospective berth, and that a laden tanker having the designated dimensions can discharge sufficient cargo to lighten to a safe draught, inclusive of "bottom clearance" before first low water after berthing at or near high water at tidal berths. This latter consideration is related to the maximum receiving capacity at which each grade of petroleum product can be safely passed through the terminal pipelines and associated flexible hoses having regard to the diameter and length of the lines, elevation of shore storage tanks, and maximum permissible pumping pressures allowed in the pipelines, flexible hoses and connexions. Consideration must also be given to whether or not the tanker berth and its associated structures (mooring dolphins, bollards and/or buoy moorings, also fendering arrangements) are suitably positioned and of sufficient stability, weight and strength safely to withstand the breasting mooring and "pull loads" of a tanker having the designated dimensions under the worst conditions of weather and tide likely to be experienced in the locality. It is also important that the positioning of the tanker's discharge manifold connexions when at the berth, are in alignment with the dock manifold connexions, when the parallel middle body length of the tanker is evenly distributed over the breasting face of the berth which it is lying alongside.—*Cargo Handling, Vol. 1, June 1953; pp. 34-35.*

Research in Dredging Problems

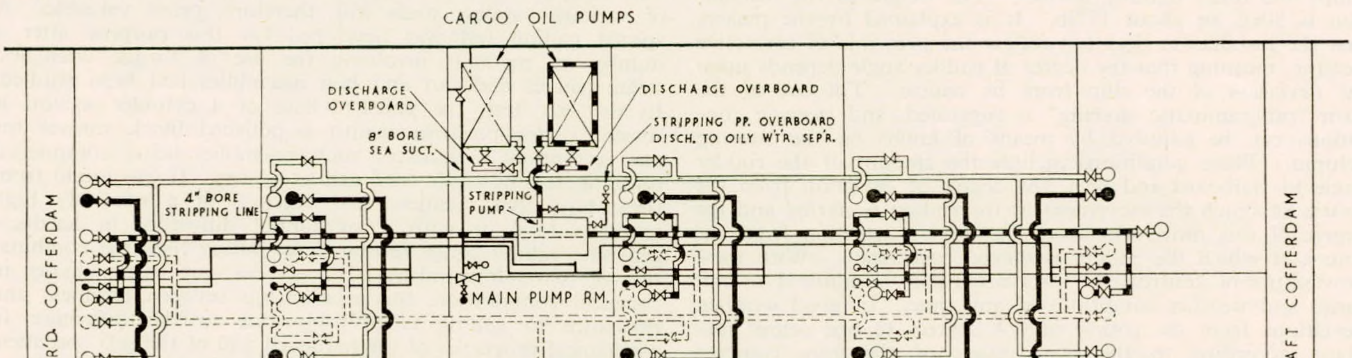
A recent extension of the Mineral Technological Institute at Delft, comprises a tank for studying the behaviour of dragheads of trailing suction hopper dredgers. The spoil output of a trailing suction dredger is determined, amongst other things, by the volume of the mixture of spoil and water which is pumped up during a certain unit of time, and the proportion of solids in the mixture of spoil and water; i.e. concentration. For a given shape of draghead the output of mixture and its concentration depend upon: 1, Vacuum. 2, The speed at which the draghead is drawn along the bottom. 3, The position of the suction opening in regard to the bottom surface. 4, The force by which the draghead is pressed upon the bottom. 5, The nature of the soil. When a dredger is at work the influence of factors 1-5 cannot be verified, since it is not possible to control the conditions of dredging with sufficient accuracy, and this afforded an adequate reason for proceeding with the construction of a testing installation. The installation consists of a concrete tank 90 feet (27m.) long, fitted with glass panes on one side. This tank is constructed at a fair height above floor, and below it, on ground level is a second tank which serves as a collector. Over the main tank a travelling carriage is fitted, from which a suction tube is suspended. The suction tube is divided lengthwise on its centre line, and is also fitted with a glass wall. The speed of travel of the carriage and consequently the speed at which the draghead is drawn along the bottom of the tank, can be regulated from very slow to very fast by means of a hydraulic variator. On the carriage a platform is fitted with measuring instruments, a film apparatus with lamps, and room for the attendants and their controls. By means of this installation it is possible to regulate and to measure the factors 1-5, and furthermore, to determine the force required to draw the draghead along the bottom. The results of research carried out up to the present are very interesting in that they have produced figures hitherto unknown, which in combination with visual observations greatly widen the views on the working of dragheads.—*Holland Shipbuilding, Vol. 2, May 1953; pp. 14-15.*

New Lubricating Oil Tanker

The *Vacuum Pioneer,* which is a single-screw steam tanker for the carriage of lubricating oil, petrol or kerosene, has been completed by the Grangemouth Dockyard Co., Ltd., for the Vacuum Oil Co., Ltd. This tanker is intended for service between the owner's new Coryton refinery on the Thames and their oil compounding and blending and grease manufacturing works at Birkenhead. Due to the multiplicity of grades of lubricating oil which are to be carried, special pumping and pipeline arrangements have been incorporated in order to give her the maximum flexibility in cargo handling. The tanker is so designed that she will be suitable for foreign going as well as coastal service. The principal particulars of the *Vacuum Pioneer* are as follows:—

Length b.p.	245 feet
Breadth moulded	40 feet
Depth	16ft. 6in.
Draught	15ft. 3in.
Tonnage d.w.	1,640 tons
Speed	$11\frac{1}{2}$ knots
Dry-cargo hold	8,200 cu. ft. bale

There are ten separate tanks, each one varying in capacity between 130 and 170 tons, for the carriage of various grades of lubricating oil. Heating coils of malleable steel piping are installed and have a heating capacity of one square foot for each 42 feet of tank capacity. The pumping arrangements are unusual due to the multiplicity of grades of oil that are to be carried. There are four 8-in. pipelines, two of the lines running through the starboard tanks. Each pipe has an 8-in. suction with bellmouth strum and a valve with extended spindle in each cargo tank, making a total of four suction pipes in each of the ten tanks. Each pipeline is connected to one of the four steam-driven cargo pumps and there are four separate overboard discharge lines on deck, port and starboard, one from each pump. A 4-in. stripping line is fitted, with separate connexions to each oil cargo tank and to a recovered oil tank in the engine-room. The pump discharges overboard, port and starboard, and to the oily water separator. There is an 8-in. loading and discharging connexion on deck from each pump with connexions port and starboard. A direct-loading connexion is arranged to each tank line, bypassing the cargo pumps, and fitted with the necessary valves. The valve operating hand wheels are painted in distinctive colours in order to minimize errors. A four-tons oil recovery tank and a 20-tons per hour Coastguard oily water separator are installed to prevent oil being discharged overboard when pumping out ballast, thus polluting the water around the coast. Four horizontal duplex main cargo pumps made by Carruther, each of 150 tons capacity, and a 60-tons capacity stripping pump, are installed in the pump room. A North Eastern Marine direct acting, triple expansion, reciprocating type of reheat engine is installed and gives the vessel a speed of about $11\frac{1}{2}$ knots at 115 r.p.m.—*The Shipping World, Vol. 129, 15th July 1953; pp. 56-58.*



Layout of cargo pipelines in the Vacuum Pioneer

New German Gyro Pilot

With the object of reducing the amount of yaw which may take place when a gyro pilot installation is in use, the Allgemeine Elektricitäts Gesellschaft have introduced a device designed to ensure the greatest possible steadiness of a ship's course. It is stated that the adjustment of large rudder angles, in accordance with the amount of deviation from the ship's course, is no longer necessary and the new gyro pilot is suitable both in combination with gyroscopic and magnetic compass installations. In operation, fixed rudder angles are determined for the port and starboard sides of the helm. Once off the adjusted course, as soon as the ship begins to turn back again, the helm is put amidships and yawing with the rudder is avoided. A handle is provided which allows the course to be altered without the apparatus being switched over to hand steering. On the steering pedestal are red and green control lamps, indicating port and starboard helm. In the centre of the pedestal cover, which is arranged at a suitable angle for easy examination by the helmsman, is a repeater

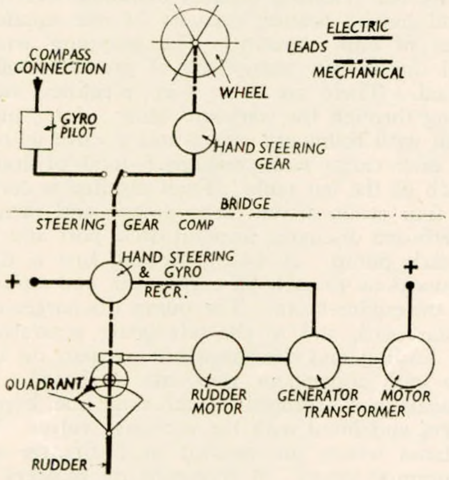


Diagram of electrical and mechanical connexions

compass which can be used as a steering compass when hand steering is employed. A buzzer is installed for giving warning if unintended deviations from the ship's course occur. The cover, with all the instruments, shown in the accompanying illustration, can be raised and the entire apparatus is thereby accessible by a single movement. The column contains a switch for selecting automatic or hand steering, together with switches for dimming the indicator lamps and other lighting devices. The weight of the installation is 80kg. or about 177lb. It is explained by the makers that the installation does not follow the principle of corrective steering, meaning that the degree of rudder angle depends upon the deviation of the ship from its course. The use of the term "programmatic steering" is suggested, and steering conditions can be adjusted by means of knobs on the steering column. These conditions include the amount of the rudder angle to starboard and port, the degree of deviation from the course at which the movement of the rudder is started and the degree of this movement from the largest deviation from the course at which the rudder is moved back again. With these possibilities of control, the installation can be adjusted to suit cargo and weather conditions at any time. In good weather, deviations from the course of 0·4 degrees to one degree take place, according to the sensitiveness of the gyro compass installation. In bad weather, the amount of deviation naturally depends on the yawing of the ship.—*The Motor Ship, Vol. 34, July 1953; p. 147.*

Flow About Slender Bodies

A linearized theory is developed for steady, two-dimensional cavity flows about slender symmetric bodies. The theory is applied to the cases of zero and non-zero (positive) cavitation numbers. It is shown that, for the case of finite cavities, the linearized theory avoids the necessity for choosing an artificial cavitation model as must be done in any exact theory attempts. The problem of calculating cavity shapes and drags for arbitrary slender bodies is reduced to one of quadratures. As an example, calculations are made for the family of wedge profiles and results are shown to be in good agreement with "exact" theory results for sufficiently slender bodies. In particular, the example demonstrates that the linearized theory is a valid first order theory.—*M. P. Tulin, Navy Department, The David Taylor Model Basin, Washington, D.C., Report No. 834, May 1953.*

Tank Cleaning Installation at Cardiff

The Mountstuart Dry Dock Co., Ltd., have built a large tank cleaning plant in the Roath Dock, Cardiff, where ships up to 31,000 tons d.w. can be dealt with. This new plant has been designed to gas-free and clean the tanks of oil tankers after discharge and obviates the necessity for their proceeding to sea for 6-8 days in order to do so. Sea pollution, which is causing so much concern at the present time, is also avoided. The installation consists of a boiler house, an oil separator designed by the company, filter and oil storage tanks. The cargo or fuel tanks are washed out by Mountstuart employees, using the Butterworth system, and the oily ballast is pumped ashore through the separator and filter, whence the oil-free water is discharged into the dock. The separator and the associated oil storage tanks are protected from the possibility of fire by a new system of fire prevention, by W. C. Holmes Ltd., of Huddersfield. This involves the generation of an inert gas which fills the space between the surface of the oil and the top of the tank, and is maintained there under a slight pressure. In addition, and to comply with existing Home Office requirements, a complete Pyrene foam generating and distributing system is fitted, connected to all tanks and the boiler house. All parts, including pipelines, are protected against the possibility of damage caused by the existence of static electricity. The plant has dealt very successfully with two tankers belonging to the Anglo-Saxon Petroleum Co., Ltd., the first ship being completed in two days and the second one in three days, before proceeding to the repair berths. The Mountstuart Combine is the largest ship-repairing company in the Bristol Channel, and, in fact, in the whole country, and owns eleven dry docks at Cardiff, Newport and Barry, together with a modern repair establishment at Avonmouth.—*The Marine Engineer and Naval Architect, Vol. 76, June 1953; p. 239.*

Preventing Stainless Steel Galling

Where ordinary lubrication methods are not applicable, the prevention of galling and seizure of stainless steel surfaces in mutual rubbing contact presents a considerable problem. The results of an investigation of the galling characteristics of various stainless steels will, therefore, prove valuable. A special galling test was developed for this purpose after a number of methods involving the use of torque wrenches, strain gauges and nut and bolt assemblies had been studied. In the new test, the polished base of a cylinder section is rotated under pressure against a polished block surface for one revolution, a series of such assemblies being compressed under increasing loads until galling occurs. It was found from these tests that stainless steel sections at a relatively high hardness level or with a substantial difference in hardness generally exhibit better resistance to galling than the combination of two soft members. Since there may be no change in chemical composition, this relationship between hardness and resistance to galling is probably due to the difference in mechanical properties of the hardened and of the soft specimen. The explanation has been put forward that the hardened sections deform elastically near the contact point under loading, while the softer, weaker pieces yield plastically for a significant distance beneath the contact points. During movement, hardened surfaces may recover elastically with decrease in pressure, and this motion tends to sever any metallic welding, whereas, in the case of the soft specimens, the surfaces probably continue to adhere closely during the movement of the specimens with relatively little elastic recovery and separation.—*The Engineers' Digest, Vol. 14, June 1953; p. 199.*

ENGINEERING ABSTRACTS

Section 3. Shipbuilding and Marine Engineering

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Neither The Institute of Marine Engineers nor The Institution of Naval Architects is responsible for the statements made or the opinions expressed in the following abstracts.

Pressure Welding

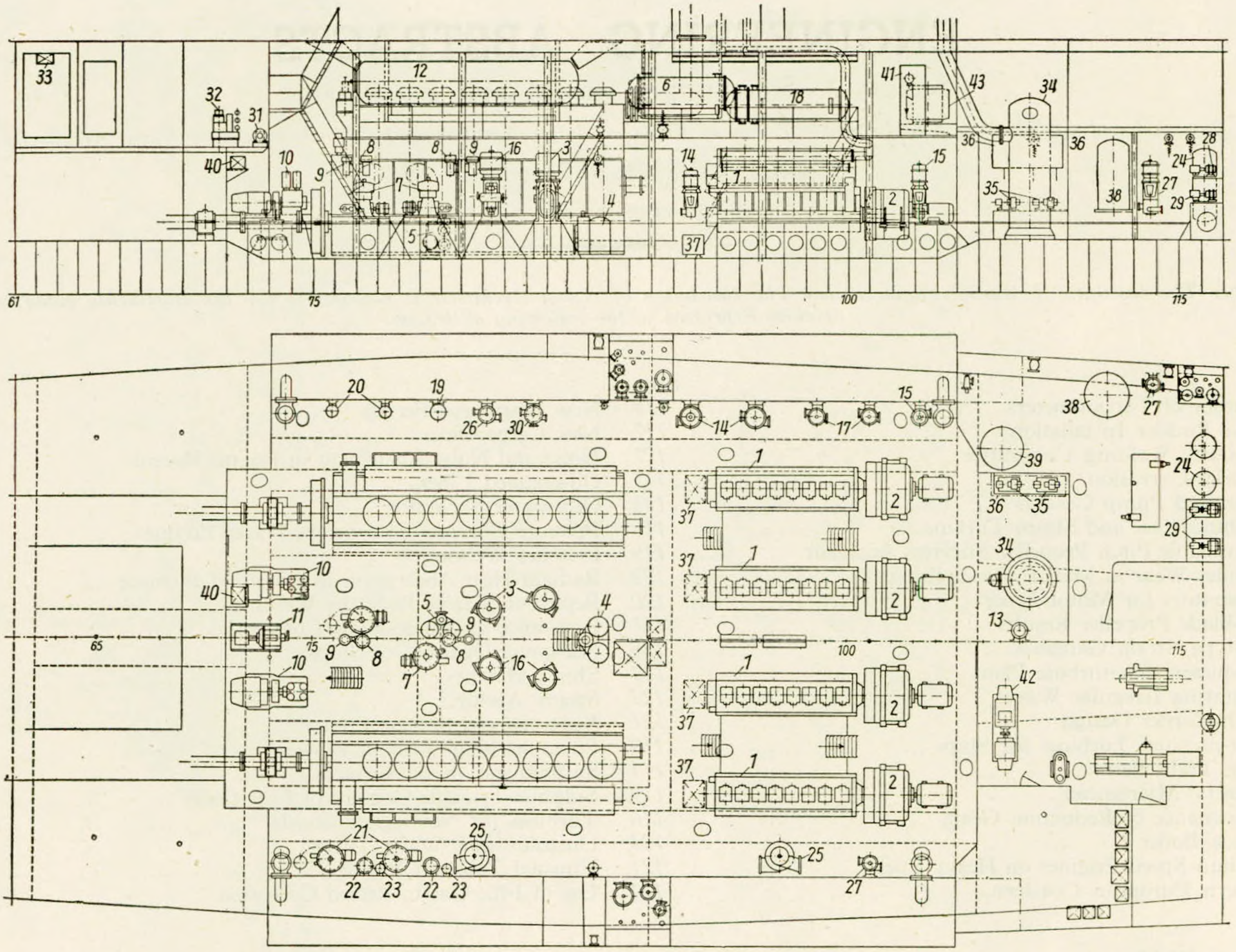
Pressure welding, or solid-phase welding as it is also termed, has attracted considerable attention in recent years. It is true that in the early stages of development considerable trouble was experienced in obtaining consistent results, but the fact that the process is now in full commercial use, although still on a limited scale, bears witness to the utility of the process as such. It is not always realized that this process is basically different from ordinary welding methods. In fusion-welding, metallic contact is achieved by the use of molten metal, with or without flux. Once intimate contact has been produced by the merging of molten metal or by the wetting of solid metal by molten metal, solidification produces a continuous crystalline structure between the faying surfaces and a sound weld is obtained. The molten metal serves two vital functions: (1) it eliminates the need for smooth faying surfaces to secure intimate metallic contact, because molten metal accommodates itself to the contour of the metal being joined; and (2) the molten metal, with the help of fluxes or parent metal melting, removes non-metallic matter from the surfaces being welded, thus assuring intimate metallic contact. In pressure welding, no molten metal is present to perform these two essential functions; some other mechanism must function to accomplish this vital need. Careful study and analysis have shown that this mechanism is recrystallization and that the so-called pressure-welding process should be more correctly known as recrystallization welding. The fundamental mechanism causing recrystallization is not as yet established, although there are several postulates concerning it. No explicit relationship relating quantitatively the dependence of recrystallization temperature to the deformation has yet been found. Metallurgists have been content to observe that the greater the degree of deformation, the lower will be the recrystallization temperature, and have contended that the many variables involved defy statistical analysis of the limited data available.—*The Engineers' Digest, Vol. 14, July 1953; p. 239.*

Large Train Ferry

The first post-war train ferry to be put into service by the German State Railways is the twin-screw m.s. *Deutschland,* recently completed at Kiel by Howaldtswerke A.G. This vessel is designed for the Grossenbrode-Gjedser service, which will now be maintained by the Danish and German railway concerns. Built to the requirements of Germanische Lloyd and specially strengthened for navigation in ice, the *Deutschland* has been constructed within six months, as the keel was laid on 1st November 1952, and the ship completed on 22nd April. She will carry either ten passenger railway coaches or twenty-four goods waggons, or a corresponding cargo of motor cars and lorries. Accommodation is provided for 1,000 passengers. The principal characteristics are:—

Length overall	375ft. 10in.
Length b.p.	356ft. 4in.
Breadth moulded	56ft. 6in.
Depth to train deck	23ft. 2in.
Depth to boat deck	47ft. 9in.
Draught	14ft. 9in.
Gross register	3,863 tons
Deadweight capacity	1,200 tons
Displacement	4,900 tons
Brake horsepower	5,500
Service speed	16 knots
Maximum speed	17·5 knots

A bow rudder is fitted and all rudders are streamlined and strengthened for navigation in ice, the general extent of the strengthening of these, the bow, stern and other parts of the



Engine room plans of the ferry ship Deutschland

1.—Auxiliary engines. 2.—Generators. 3.—Lubricating oil and piston cooling pump. 4 and 5.—Lubricating oil filters. 6.—Lubricating oil cooler. 7.—Lubricating oil separator. 8.—Lubricating oil heater. 9.—Water heater. 10.—Starting air compressor. 11.—Auxiliary compressor. 12.—Starting air receivers. 13.—Starting air bottle for auxiliaries. 14 and 15.—Sea water cooling pumps. 16 and 17.—Fresh water cooling pumps. 18.—Fresh water back cooler. 19.—Fuel oil transfer pump. 20.—Fuel oil daily service pump. 21.—Fuel oil separator. 22.—Fuel oil heater. 23.—Water heater. 24.—Warm water geyser. 25.—Ballast pump. 26.—Bilge and deck washing pump. 27.—Fire-fighting pump. 28.—Sea water pressure tank pump. 29.—Fresh water pressure tank pump. 30.—Deck washing and fire-fighting pump. 31.—Sea water cooling pump for refrigerating machinery. 32.—Refrigerating machinery. 33.—Dry air cooler. 34.—Auxiliary boiler. 35.—Boiler feed pump. 36.—Preheater. 37.—Lubricating oil drain tanks for auxiliaries. 38.—Sea water pressure tank. 39.—Fresh water pressure tank. 40.—Fuel oil container for auxiliary compressor. 41.—Switchboard. 42 and 43.—Transformers.

vessel well exceeding the requirements of the classification society. The hull, which is almost wholly welded, is divided by eleven watertight bulkheads extending to the main—or train-deck. Sliding watertight doors are fitted to four of the bulkheads and those for the main engine room have been fireproofed with asbestos linings. The propelling machinery consists of two Howaldtswerke-M.A.N. single-acting, two-stroke engines with eight cylinders with a bore of 520 mm. and a stroke of 900 mm. Each engine has an output of 2,750 b.h.p. at 155 r.p.m., and the corresponding fuel consumption is estimated at 165 grams per b.h.p.-hr. (0·36lb.). Oil cooling is used for the pistons, the cylinder jackets and covers being freshwater cooled, as are also the shaft bearings. The steering gear for each rudder is of the A.E.G. electro-hydraulic type, either of which can be controlled from the forward and after bridge. Arrangements are made in the double-bottom tanks for the carriage of 22 tons of lubricating oil—sufficient for eight days with three double trips daily—and there is provision for 140 tons of fuel. The service speed of 16 knots is attained at 68 per cent of the full power output under favourable weather conditions.—*The Motor Ship, Vol. 34, July 1953; pp. 152-155.*

Free-piston Gas-turbine Plant

In this paper the latent possibilities in heavy-duty free-piston machinery are discussed. The cycle promises Diesel efficiency from a simple low-temperature gas turbine having low first cost. Many features are described in the paper. Cooper-Bessemer has built a test plant and is now investigating its operating characteristics. No conclusions have been announced. Economic evaluations are being made requiring additional field work, which includes an examination of the general acceptance by engineering and operating people who will ultimately decide the actual utility. Detailed cost studies of complete plants, including careful estimates of the machinery, have been made for electric-power generation, pipe-line pumping, and marine installations. Some of these have been made available for use in this paper. The Cooper-Bessemer investigation of a heavy-duty free-piston turbo-power plant has been followed with interest by the U.S. Navy Bureau of Ships because of its possible future application to all types of auxiliary vessels such as cargo ships, troop transports, tankers, and others. Free-piston machinery may be applied to wide ranges of weights and outputs, some of which have only recently been undertaken. However, these findings indicate a very definite suitability to many heavy-duty applications. In this case the free-piston machine weighs about 30lb. per h.p. normally aspirated and will weigh less than 20lb. per h.p. when supercharged. It is designed structurally as heavy as a 100lb. per h.p. motorship Diesel. The paper has three objectives: 1. To describe the potential effectiveness of heavy-duty free-piston machinery. 2. To discuss an investigation of a heavy-duty free-piston turbo prime mover. 3. To make a realistic appraisal of free-piston turbine power in three heavy-duty service applications—an electric-power generating station, a pipeline pumping station, and a cargo vessel.—*Paper by J. J. McMullen and R. P. Ramsey, read at the 1953 A.S.M.E. Spring Meeting. Paper No. 53-S-13.*

Ship's Hatches

The invention consists of a hatch opening in which the covers, when closed, lie flush with and form a continuation of the deck, guard rails being provided for the hatch which are automatically erected when the covers are opened. Referring to Fig. 4, the hatch covers (1) are arranged in pairs each hinged at an outer edge (2) and opening in an athwartships direction. The covers may be formed in any convenient length and in sections to provide the desired opening. The covers (1) may be constructed of any suitable material and may comprise a steel cover structure (3) with wooden sheathing (4) to conform to the sheathing of the deck (5). Each cover is hinged to the deck (5) by hinge links (6) providing double hingeing axes (7 and 8) so that the cover (1) when open can lie flat on the deck. When the cover reaches a vertical position by hingeing about the first axis (7), it engages the hinge link with a squared edge (9) so that further movement takes place about the second hinge axis (8). The hinge links (6) are provided with a square edge (9) to ensure the reverse action occurring on closing the cover. When the length of the covers warrants their use, swinging beams (10) may be provided to give additional longitudinal strength to the hatch covers when in the closed position. These beams are carried by the covers on their inner surfaces, the beams are secured to the covers by cleats (14) and are cut away to allow them to swing into position as the covers are closed.—*Patentees: L. Walker and V. Walker. (British Patent No. 686,015.) World Shipbuilding, Vol. 3, May 1953; pp. 68-69.*

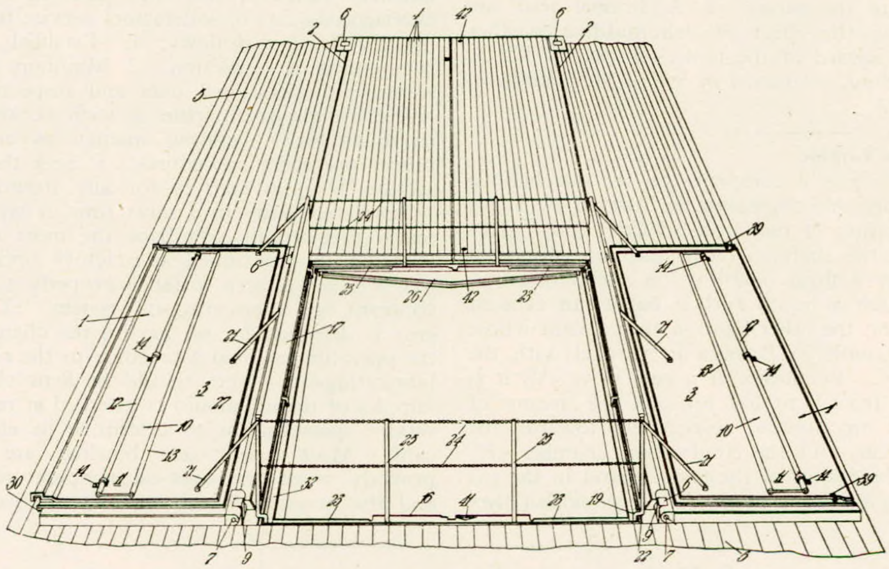


FIG. 4

Sulphinuzation of Surfaces

The "Sulphinuz" process of surface-finishing takes advantage of the property of sulphur of combining with ferrous metals, and forming a surface layer that reduces friction. The process consists of immersing the steel or cast-iron component to be treated in a bath of an appropriate composition at a temperature of about 1,040 deg. F. for a time ranging from one-half to three hours. The surface hardness is not increased to any notable degree, but the coefficient of friction is reduced and the resistance to wear becomes superior to that of the hardest steels. The results of tests on treated and untreated specimens are discussed. All ferrous metals so far tested have proved capable of being "sulphinuzed". Applications include the cylinder-head nuts for large marine oil engines, which then do not seize or oxidize even after long service at sea. The rate of wear of piston rings treated by this process is only a fraction of that of similar untreated rings. The process is also used on tools, when the life of the tool before regrinding becomes necessary is found to be considerably increased.—*G. de Lavalette and E. Partiot, J.Soc.Ing.Auto.France, 1952; p. 119. Journal, The British Shipbuilding Research Association, Vol. 8, June 1953; Abstract 7,634.*

Bacterial Corrosion

Reports published by the British Admiralty and by individuals in England and the United States, show that sulphate-reducing bacteria can cause corrosion of iron under some conditions. Because of the importance of maintaining in good condition the steel piling of offshore structures in the Gulf of Mexico, a research investigation of the possibility of bacterial corrosion of these installations was inaugurated. Laboratory tests were directed toward developing fairly pure cultures of sulphate-reducing bacteria and using them in both natural muds and synthetic culture media for the purpose of determining their effect on the corrosion rate of steel coupons under anærobic conditions. Maximum corrosion rates obtained were of the order of 0·001 inch per year. Several assemblies of twenty coupons each were installed at a marine platform approximately seven miles off Grand Isle, Louisiana; the coupons on each assembly were spaced to extend a few feet above and a few feet below the mud line. Somewhat similar assemblies were installed immediately offshore at Barbour's Cut, near La Porte, Texas. Results of the study show that the corrosion rate due to bacterial action is low, and that cathodic protection is a sufficient safeguard against bacterial corrosion.—*J. A. Caldwell and M. L. Lytle, Corrosion, Vol. 9, June 1953; pp. 192-196.*

Use of Flue Gas to Retard Corrosion

Tankers in gasoline service are subject to excessive corrosion, greatly shortening their service life. The paper discusses various methods utilized in the past to correct this situation and presents not only data on corrosion rates but discusses the economics of previous methods and this present method now being evaluated by the Navy. While dry weather air introduced into the compartments can reduce corrosion substantially, tests indicate it may increase the hazard of explosion. Dry flue gas blankets the cargo and prevents explosion. The primary problem involved in the work was the elimination of contaminant particles in the flue gas by various methods of filtering and cleaning to prevent contamination of cargo. The control system for the subject system is described and many charts and graphs are included giving information on efficiency of filters, tank thermal pressures plotted against atmospheric temperatures; vapour concentration versus temperature of gasoline and similar data. Conclusions are as follows: 1. Flue gas in above the minimum concentration of 38 per cent by volume will prevent explosion in gasoline tanks. 2. Flue gas is the most acceptable means of inerting the tanks when properly cleaned. 3. Introduction of clean flue gas into the tank will not injure the cargo. 4. Additional tests are required for establishing the effect of dehumidified weather air on the explosibility hazard of the tanks.—*Paper by W. P. Stewart and H. W. Keeling, abstracted in S.N.A.M.E. Bulletin, Vol. 8, June 1953; p. 32.*

Combined Gas and Steam Turbine

The plant shown in Fig. 1 comprises on the one hand a gas turbine unit with an air compressor (C) driven by a gas turbine, the latter consisting of two units (TgHP and TgBP) arranged in series with the shafts of both turbines connected to one another through a fluid coupling (A), a combustion chamber (F) inside which a liquid fuel is burnt, an exhaust gas air heater (R); and on the other hand, a steam plant whose boiler (Ch) is arranged with its furnace in parallel with the combustion chamber (F). By means of a gate valve (V) it is possible to vary at will from 0 to 100 per cent the streams of compressed and heated air directed respectively towards the furnace of the boiler (Ch) and the combustion chamber (F). The flue gases mix at the outlets of the two, expand in the gas turbine and flow through the air heater (R) from which they escape to the atmosphere. In the steam circuit the ahead and astern turbines (TvAV and TvAR) are not provided with any bleeder point for stage heating, but the evaporating tubes are preceded by an economizer (E) which completes the abstraction of heat from the flue gases after they have passed through the air preheater.—*Patentees: Société Rateau and R. Anxionnaz. (British Patent No. 684,959.) World Shipbuilding, Vol. 3, May 1953; p. 68.*

Maintenance of Reduction Gears

The care which should be exercised in order to experience the greatest amount of trouble-free service from gears in any rating is not particularly influenced by the size or rating of the gear. However, the amount of time and expense involved in repairing or replacing a gear as a result of improper operation or maintenance may be influenced greatly by the size of the unit. It behoves the operator, therefore, to be sure his equipment is operated and maintained in such a manner as to retain its desirable good operating qualities. One of the best ways to accomplish this aim is through preventive maintenance. Some of the rules followed by operators who have experienced years of satisfactory service from their marine gears may be listed as follows: 1. Establish and follow a regular programme of inspection. 2. Maintain accurate and complete records of operational data and inspection reports. 3. Take care of minor irregularities as soon as possible. 4. Refer to the manufacturer's operating manual as an aid to establishing proper operating procedures. 5. Seek the assistance of manufacturer's representatives for any irregularities which cannot be corrected easily in a short time or which appear to assume major proportions. Perhaps the most important single item affecting the continued satisfactory operation of a reduction gear which has been installed properly and placed in operation concerns the lubricating-oil system. The oil system of the gear is designed for oil having the characteristics set forth in the operating manual as a guide to the oil vendor. The ship's lubricating-oil system should be kept clean at all times, and samples of the oil should be checked at regular intervals against vendor specifications to determine its effectiveness as a lubricant. Most marine-gear bearings are designed to operate properly when the inlet-oil temperature is 120-130 deg. F. and the pressure 8-10lb. per sq. in. gauge. Oil-temperature

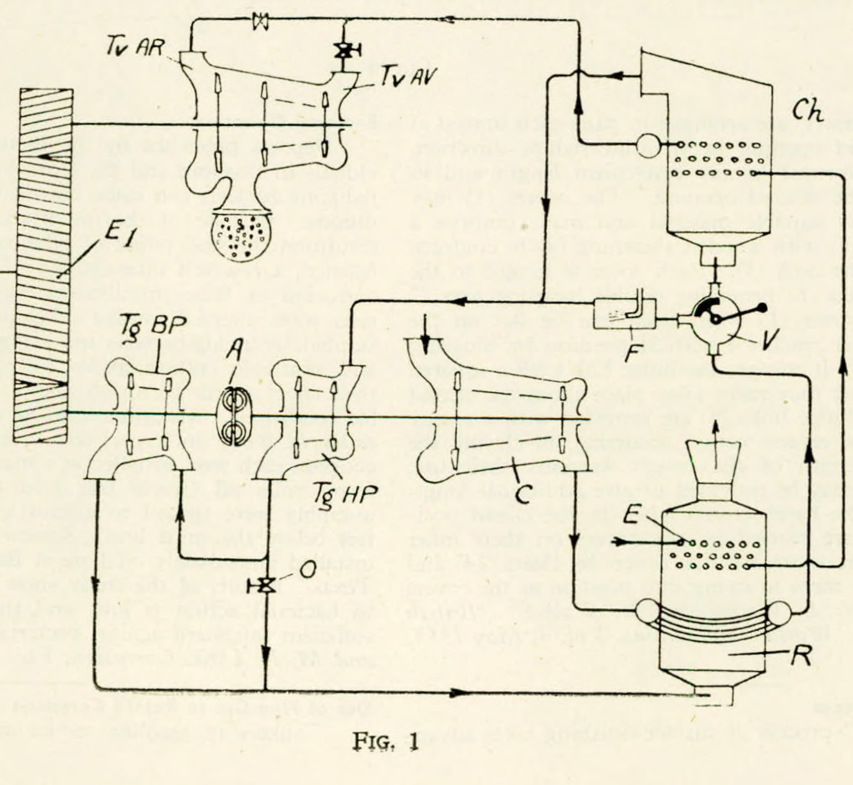


FIG. 1

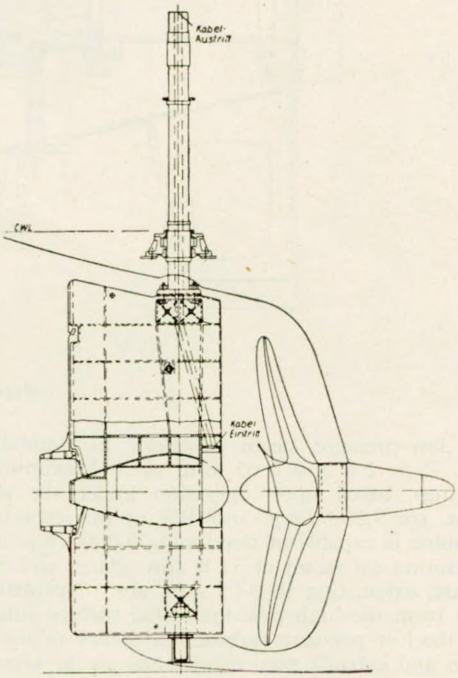
rise, as indicated by the temperature of the oil leaving a bearing, should not exceed 50 deg. F. under normal conditions. The maximum temperature of the oil leaving a bearing should not exceed 180 deg. F. Higher exit temperatures indicate insufficient bearing cooling. Propulsion-gear high-speed couplings connecting the prime mover(s) to the reduction gear pinion(s) are lubricated in one of two ways; by a separate orificed nozzle supplying oil to a lip on the coupling sleeve, from which oil flows about the coupling teeth, or by an orificed nozzle supplied with oil from the adjacent pinion bearing to a lip on the pinion flange from which oil flows about the coupling teeth. Recently designed couplings are equipped with dams to maintain a level of oil at the coupling teeth so that they operate in a bath of oil. Provision is made for trapping foreign matter deposited by centrifugal action in a groove located at one side of the coupling teeth. Material trapped in the groove should be removed when inspection indicates an accumulation is present. First-reduction propulsion-gear thrust bearings may be either the babbitted-plate type or the pivoted-shoe type. The pivoted-shoe type bearings are flood-lubricated. The second-reduction, or main, thrust bearing is the pivoted-shoe type for both ahead and astern thrust, and is flood-lubricated from the gear lubricating-oil system. Intermediate couplings are of the tooth type, lubricated through an orificed nozzle supplied with oil from the adjacent second-reduction pinion bearing. Auxiliary gears differ from propulsion gears, in so far as lubrication is concerned, in that the auxiliary gear normally is lubricated by a system which is complete within the auxiliary unit, whereas the propulsion-gear lubrication is supplied by the ship's lubricating-oil system. Therefore, the oil pump and pressure-regulating valve supply high-pressure oil for the governing system of the prime mover (if required) and low-pressure oil for lubricating bearings and gear mesh. Such an auxiliary system, therefore, has its own oil cooler and strainers. The principles of proper operation regarding cleanliness, oil temperatures, pressures and other factors are the same as for propulsion gears. Marine-gear rotors generally are supported in cylindrical babbitted bearings. Supplied with clean oil of proper characteristics, pressure and temperature, these bearings will give years of trouble-free service. Proper maintenance of gear-rotor bearings is of particular importance because maintaining proper gear-tooth contact is dependent almost entirely on the bearings in which the rotor journals operate. Irregularities in tooth contact result in rapid wear of the teeth, objectionable noise, and other indications of improper operation.—*A. D. Sutton, Marine Engineering, Vol. 58, August 1953; pp. 79-83.*

Five-blade Propeller Results

Substituting five-blade propellers for the original four-blade wheels on its C-3 passenger and cargo ships *Del Norte, Del Mar* and *Del Sud*, has produced worthwhile economies for the Delta Line of the Mississippi Shipping Company, Inc. Operation with the five-blade propellers manufactured by Baldwin-Lima-Hamilton Corporation, has shown a marked improvement in fuel consumption, cutting it 3·75 to 8·15 per cent, depending on boiler pressures and number of turbine nozzles used. Furthermore, there is a considerable reduction in machinery and tailshaft vibration and, therefore, less maintenance is needed on the mechanical equipment. Reduction in vibration is especially evident when the ships are slowed down and when going full speed astern. During performance tests with 24 nozzles in operation on the turbine, ship's speed increases half a knot over that with the old propeller. If the turbine is operated on 21 nozzles and the steam pressure raised to bring the speed to slightly above that under the old conditions the fuel economy is greatest.—*Marine Engineering, Vol. 58, August 1953; p. 64.*

Active Rudder Installation

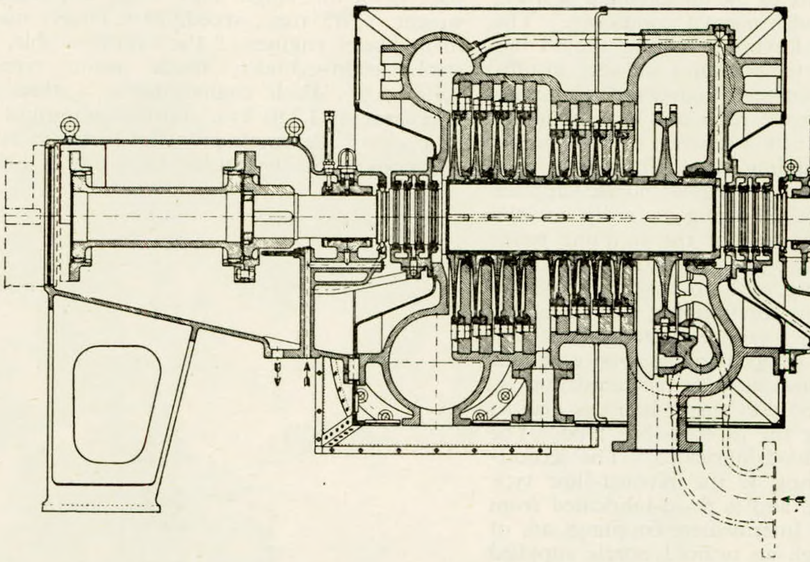
The new Diesel-electric dry cargo ship *Falkenstein* built by H. C. Stülcken Sohn, Hamburg, for Robert Bornhofen, Hamburg, is fitted with an active rudder of the Pleuger type. The salient particulars of the vessel are as follows: Length o.a., 134·4 m.; length b.p., 120·0 m.; breadth, 16·0 m.; deadweight, 7,685 tons; speed, 14·6 knots; machinery aft. Four main Diesel engines of the non-reversible, single-acting four-stroke eight-cylinder, trunk piston type, each developing 1,400 h.p. Each engine drives a three-phase synchronous alternator of 1,150 kva. continuous output at 3,000 volts and 50 cycles. The single propeller is driven by a three-phase synchronous motor developing 4,650 h.p. at 125 r.p.m. The design



of the active rudder is shown in the accompanying engraving. It incorporates a submerged electric motor of the squirrel cage type capable of a continuous output of 500 h.p. at 720 r.p.m. Overall diameter of the motor is 680 mm., and length without propeller is 2,400 mm. It is claimed that when both main propeller and rudder propeller are running, the efficiency of the main propeller is increased by the presence of the rudder propeller.—*Schiff und Hafen, Vol. 5, June 1953; pp. 245-268.*

Turbines for Norwegian Vessels

The reason why the Scandinavian shipowners are now interested in steam turbine drive seems to be the tendency to build large tankers of about 30-40,000 tons d.w. Up to 50 per cent of the Norwegian merchant fleet, reckoned in t.d.w., consists of tankers, and AB de Lavals Angturbin is therefore watching with great interest this renaissance of steam for the propulsion of vessels. De Laval has the following standard types: 7,300/8,100, 8,300/9,200, 12,500/13,700, 15,000/16,500 and 18,000/20,000 s.h.p. The consumption of fuel varies between 270 and 235 g./s.h.p. depending on the size of the turbine and the preheating arrangement for the feed water. At present three Norwegian ships are ordered with de Laval turbines: Anders Jahre & Co., Sandefjord, 8,900/9,800 s.h.p., Halfden Ditlev Simonsen and Co., Oslo, 12,000/13,700 s.h.p., and Hilmar Reksten, Bergen, 8,300/9,200 s.h.p. The 8,900/9,800 s.h.p. machinery is a modified 8,300/9,200 s.h.p. standard turbine. The 8,300/9,200 s.h.p. turbine consists of a 4,800 r.p.m., nine-stage, high-pressure turbine, a 4,150 r.p.m., six-stage, low-pressure turbine, a two-stage astern element which is built into the low pressure turbine, and a set of double helical, double speed reduction gears. The units are designed for normal operation requiring 8,300 s.h.p. at 104 r.p.m. of the propeller with the turbine taking steam at 1.36 atm. gauge, and 390 deg. C. temperature, exhausting against 0·05 atm. abs., and bleeding steam for air heater, feed water heating, evaporat-



High pressure turbine assembly

ing, and low-pressure steam generator in normal service at about 16, 7, 5, 2·6 and 0·05 atm. abs. Maximum load and speed ratings, based upon operation under the above steam conditions, are 9,200 s.h.p. and 108 r.p.m. respectively. The astern turbine is capable of developing 5,000 s.h.p. at 84 r.p.m. when operating on steam at 31·6 atm. gauge and 390 deg. C. temperature, exhausting to 0·05 atm. abs. Normally, steam is exhausted from the high pressure ahead turbine into the ahead stages of the low pressure turbine and hence to the condenser. The steam and exhaust pipe connexions are so arranged, however, that in an emergency either turbine can be operated alone on high presssure steam, exhausting into the condenser. The high pressure ahead turbine is supported at the aft end by the gear casing, to which it is secured by fitted bolts. The forward end is carried on a welded steel pedestal which is bolted to the hull structure and which is so designed as to permit axial expansion and contraction. The turbine shaft is connected by means of a flexible coupling to the starboard high speed pinion. The first stage is velocity compounded, having two rows of revolving buckets separated by a row of stationary buckets. They are dimensioned to stand stresses of speeds considerably higher than the normal one, and they are tested to a speed 30 per cent in excess of the latter. The combined low pressure ahead turbine and astern turbine is connected by means of a flexible coupling to the port high speed pinion. The ahead and astern elements are located in the aft and forward ends of the casing respectively, and are separated by a common exhaust chamber. As at the high pressure turbine, the aft end is supported on the gear casing and the forward on a flexible pedestal, bolted to the hull structure, and the casing is divided horizontally into two main sections. All six pressure stages of the low pressure ahead turbine and the second stage of the astern turbine are of the single bucket row type. The first astern stage is velocity compounded. The first stage nozzle elements in both ahead and astern turbines, as well as the steam pipe connexions, are placed in the lower sections.—*E. Hildebeck, European Shipbuilding, Vol. 2, No. 3, 1953; pp. 65-71.*

Noise and Noise Abatement in Engine Rooms

This report gives a survey of the literature on noise and noise abatement in ships' engine rooms. The acoustical principles involved are first briefly reviewed. The main points considered are the difference between air-borne and structure-borne sound, methods of reducing the strength of the sound sources, methods of reducing the propagation of both types of sound, and reducing sound levels by means of sound absorption. The application of these principles to the abatement of noise in engine rooms is then discussed. The logical sequence for noise reduction is silencing of engines, shielding the engines by means of sound-absorbing casings, the application of resilient mountings, and the reduction of noise levels by using sound-absorbent materials. None of these measures is likely to be sufficient by itself, and attention will probably have to be paid to all the points mentioned.—*Netherlands Res. Centre Shipbuild. Navig. Report No. 12M, April 1953. Journal, The British Shipbuilding Research Association, Vol. 8, July 1953; Abstract No. 77773.*

Unusual Tailshaft Repair

Bethlehem Pacific San Francisco Shipyard recently completed an unusual repair to a new Federal P-2 type tailshaft whose liner had been damaged because the shaft had been improperly prepared for shipment. Damage consisted of parallel grooves for the full length of both liners on the shaft. These were approximately $\frac{7}{16}$ inch wide and $\frac{1}{16}$ inch to $\frac{1}{8}$ inch deep. The owners were desirous of repairing these grooves only in the way of the packing gland area. The grooves were cleaned for 36 inches from the end of the liner, preheated uniformly with natural-gas ring heaters to 300 deg. F. and welded with the Heliarc process. After welding, temperature was equalized throughout the repaired area (approximately 300 deg. F.). The ring burners were cut off and the liner was wrapped heavily with asbestos. The shaft was then allowed to cool. The repair was successful and there was not distortion in the shaft. The welds were then machined off flush with the liner.—*Marine Engineering, Vol. 58, July 1953; p. 52.*

Unsteady-flow Water Tunnel

A project to investigate fluid friction and cavitation phenomena in unsteady motion is under way in the hydrodynamics laboratory at Massachusetts Institute of Technology. Its objective is to obtain for transient-flow conditions through conduits and around immersed bodies the same kinds of basic information that have been determined for steady-flow cases. A pilot-model water tunnel, developed to produce flow in which the basic motion is accelerating or decelerating, is of unconventional design. It is of the blowdown type and consists essentially of two tanks 20 inches in diameter and 6ft. 8in. long, mounted vertically, one above the other, and connected by a tube, or working section of 1 inch ID. Flow from the upper tank passes through guide vanes and an entrance nozzle, through a quick-opening valve. The velocity and acceleration of the flow through the tube are governed by the difference in air pressure above the water surfaces in the upper and lower tanks. This differential pressure, in turn, is controlled by a servo-system. High-frequency-response electronic cells were developed for measuring differential water presssures directly. These cells have been employed in experiments to determine loss of head along the test section and for measuring the instantaneous average velocity.—*Paper by J. W. Dailey, read at the 1953 A.S.M.E. Spring Meeting; Paper No. 53-S-31.*

Controllable Pitch Propeller Survives Accident

One of the two 9-ft. diameter KaMeWa controllable pitch propellers of the 3,000 h.p. towboat *Delta Cities* owned by Lake Tankers Corporation, New York, was recently involved in an unusual accident and survived with only a minimum of damage. The *Delta Cities* grounded while turning its tow around in St. Louis harbour on 19th January. The starboard rudder touched bottom, and the force of the grounding broke the rudder loose from its connexions inside the vessel to such an extent that the after portion of the rudder actually came into contact with the tips of the propeller blades. These blades, which are made of stainless steel, cut through almost an inch of steel plates to make a hole in the rudder more than a foot wide and nearly two feet high. Since it was evident that there had been no serious damage to the propeller and since the rudder connexions inside the vessel were repaired quickly, it was decided to continue operation of the vessel until a more convenient time for drydocking and repairs of the underwater damage. The *Delta Cities* continued operation with only a slight loss of efficiency for over a month before repairs were undertaken.—*Marine Engineering, Vol. 58, July 1953; p. 54.*

Accuracy of Torsionmeters

Doubt is frequently thrown upon the accuracy of torsionmeters, but the question is of more importance with turbine-driven ships than with those equipped with Diesel engines. There is no other ready means of ascertaining the shaft horsepower developed by a steam turbine when installed in a ship other than by torsionmeter, whereas with Diesel machinery indicator cards are always available. Nevertheless, torsionmeters are being installed to an increasing extent in motor ships, and with this in mind some experimental work was recently carried out in France in connexion with the calibration of the apparatus. The results were given in a paper read by P. Villepelet and M. Jourdain before the meeting of the Association Technique Maritime et Aeronautique, recently, entitled "Tarage Statique et Dynamique des Torsionmètres", the torsionmeters in question being of the Siemens-Ford type. The experiments were associated with the trials of the starboard engine of the motor passenger liner *Jean Laborde* at outputs ranging from 3,000 b.h.p. at 112 r.p.m. to 6,600 b.h.p. at 150 r.p.m. The results of the investigation led the authors to the conclusion that if the torsionmeter is fitted correctly and if the calibration is carried out with modern and efficient equipment, the measurements of the torsionmeter in normal service at sea can be relied upon to within a very small margin.—*The Motor Ship, Vol. 34, August 1953; p. 198.*

High-pressure Turbines for Ships

For a number of single-screw vessels of 10,250 tons d.w. the Siemens-Schuckert Werke have supplied turbines of 9,000 s.h.p., which gives these vessels a speed of 17½ knots, the propeller speed being 115 r.p.m. The turbines are supplied with superheated steam of 40 atm. g. and 425 deg. C. at the turbine throttle. The high-pressure and low-pressure cylinders develop approximately equal power. The plant is so designed that in an emergency either the h.p. or the l.p. turbine can be operated independently. The boiler feed water is preheated to 165 deg. C. by bled steam taken from two bleeder points at the high-pressure turbine. Both the h.p. and the l.p. unit are built as multi-stage reaction turbines; the astern turbine is of the Curtis type. The capacity of the astern turbine is 40 per cent of the ahead power, that is, 3,600 s.h.p. The turbines are coupled with the propeller shaft by a double reduction gear.—*E. Ehmsen, Hansa, Vol. 90, 21st July 1953; pp. 1184-1189.*

Magnetic Micrometer

The problem of measuring the horsepower being delivered by the propulsion shafting of full-scale vessels is exceedingly difficult. The huge amounts of power transmitted by propulsion shafts (up to 100,000 h.p. per shaft in naval vessels) preclude any possibilities of using reaction dynamometers in such limited spaces. Since the torsional deflexion of a propulsion shaft is proportional to the torque being delivered by the shaft (within Hooke's law), the most logical method of obtaining such shaft horsepower is to measure the torque by means of the torsional deflexion. The shaft horsepower may be readily determined when the shaft revolutions per minute are known. Almost every known type of strain-measuring device has been used for the purpose of measuring this torsional deflexion. Most of these devices have not proven satisfactory in operation due to one factor, namely, inability to measure the zero torque position effectively. As the propulsion shafts are carried in a large number of bearings, part of them being outboard, the propulsion shaft is always in an indeterminate state of strain due to the restraint of these bearings. The T.M.B. Magnetic Micrometer and Magnigage was developed to overcome the difficulty of measuring the zero torque position. The advantage of this measurement system is that all measurements can be reduced to a dimensional basis. A special torsionmeter shaft has been developed for use with the T.M.B. Magnetic Micrometer and procedures established for the measurement of the zero torque position before installation in the ship. With the advent of the T.M.B. Magnigage a linear, variable, differential transformer which is inherently magnetically shielded, a whole new field of application was opened up. It was now possible to apply this magnetic strain gauge without regard to the magnetic ambient, i.e. such as placing it in the centre of the shaft. As a result, the T.M.B. Dynamometer Shaft came into being. Here the magnigage is placed in the centre of a hollow steel propulsion shaft and arranged so as to measure the deflexion of an axially deformable portion of the shaft in order to obtain thrust. A portion of the shaft utilizes the magnigage to measure the torsional deflexion while a counter ring in the slipring assembly furnishes pulses to actuate a totalizing revolution counter. Hence, by utilizing the T.M.B. Magnetic Micrometer, the T.M.B. Dynamometer Shaft is completely self-contained for the measurement of torque, thrust, shaft revolutions and shaft horsepower. It has been built and used in sizes ranging from a 35-h.p. model dynamometer shaft to a 7,000-h.p. full-scale submarine dynamometer shaft. Design studies are also under way for utilizing the T.M.B. Dynamometer Shaft in a giant aircraft carrier having upwards of 100,000 h.p. and 3 million pounds of thrust per shaft. The T.M.B. Magnetic Micrometer principle has also been developed to cover the entire range of ship model testing in the form of transmission dynamometers and resistance dynamometers. The use of exactly the same instrumentation for both full-scale and model testing has greatly improved the predictive value of model data. The complete instrument consists of a manually operated indicator unit, a pick-up gauging element, and regulated power supply. A specially designed calibrator is also provided for mechanically calibrating the movement of the pick-up gauging element with the indicator prior to installation. The electrical circuit for the T.M.B. Magnetic Micrometer Mk. II consists essentially of two variable-reluctance magnetic gauges connected to form a Maxwell bridge. The output of the Maxwell bridge is fed into an electronic amplifier and null-detector network which indicates on a nullmeter when the two magnetic gauges are magnetically balanced or the direction and magnitude of unbalance.—*J. R. Pimlott, (U.S.) Navy Department, The David Taylor Basin, May 1953; Report 847.*

Oily-water Separator

In the White patented oily-water separator, the oil and water mixture to be separated is pumped through a mixture inlet valve and enters the primary separation chamber through vertical apertures in the distributor. Something like 90/95 per cent of the oil in the mixture is separated in the primary chamber, the oil rising and collecting in the upper section, the remaining mixture passing down through the primary chamber. Because of a convergent passage, the oil is compacted into larger droplets and progressive separation occurs. The mixture then passes into the secondary chamber via compacting nozzles before entering the tertiary chamber containing the filter bed. As the mixture changes direction to enter the filter bed, the small amount of compacted oil remaining is released and rises through passages into the top section of the primary chamber. The filter bed mentioned performs an important function by removing the small percentage of oil, or grit having a coating of oil, still remaining in suspension in a very finely divided condition. The filter consists of specially graded material through which the discharge water passes downwards and where all traces of oil contamination are removed. An interesting feature of the White separator is the provision of an electronic method of control. This is independent of the specific gravities of the liquids involved and therefore no adjustments are necessary on change of specific gravity. For this type of control the electrostatic properties of the liquids are utilized. This is achieved by inserting two metal probes in the oil chamber at the desired upper and lower water levels, and applying a voltage at radio frequency to each. Each probe, the liquid surrounding it, and the tank of the separator, acts as an electrical condenser. Because different liquids have different dielectric properties, the capacitance of this condenser changes with the change of liquid surrounding the probe. The change of capacitance causes a relay to operate, which in turn operates the contactor controlling the discharge valves, e.g. when the oil level reaches the lower probe, the relay operates and the oil discharge valve is opened by its electric motor, thus discharging oil from the separator until such time as the water level reaches the upper probe; when this occurs another relay operates and similarly opens the water discharge valve, thus discharging water from separator until the oil level again reaches the lower probe. The oil and water discharge valves are electrically and mechanically interlocked to prevent both oil and water being discharged from the separator at any one time. Because of the low voltages and powers applied to the probes, there is absolutely no danger of spark or fire hazard to the oil in the separator.—*The Marine Engineer and Naval Architect, Vol. 76, August 1953; pp. 323-325.*

Steam Motor

The M.A.N. works have developed a "steam motor" which develops 60 h.p. per cylinder at 1,000 r.p.m. Vertical sections of this engine are shown in the accompanying engraving. The engine is designed as a single-acting uniflow engine with poppet valves. Piston and crosshead are joined together so that no piston rod is required. Condensate is drained off between cylinder and crosshead guide. The engine is available as three, four, or six-cylinder unit for pressures up to 30 atm. g. and 400 deg. C. steam temperature.—*Hansa, Vol. 90, 21st July 1953; p. 1,222.*

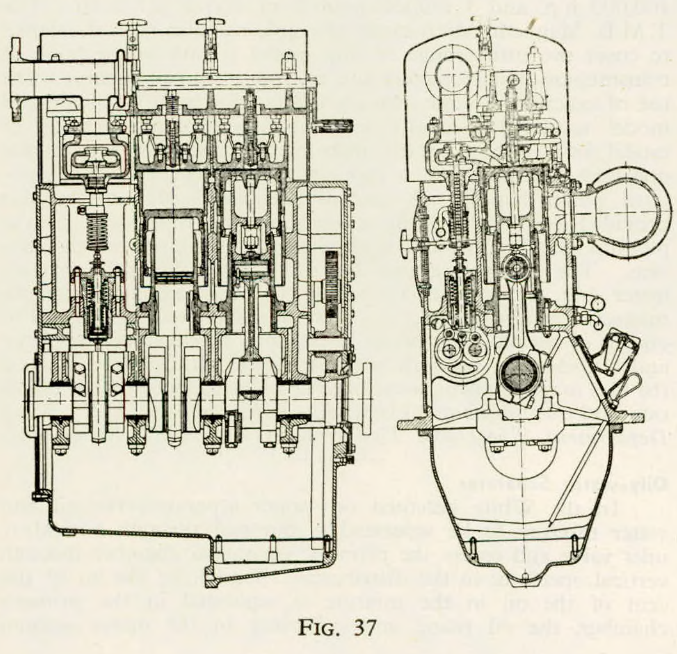


FIG. 37

Evaporators for Motor Liner

For the motor liner *Kungsholm,* under construction at the De Schelde yard, Flushing, for the Swedish American Line, two evaporating installations of the Schelde-Prache-Bouillon type will provide all the fresh water for drinking, showers and baths required for about 800 passengers. The evaporators, of

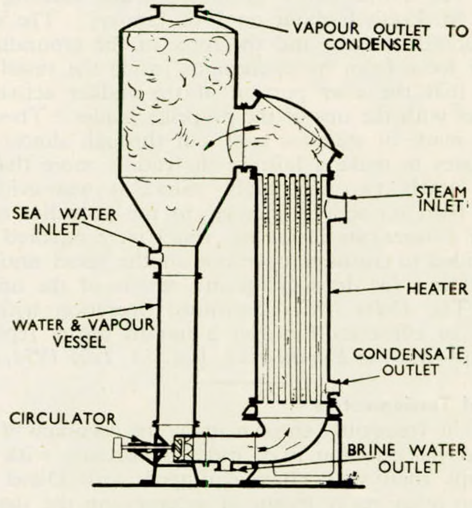
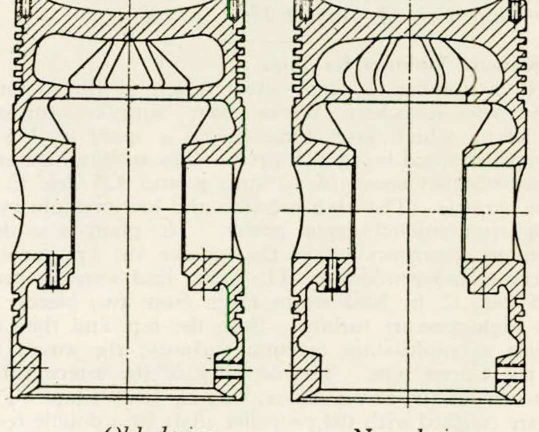


Diagram showing operation of evaporating plant in the motor liner Kungsholm

which the design is illustrated, are of the forced circulation type and the total capacity is 90,000 gallons daily. In order to avoid mechanical cleaning of the evaporators, chemicals are added to the sea water which prevent the precipitation of salt deposits and allow the plant to work for longer periods than is normally the case.—*The Motor Ship, Vol. 34, August 1953; p. 189.*

Pressure Charged Four-stroke Diesel Engines

The thermal and mechanical loading of a pressure-charged Diesel engine increases with the degree of pressure charging employed. Tests have shown that with small degrees of pressure charging the wall temperatures increase but little because of the cooling effect produced by the larger amounts of cold air



Old design *New design*

FIG. 1

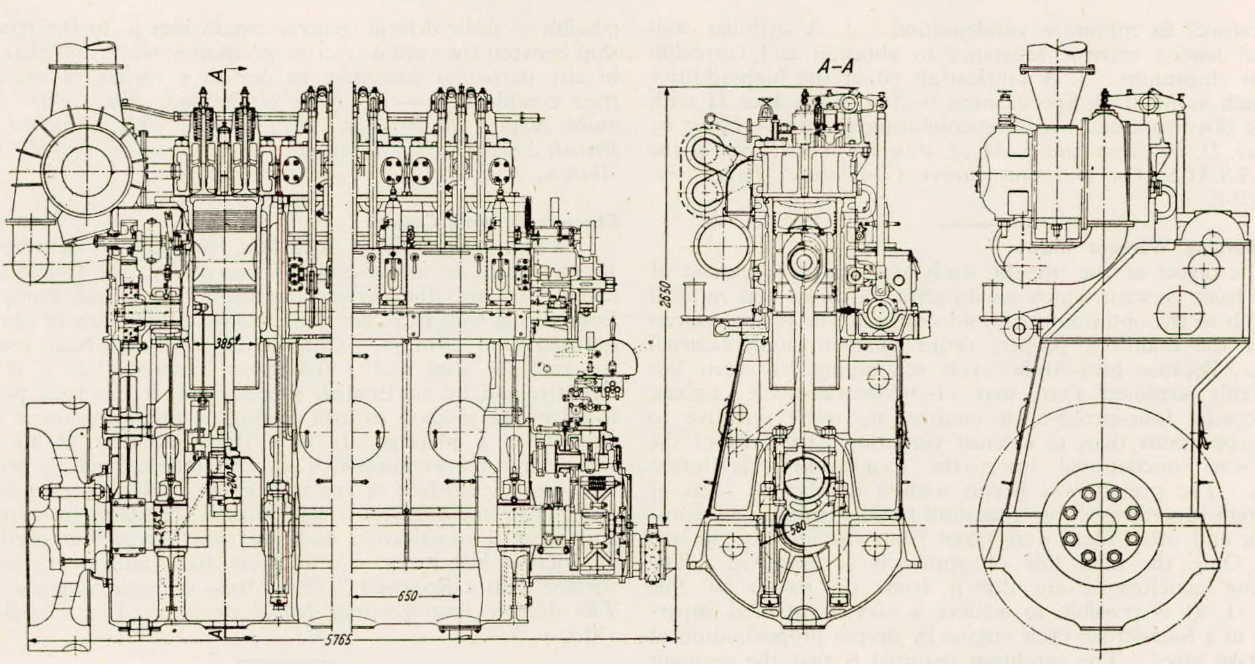


FIG. 8

introduced into the engine. But this observation does not apply to all engine parts to the same extent, and changes in design will have to be resorted to when pressure charging in excess of 30 per cent is intended. The greater heat supply to the piston can be mitigated by improving heat transfer conditions as far as the piston rings are concerned and by cooling the piston crown. It is essential to keep piston temperatures as low as possible, especially near the piston rings, in order to prevent sticking of the latter because of coking of the oil. As shown in Fig. 1, this can be achieved by relatively simple means. With the original piston a maximum of 30 per cent pressure charging would have been permissible, but by a few changes in the wall thicknesses of the piston it was found possible to maintain a pressure charging of 50 per cent with the temperature of the first piston ring still being 30 deg. C. below that of the original piston. This makes it possible to increase charging to 60 per cent. In this case lowering of the temperature of the first piston ring is due to a greater heat transfer via the second and third piston rings. If it is necessary to lower the temperature of the piston crown, this can be effected by oil jet cooling, which in the case of medium size and of high speed engines provides an economically more attractive solution than cooling by forced oil circulation which is more applicable to the large engine. In oil jet cooling the temperature rise of the oil does not exceed 10 to 16 deg. C. and there is thus no danger of premature ageing of the oil. Fig. 8 shows a pressure-charged M.A.K. Diesel engine of the trunk piston type. This engine type is built as six- or ten-cylinder unit for powers up to 2,000 h.p. Exhaust turbo-chargers of the Brown Boveri type are fitted. The specific fuel consumption of the ten-cylinder engine at one-half load is 163 grams per b.h.p. hour and 160 grams at full load.—*P. Kornacker and P. Grossmann, M.T.Z., Motortechnische Zeitschrift, Vol. 14, July 1953; pp. 207-211.*

Medium Speed Engines on Heavy Fuels

The increasing use of heavy fuels in the main propelling engines of ships has every indication of becoming general. It seems desirable from every angle that the auxiliary engines should be capable of operating on the same type of fuel, the paper states. This is dictated by fuel cost as well as convenience. The technical problems associated with heavy fuels in large low-speed main engines have had some years of development and study with such success that there are now close to 500 motorships operating on boiler fuels. Problems of the low-speed auxiliary engine (under 500 r.p.m.) may be similar to many of those problems encountered in the large low-speed main engines. However, as engine speeds increase, the old problems become more acute and new difficulties arise. Higher-speed auxiliary engines will be demanded because of increased power requirements, lighter weight, and smaller space. Such engines operating on regular Diesel fuels are already highly developed as typified by present-day locomotive and marine engines. The most desirable course would seem to be to adapt this type of engine to heavy fuels even though such a course involved development and design changes. In conclusion, the paper states that in spite of the great number of difficulties involved in burning residual fuels in high-output medium-speed Diesels, the attainment of the goal is worth the development effort. The work reviewed in this indicates that this can be accomplished by recognizing the problems and applying sound available engineering remedies. One point seems definitely established—a standard engine, well developed and giving eminently satisfactory service on good No. 2 Diesel fuels, cannot be expected to give equivalent performance nor reasonable maintenance when operating on residual fuels. To achieve this, the basic engine's accessories and certain parts must be retailored for the special conditions. This includes provision for burning the high viscosity and greatly contaminated fuels. Preparation of the fuel requires further study as it is a major problem. Filtering alone is not sufficient nor is centrifuging combined with filtering. Possibly double centrifuging—the purified and clarifier—may suffice but there is a further step holding promise which is now under development, comprising water washing of the fuel. It may seem strange to add water to the fuel when every effort has been made to remove it. However, a great percentage of the corrosive metallic contaminants in the fuel are present as water-soluble salts. Mixing water with the fuel results in such salts going into solution. Subsequent centrifuging separates the contaminated-ash-bearing water from the fuel and gives the engine a more digestible diet. In order to efficiently handle the higher viscosity fuel even at 200 deg. F. the injection system requires changing in nozzle orifices, plunger diameter, and mechanical strengthening of cams, rollers, and pump parts to withstand higher injection pressures. Improved fuel preparation will reduce ash and deposit tendencies in the exhaust system but Stellite or equivalent valve facing as well as valve rotation seem necessary. The solution to abrasive and corrosive cylinder wear lies in the combination of the following: 1. High jacket-water temperatures to minimize condensation. 2. A cylinder wall material having extreme resistance to abrasion and corrosion such as chromium. 3. A lubricating oil of the high-additive type such as Military Specification 0-2104 Series I or II with extreme dispersant and corrosion-inhibiting qualities.—*Paper by R. Pyles, D. P. Cryor and J. M. A. Van der Horst, read at the 1953 A.S.M.E. Oil and Gas Power Conference; Paper No. 53-OGP-6.*

Supercharging Without Blower

The object of the present study was the development of a clear basic concept which would afford a direct and rational approach to the optimum manifold design. The major interest was in the manifold proper, rather than in long external piping. Because four-stroke cycle scavenging has been less thoroughly explored than that of two-stroke-cycle engines, and because four-stroke-cycle engines are more sensitive to intake conditions than to exhaust conditions, the bulk of the work was concentrated upon the four-stroke-cycle intake system. The project was begun with a number of ideas of how "ram supercharge" or "manifold tuning" might be accomplished, and other ideas were taken from the literature in the field. Only the final line of approach is discussed. The following conclusions are drawn from the results of this study: 1. It is possible to achieve a very substantial supercharge in a four-stroke-cycle engine by proper proportioning of the intake pipes. The condition required is that the resonant frequency of the Helmholz resonator, consisting of the intake pipe and the cylinder with the piston in its mid position, be approximately twice the operating speed at which the peak supercharging effect is desired. The supercharging effect is maintained over a wide speed range. 2. It is possible to increase the power output of an engine of the four-stroke-cycle type by tuning the exhaust system in substantially the same fashion. The ratio of resonant frequency of the cylinder and pipe resonator to operating speed range has yet to be determined. 3. The scavenging of blowerless two-stroke-cycle Diesel engines is simple and rationally explained as a resonance phenomenon, considering the cylinder as a cavity resonated by the intake and exhaust pipes in parallel. The same approach also serves to explain blowerless supercharging. The characteristics of a two-stroke-cycle engine which is to scavenge itself without a blower are dependent upon the intake pipe as well as the exhaust pipe.—*Paper by H. W. Engleman, read at the 1953 A.S.M.E. Oil and Gas Power Conference; Paper No. 53-OGP-4.*

Cylinder Wear in Marine Diesel Engines

This report is based on recent publications on the subject of cylinder and piston-ring wear in marine main and auxiliary Diesel engines. In view of the small amount of information published regarding tests performed on engines of this class, use has also been made of data obtained from higher-speed engines. The author first considers the effects of wear; during the initial running-in period, the wear results in the sliding parts bedding-down, and the effectiveness of the seal between cylinder and piston rings increases. A lengthy period then follows during which the wear is small and the leakage and oil consumption are at a minimum; this is followed by a period during which the wear increases and the leakage finally reaches such proportions that the liner and rings must be renewed. Various types of wear can be distinguished, namely mechanical wear, scuffing, wear caused by impurities in the fuel or air, plastic deformation of the surface, and corrosion which may be of either chemical or electro-mechanical origin. The author discusses the effects on the wear of the structure, composition and hardness of the material, of machining and of the engine design; and the results of a number of tests carried out by various investigators to determine the effects of variations in fuel and lubricating oil on the wear are summarised. A section is devoted to the effects of such operating conditions as starting from cold, the temperature of the cooling water, the load and the speed. Finally, brief details of the two main methods of measuring cylinder wear are given. It appears that it is not possible to draw definite general conclusions as to the relationship between the various factors promoting wear, and that wear in any particular case may be due to a variety of causes or their combination.—*H. Visser, Netherlands Res. Centre Shipbuild. Navig. Report No. 7M, December 1952. Journal, The British Shipbuilding Research Association, Vol. 8, July 1953; Abstract No. 7747.*

Chromized Pump Casings

Wilton-Fijenoord, of Schiedam, have recently delivered the 17,300 tons d.w. motor-tanker *Hilversum* to the Ostzee Company. Her two fuel service pumps by the Comet Pump and Engineering Co., Ltd., are the first ever to be made of chrome-diffused iron castings. Chrome-diffusion has been used in the past for steel and a few other materials but it is only recently, and by an English process, that it has been possible to adapt the method to iron castings. The diffusion is not a deposit or a plating. It is a transformation of the iron surface into a chromium-rich alloy by inter-crystalline penetration into the surface of the iron under prolonged and intense heat. Chrome-diffused iron is highly resistant to corrosion and thermal oxydization and not only has a low coefficient of friction but it has an intensely hard and wear resisting surface with a Rockwell "C" hardness of approximately 70.—*The Marine Engineer and Naval Architect, Vol. 76, August 1953; p. 338.*

Repair of Marine Boiler by Welding

A paper by D. E. Herbert in New Zealand Engineering describes the methods used in the repair of two ship's boilers of identical pattern. On inspection of the first boiler, it was found that the lower part of the back end plate had intermittent cracking in the knuckle for a distance of approximately 3 feet. It was decided to remove the damaged part and replace with a patch which was to be partly riveted and partly welded. After all lagging, pipes and floorplates had been removed from around the outside of the back end, the collision chocks were then dismantled and the lower end laid bare. The damaged part was then marked off and centre popped ready for cutting. The part to be cut away measured approximately 4ft. × 10in. and went the full depth of the flange. The seam was a treble riveted one with rivets 1⅜ inches in diameter, countersunk in the shell plate. In all, 21 of these had to be removed. This was a very awkward job, for between the bottom of the boiler and the tank top was only 19 inches, which meant burning overhead. Cutting the rivets through 1⅜ inches of shell plate and ⅞ inch of end plate without damaging the holes was not an easy task. When the rivets were removed, the rest of the plate was cut away, and care also had to be taken in cutting the flange of the end-plate so as not to damage the shell plate. The damaged plate was then removed, and all edges cut and ground ready for welding, all preparations being made for a single vee butt. A template was fashioned from the old patch, and a new one (also prepared with a single vee butt) cut to size and fitted. The plate was offered in position, held, and the position of the rivets marked. It was then removed and rivet holes drilled and the edges finished. The plate was finally placed in position and bolted up tight with a 1⅜ inch bolt in every rivet hole until both the shell-plate and the patch were hard up. The patch was then welded with electrodes which possessed excellent hot forging properties. The sequence of welding was as follows: First, the joint between the line of rivets was welded, then along the top, after which the sides were completed. After every run the weld was caulked lightly with a pneumatic caulking tool, so as to remove all slag and to ease the weld. All the welding was done at the one time so as to weld the patch completely while the plates retained the heat. When the welding had been completed, there was no distortion of the plates. The bolts were then removed and the patch finally riveted. Having riveted up the patch, the welded joints were then ground nearly flush, and all seams adjacent to the patch caulked. The first row of screw stays was also caulked, and when all this had been completed, the boiler was ready for testing. The test was applied, the patch showing no signs of leaks or deformation whatsoever.—*The Marine Engineer and Naval Architect, Vol. 76, August 1953; p. 341.*

Supercharged Marine Engine

In Fig. 3 is shown a sectional elevation of a six-cylinder supercharged marine Diesel engine built by Mirrlees, Bickerton and Day, Ltd. The engine has a bore of 15 inches and a piston stroke of 18 inches. The in-line engines of this class are available in normally aspirated form with from five to eight cylinders, the powers ranging from 485 b.h.p. at 300 r.p.m. to 1,104 b.h.p. at 428 r.p.m. With the b.m.e.p. increased from 80lb. per sq. in. by turbo-charging up to about 120lb. per sq. in., the six-, seven- and eight-cylinder engines deliver from 870 b.h.p. at 300 r.p.m. to 1,656 b.h.p. at 428 r.p.m. With the 12-cylinder vee-class of engine, the output is 1,164 b.h.p. at 300 r.p.m., when normally aspirated, but when supercharged, the output is 2,484 b.h.p. at 428 r.p.m. The six-cylinder engine is a supercharged direct-reversing unit with a 12-hr. rating of 955 s.h.p. at 250 r.p.m. The temperature at the exhaust is 700 deg. F., at the cylinder, while that at the turbo-charger inlet is 775 deg. F. At 330 r.p.m. the piston speed is 990ft. per min. and the maximum cylinder pressure 950lb. per sq. in., while the fuel-injection pressure is 180 atm. The exhaust-gas turbo-charger runs at 12,500 r.p.m., the exhaust pressure being 8·2 inches of mercury. The mechanical efficiency is given as 85 per cent, the thermal efficiency being 39·2 per cent. The cylinder casings carrying close-grained cast-iron liners are mounted on top of the columns; the cylinder heads are also of cast iron and house two inlet and two exhaust valves, also the air starting, cylinder relief and the centrally disposed injection valves. The exhaust valves are caged to facilitate easy removal. The cylinder head cooling arrangements are particularly interesting, each head being divided into an upper and lower chamber. Vanes are cast in the water space to direct the flow of water and ensure that the hottest part of the cylinder head, around the fuel nozzle, is adequately cooled. Reversing is effected pneumatically, using a cam changeover mechanism so that the follower gear is withdrawn, moved longitudinally and re-inserted over the appropriate cams. The return flow of piston-cooling oil may be observed through glass inspection ports on the starboard side of the engine. The fuel consumption at the 12-hr. rating is 0·351lb. per b.h.p. hr. No propelling engines of this class are yet in service on residual fuel, although developments are proceeding satisfactorily in this direction as an outcome of research on a three-cylinder engine now running on fuel of about 950 sec. Redwood No. 1 at 100 deg. F. and with 2½ per cent sulphur content.—*The Motor Ship, Vol. 34, August 1953; pp. 210-212.*

Generating Irregular Waves

Natural waves such as those to which a ship is subjected when proceeding in a seaway are not usually of regular form. It follows therefore that the conventional type of wave-maker employed in experiment tank work does not exactly simulate the conditions experienced at sea. Attention may therefore be drawn to a paper entitled "Laboratory methods for the generation of irregular waves whose principal characteristics correspond to those of natural waves" recently published in "Comptes Rendus" de l'Academie des Sciences (Paris). The authors give brief details of a statistical analysis of wave motion from which it appears that there are three parameters involved, viz. the average amplitude, the average frequency, and an irrgularity factor which is a function of the amplitudes and frequencies of the large number of component waves that make up the wave pattern. The conventional type of wave-maker in which two or three waves of different amplitudes and frequencies are combined to make up a wave pattern, the authors consider to be unsatisfactory. They suggest a wave pattern which for practical purposes corresponds sufficiently closely to the natural wave pattern can be obtained by using a cam or an electronic or electro-magnetic timing device which subjects the wave-maker to a cycle comprising 20 to 50 different wave formations. A second method is described in which air jets are employed blowing across the surface of the wave.—*Shipbuilding and Shipping Record, Vol. 82, 2nd July 1953; p. 3.*

Marine Boiler

This invention concerns a boiler which supplies superheated steam to a turbine. The thermal efficiency of such plants can be increased by preheating the combustion air, for which

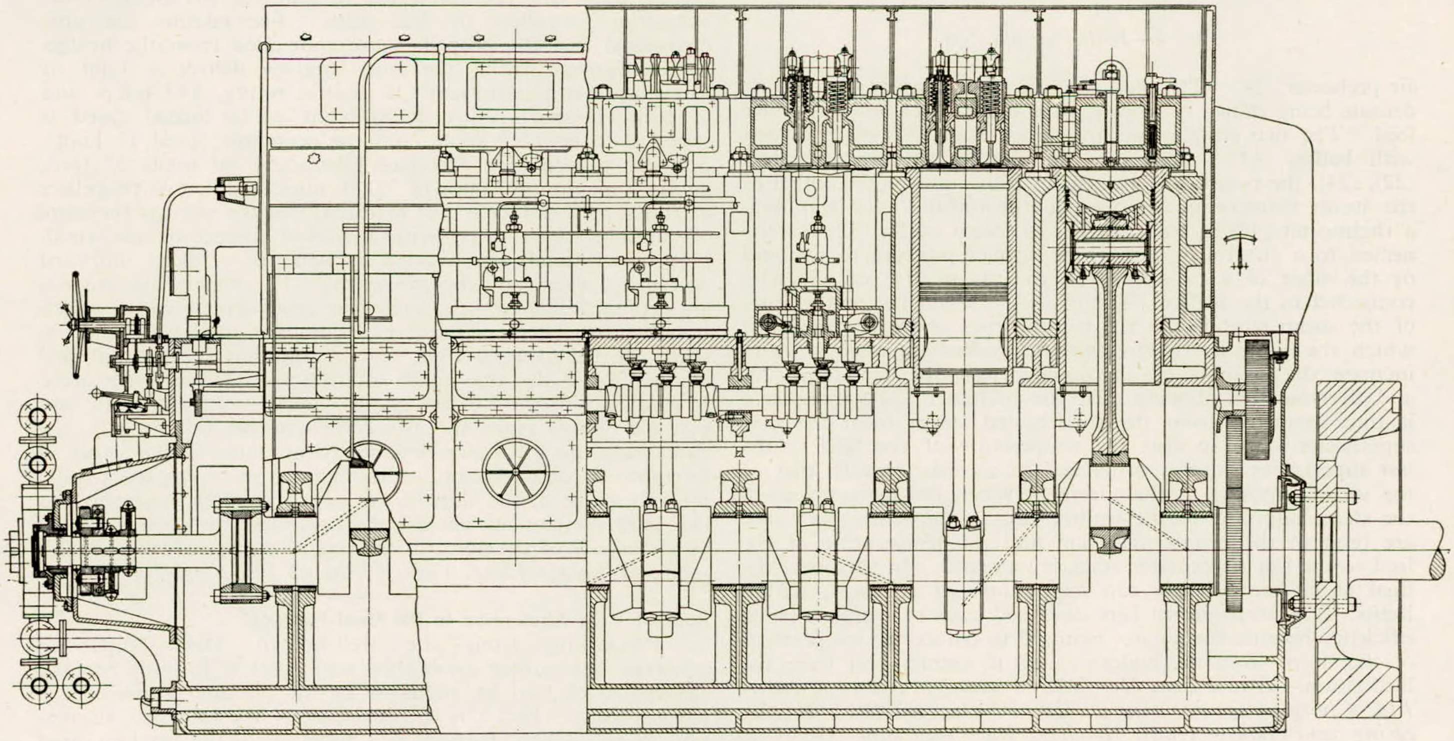


FIG. 3—*Longitudinal section through a six-cylinder supercharged engine*

purpose low-pressure steam is sometimes bled from the turbine and made to give up some of its heat to the air. Fig. 4 depicts the outline arrangement of a boiler with a steam drum (10) and a water drum (11) connected by steam generating tubes (12) and a two-stage superheater (14), (16). The boiler is fired by means of burners (18) to which air is supplied through a duct (20). This air duct (20) is divided by a partition to form two parallel passages which contain two indirect heat exchange aid preheaters (22) and (24). Steam from the steam drum (10) passes through a line (26) to the first superheater stage (14) and thence through a line (28) to the air preheater (22). In the latter, its temperature is lowered, and it then passes through a line (30) to the last superheater stage (16). From the outlet (32) the steam passes through a line (34) to a turbine (36). Bled steam from the turbine is drawn off from one of a number of outlets (38) and passed through a line (40) to the

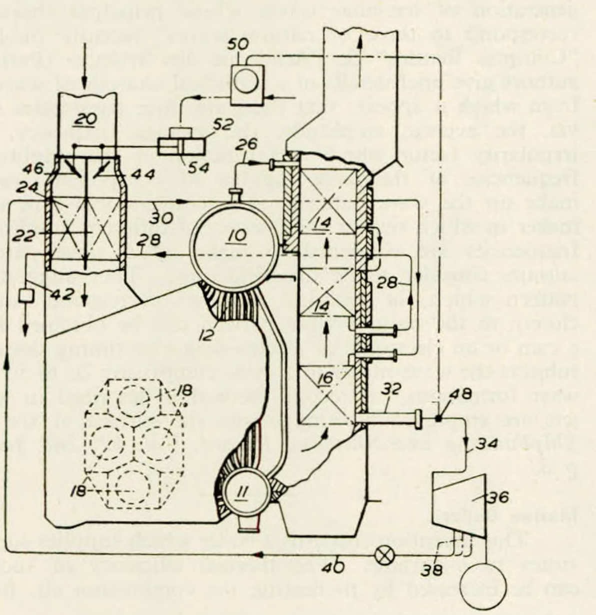


FIG. 4—*Brit. Pat. 690,708*

air preheater (24). The bled steam is thus condensed, the condensate being drawn off through the line (42) to serve as boiler feed. The two air passages in the air duct (20) are provided with baffles (44), (46). The air flows past the preheaters (22), (24), the position of the baffles being under the control of the steam temperature supplied to the turbine. To this end, a thermostat (48) in the superheated steam outlet (32) is connected to a controller (50) which applies pressure to one end or the other of a cylinder (52), the piston (54) of which is connected to the baffles (44) and (46). When the temperature of the steam supplied to the turbine rises above the limit for which the controller (50) is set, the baffles are reset so as to increase the amount of air flowing past the preheater (22) and decrease that flowing past the preheater (24); more heat is thus extracted from the superheated steam from the first superheater stage so that the temperature of the feed to the last superheater stage is controlled in accordance with that of the steam supplied to the turbine. When the temperature of the steam supplied to the turbine drops, the baffles (44), (46) are reset in the reverse direction and the temperature of the feed to the last superheater stage is increased. In this way, the final steam temperature can be maintained constant, within limits. The arrangement here described leads to a high thermal efficiency because the air not required to reduce the temperature of the steam from the boiler is used to extract heat from the bled steam.—*Brit. Patent No. 690,708 issued to Foster Wheeler, Ltd. Complete specification published 29th April 1953. Engineering and Boiler House Review, Vol. 68, June 1953; p. 195.*

Modern European Coasters

In most of the maritime countries of Northern Europe an upper size limit of 500 tons gross is set for ships which are to be classed as coasters and are to be allowed a two-watch manning scale. With ships of this size the number of crew to be carried is an important factor in successful operation, principally because of the wages involved but also because the provision of any additional accommodation will directly reduce the earning capacity of the ship. Since the war there has been keen competition in the short sea and coastal trades of Northern Europe, in contrast to the easier conditions which existed for quite a considerable period for ocean-going tonnage, and just as before the war the depressed state of the deep sea tramping market led to the introduction of "economy" types of ship, so has the last year or so seen the entry into service of a number of new coaster designs, intended to obtain the maximum earning capacity within the 500-tons gross limit. Although classified as coasters these ships are not, of course, confined to the coastal waters of their country of origin. They are able to trade freely in the Baltic and the North Sea, and in at least one case maintain regular services to the Mediterranean. In all the vessels described in the original article, full use has been made of modern practice in ship design; but even more advanced ideas are reflected in a recent design by a Dutch naval architect, Mr. Frank Rijsdijk. The most unusual feature of this design is the propelling machinery, which consists of high-speed Diesel engines driving twin Voith-Schneider propellers. It might have been thought that a design which includes such innovations would not be readily acceptable to shipowners, who for obvious reasons must often be conservative where new technical developments are concerned. However, this has not proved to be the case, as it is reported that two orders have already been received from Sweden, another is under negotiation from the same country, and a French firm is also interested. The design has been named *Dutch Glory*. The vessel has a tonnage of 499 tons gross, 250 tons net, and 1,060 tons d.w. Length o.a. is 214ft. 3in., and draught loaded is 11ft. 9½in. The main propelling machinery consists of four supercharged high-speed Diesel engines, made by the Caterpillar Tractor Company. They are six-cylinder engines, with a normal speed of 1,500 r.p.m. and a maximum speed of 2,000 r.p.m. and are connected in pairs to the twin Voith-Schneider propellers by Vee belts. The engines are unidirectional and the propellers are controlled from the bridge. At continuous rating the four engines deliver a total of 680 s.h.p.; at intermittent (12 hours) rating, 840 b.h.p. and at peak (1 hour) rating 1,100 b.h.p. The loaded speed is expected to be 12·7 knots, and the economic speed 12 knots. Stowage capacity for fuel and lubricating oil totals 97 tons, giving a maximum radius of 7,000 miles. With two propellers of this type a rudder is not required, and the stern is therefore of cutaway spoon shape, with a marked absence of deadwood. Directional stability is assisted by two fins or skegs outboard and just forward of the propellers. The four main engines are arranged side by side across the small engine room, which is right in the stern of the ship. The drive to the propellers is taken off the forward ends, and in addition the two inboard engines drive the two main alternators at their after ends. Immediately forward of the engines are the two propellers, and forward of this again and two compartments side by side, one containing most of the engine room pumps, the other a refrigerated compartment. Although the main engines require the full width of the ship at the stern, the aftermost cabins at this level are situated abreast the propellers, with other cabins on either side of the pump room and refrigerated compartment. —*World Shipbuilding, Vol. 3, August 1953; pp. 107-109.*

Radiant Heat Absorption in Oil Fired Furnace

Proceeding from the well-known Stefan-Boltzmann equation, the author establishes an empirical formula for the absorption of heat by radiation in the oil fired furnace of a marine boiler. Fig. 5 is the final plot of the radiation absorption in an oil fired furnace as a function of the net heat fired per square foot of effective projected radiant heat absorbing area and the per cent excess air. For convenience the temperatures of the gases leaving the furnace have been noted on this curve. The net heat fired plotted as the abscissa for Fig. 5 is the product of the pounds of oil fired per hour and the net heating value at constant volume of the oil. The utility of this curve lies in the fact that once the net effective heat absorbing area has been evaluated, the absorption and the furnace exit temperature can be quickly evaluated for many load conditions. Also this curve can be used if some of the assumptions are changed. For instance, assume that the air entering the furnace is at 300 deg. F. instead of the 100 deg. F. used to construct the curves. In this case it is only necessary to add the difference in heat content of the entering air above 100 deg. F. to the net heat fired and then use the curves as before.—*I. Granet, Journal of the American Society of Naval Engineers, Vol. 65, May 1953; pp. 319-327.*

Foil-type Strain Gauges

The majority of the advantages which foil-type strain gauges have over wire gauges are of particular interest to the shipbuilding and heavy engineering industries. The junction of the fine wire filament to the lead out tags is a vulnerable point and failure here can be costly. The foil gauge has no such discontinuity and is much more robust, while being entirely insusceptible to moisture. The ratio of contact surface area of the resistance element to volume is very high for the foil gauge which means that the heat dissipation characteristics and hence its current-carrying capacity is many times that of the wire gauge. This feature is particularly useful for the voltage output from a strain gauge per unit strain is directly proportional to the current flowing in it. This represents a considerable increase in sensitivity when the gauges are required to operate voltage-sensitive recorders or indicators. The power output of a gauge is proportional to the square of the current input. Therefore, a tenfold current increase yields a hundredfold increase in power output. In the course of their experimental work on aircraft and helicopters, Saunders-Roe, Ltd., were frequently faced with the problem of accurately measuring torque in such a manner as to absorb the minimum of power in the measuring device. It is rarely possible to measure torque by complete absorption in a dynamometer as is the practice on engine test beds, for example: strain gauges are generally used and they are applied at 45 degrees to the principal axis of the shaft or torque-carrying member, usually as a four-gauge bridge to give full compensation for shaft bending, etc. There are two parallel patterns sensitive to strains at 45 and 135 degrees to its long axis and the gauge is made accurately as a parallel sided ribbon in 6in. or 12in. lengths. All that has to be done is to cut off the length required, wrap it round the shaft and bond it in position. The electrical supply and signal wires are soldered on and connected into the circuits. It is not even necessary to mark out the shaft (note that this can be a difficult job using normal gauges). The strains in the shaft are integrated circumferentially and not picked up at only four points as in the case of the wire gauge application. The output can be taken out through sliprings.—*The Marine Engineer and Naval Architect, Vol. 76, August 1953; p. 338.*

New Bosphorus Ferries

Four new Bosphorus ferries are propelled by Sulzer Diesel engines. All four vessels have been built by the Société Anonyme des Anciens Chantiers Dubigeon at Nantes, to specifications drawn up by a technical committee of the Devlet Deniz Yollari ve Limanlari. The Sulzer propulsion engines, two of which are installed in each ship, have been constructed under licence by the Ateliers et Chantiers de la Loire at St. Denis. The same company has also supplied the four propellers for each vessel, which are driven by the engines through disconnectable couplings. The *Kizkulesi* and the *Kasimpasa*

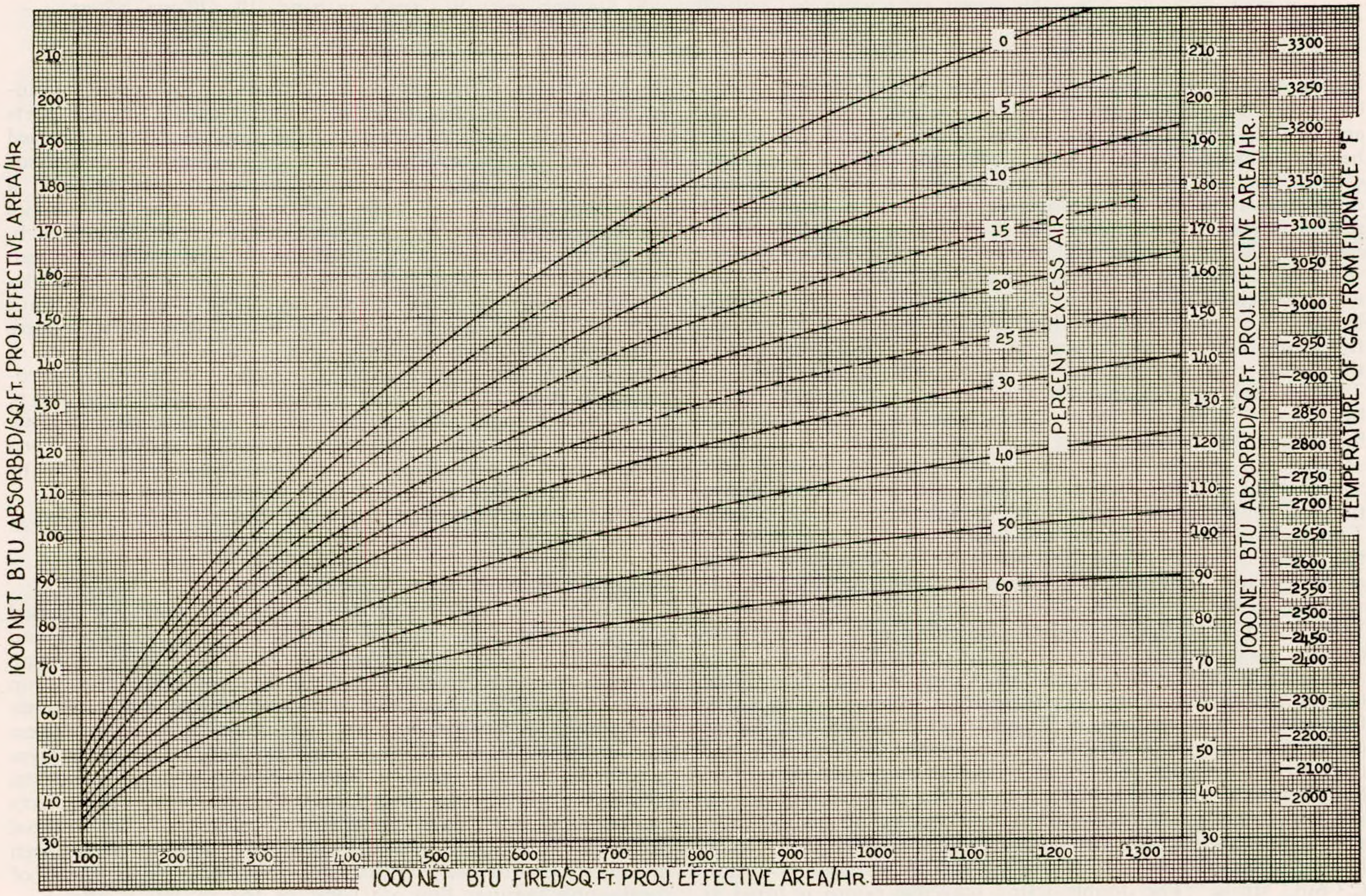


FIG. 5—*Radiation absorption in an oil fired furnace*

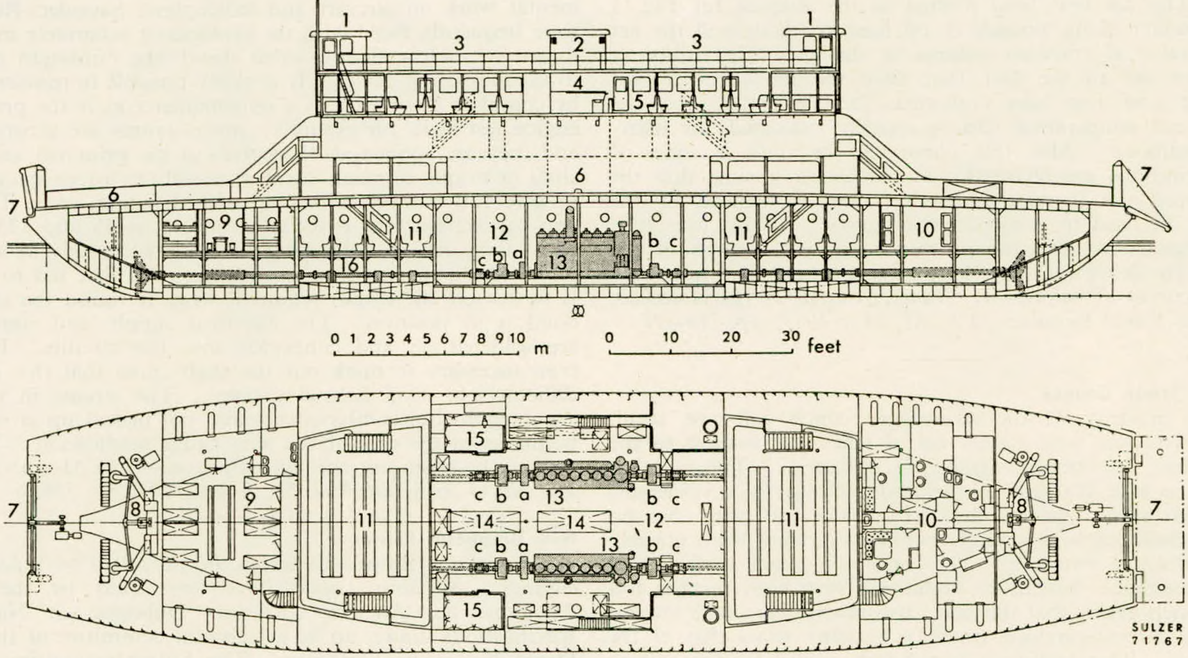


FIG. 21—*Plans of the new Sulzer-engined Bosphorus ferries*
The arrangement and equipment of the Kasimpasa *and* Kizkulesi *of 300 tons deadweight were reproduced almost unchanged in the two smaller ferries* Karaköy *and* Kuruçesme *of 200 tons deadweight. The differences are limited almost entirely to the dimensions.*

(1) Navigating stations; (2) Funnels; (3) Gangways connecting navigating stations; (4) Bar; (5) First-class saloon; (6) Runway on main deck; (7) Adjustable ramps for loading and unloading; (8) Steering gear; (9) Crew's quarters; (10) Officers' cabins; (11) Second-class saloons; (12) Engine room; (13) Propulsion engines; (a) Flywheel, (b) Disconnectable coupling, (c) Thrust bearing; (14) Auxiliary engines; (15) Store and workshop; (16) Shafting tunnel

have a deadweight capacity of 300 tons each, while the smaller but otherwise similar motor ferries *Karaköy* and *Kuruçesme* are of 200 tons deadweight. The *Kizkulesi* and *Kasimpasa* have a loaded displacement of 1,240 tons each at a draught of 9ft. 10in. The length is 197 feet, while the width of the car-deck runway is 25 feet. Each of the boats is able to take thirty-six medium-sized motor-cars or sixteen lorries of 12 tons each, in addition to the 700 passengers. The normal service speed of 12 knots is comparatively high for ships only 197 feet long, especially as their breadth is almost one-fifth of their length. The two propellers fitted at each end of the ferries and the four rudders, each in line with a transmission shaft, give the vessels excellent manœuvrability in spite of their breadth. The rudders are remote-controlled in pairs from the bridge by way of hydraulic servo-motors, less than fifteen seconds being needed for their greatest swing of 35 degrees. A diameter of 590 feet, or about three times the length of the ship, was specified for turning, but in fact these vessels can turn almost in their own length even when using only two propellers. These useful features permit unusually safe navigation even in the crowded waters of the Bosphorus. The propulsion output necessary for attaining the maximum speed of 12·5 knots specified in the contract was calculated as 2,000 s.h.p., or about 1,000 b.h.p. per engine. Sulzer engines of type 7TS36 were chosen, their output being 1,050 b.h.p. at 250 r.p.m. The arrangement of the two engines is shown in Fig. 21. Their output can be transmitted as required to the propellers either fore or aft. The shafting is engaged and disengaged by means of Messian couplings which can be operated either from the engineroom or from the bridge. The shafts of the secondary half-couplings are connected direct to the main shafts, which are supported in Michell thrust bearings. The two 7TS36 propulsion engines are direct-reversible two-stroke units, each having seven cylinders of 360 mm. (14·2in.) bore and 600 mm. (23·6in.) stroke. The double-acting scavenge pump is fitted at the forward end of the engine and delivers air into a scavenging-air receiver from which the main cylinders are fed through ports of new design. The piston rod of the scavenge pump is carried up at the top to drive a single-stage starting-air compressor which has an intake of about 1,000 cu. ft. per hr. and a discharge pressure of 425lb. per sq. in. The two ferries *Karaköy* and *Kuruçesme*, of 200 tons deadweight capacity each, are simply smaller versions, all of their equipment having been reduced almost to scale. The main dimensions and capacities of these smaller ships are: length o.a., 180ft.; beam, 45ft. 6in.; width of runway on car deck, 22ft. 4in.; displacement, loaded, 950 tons. Carrying capacity: motor cars, 25, or lorries of 12 tons each, 10; passengers, 200. Each of these ships is propelled by two 5TS36 Diesel engines developing 1,500 b.h.p. in all at 250 r.p.m. The engines are of the same type as those installed in the larger ferries, but have five instead of seven cylinders. The arrangement of the Messian couplings for the drive of the shafting to the two propellers fore and aft is also the same as described above.—*J. Biaggi, Sulzer Technical Review, No. 1, 1953; pp. 12-22.*

Resistance of Single-screw Coasters

Some years ago a great deal of research work was carried out at the National Physical Laboratory by F. H. Todd and J. Weedon. Since that time there have been major changes in model testing technique, and the analysis of model propulsion results, which have made earlier data unreliable and sometimes misleading. A new research for these vessels is now in progress and an outline of the whole scheme is given. The paper gives results of the first part of the investigation into resistance, which is confined to a length:breadth ratio of 6. The effects on resistance of block coefficient, position of longitudinal centre of buoyancy, midship area and draught have been investigated, and the results are discussed. The effect of eliminating laminar flow on some of the models is shown.—

Paper by J. Dawson, read at a meeting of The Institution of Engineers and Shipbuilders in Scotland, 10th March 1953.

Hatch Corner Design

The large number of welded ship failures early in World War II focused attention upon the destructive potential of structural notches. Complete elimination of notches from a ship is impossible. However, elimination of the destructive effects originating at structural notches is generally possible through design. This has been demonstrated by successive stages of development of satisfactory all-welded hatch corners. The original hatch corner of the Liberty vessel proved to be one of the worst crack initiators. As a result this hatch corner was modified and those of later Liberties were redesigned. In the Victory class of vessels, and other vessels subsequently constructed, additional design improvements were incorporated into the hatch corner. Because of the importance of the problem, complete service and fracture records have been analysed and full-scale laboratory tests of hatch corners, duplicating as closely as possible construction and service conditions, were performed. Results of these tests have been analysed for correlation with actual service performance. In view of the significant volume of accumulated service data and associated laboratory test data available, the Ship Structure Committee has undertaken to evaluate and interpret these data in terms of design features intended to produce crack-resistant hatch corners in welded steel vessels. The results of such undertakings are contained in this report.—*The Welding Journal, Vol. 32, July 1953; pp. 316-s - 324-s.*

Oil-engined Liners

In an article dealing with the characteristics of oil-engined liners, the author draws attention to the fact that the growing demands for fresh water have made it imperative even in oil-engined ships to fit evaporators of large size. Among the first liners to have such installations were the two Dutch ships *Oranje* and *Willem Ruys,* with their large requirements for the Far Eastern run. The new *Kungsholm,* though a trans-Atlantic ship, is one of the latest exponents. She has twin independent evaporators, each with a capacity of 45,000 gallons per day. Average consumption of fresh water in the *Kungsholm,* even in trans-Atlantic traffic, is expected to be 50 gallons per person per day, and the figure rises to 90 gallons in the tropics. These evaporators work on steam generated by exhaust gas from the main engines.—*A. C. Hardy, The Journal of Commerce, Shipbuilding and Engineering Edition, Vol. 128, 23rd July 1953; p. 5.*

Admiralty Welding Committee

A further report has been published by the Admiralty Ship Welding Committee on the trials which were carried out with the steamships *Clan Alpine* and *Ocean Vulcan.* These vessels were sister ships, one of riveted and the other of welded construction. Comparative experiments with the two ships under various states of loading in still water were followed by investigation of the loads imposed on one of the ships (*Ocean Vulcan*) during prolonged periods of sea service. From the results of these trials it is intended to calculate the probable maximum stresses at sea for both the welded and the riveted ship. The report now published, Report No. R.8, is concerned with the sea trials carried out on the *Ocean Vulcan.* The conclusions reached are as follows: (1) The greatest range of vertical bending moment derived from these observations at sea was 190,000 tons-ft., corresponding to a range of stress of 8 tons per sq. in. at the top of the steer strake amidships. There is no experimental evidence to show the actual division of this range between hogging and sagging, but theoretical considerations suggest that the sagging moment constituted about 55 per cent of the total range. (2) The above range of vertical bending moment was associated with waves 35 feet high and between 600 and 700 feet long. According to oceanographical data, waves of even greater severity may be encountered, but it is estimated that for the *Ocean Vulcan* the maximum bending moment range due to waves is never likely to exceed 260,000 tons-ft. This corresponds to a stress range of 11·0 tons per sq. in. at the top of sheer strake amidships. (3) Slamming action was observed to cause stress ranges of up to ± $1\frac{1}{2}$ tons per sq. in. at the top of the sheer strake amidships. In the few cases in which slamming stresses were recorded these increased the peak sagging stress but did not increase the peak hogging stress. (4) Horizontal longitudinal bending moments were observed in almost all wave conditions, and sometimes caused stresses of similar magnitude to those due to the concurrent vertical bending. The maximum range of horizontal bending moment observed was 80,000 tons-ft. corresponding to a stress range of $2\frac{1}{2}$ tons per sq. in. at the sheer strake amidships. The horizontal and vertical bending moments were frequently in phase, and this resulted in the stress range on one sheer strake (or bilge) being very different from that on the other. (5) The greatest horizontal bending moments occurred when the inclination of the wave advance relative to the ship's course was between 20 and 50 degrees. In head or following seas, which caused the highest vertical bending moments, the horizontal bending moments were relatively small. (6) Torsion moments derived from the records were small and therefore had little effect on the stresses on the hull girder. (7) The records show that, within the limits of the experimental accuracy, the water pressures measured on the hull were in agreement with those predicted from the trochoidal theory of waves. (8) Theoretical considerations, supported by the experimental evidence show that for this ship in a seaway, the dynamic terms in the determination of the vertical longitudinal bending moments at amidships tend to cancel each other. In consequence a reliable estimate of these bending moments for any particular wave length and height in head or following seas can be made from the assumption that the ship is in static equilibrium on the wave, provided the Smith correction is applied. (9) For the *Ocean Vulcan,* the greatest range of bending moment at amidships occurs when the length of the wave approximates to the length of the ship (416ft. b.p.) and according to oceanographical data the maximum height ever likely to occur for such a wave is about 35 feet.—*The Shipping World, Vol. 129, 12th August 1953; p. 128.*

Sea-going Qualities of Ships

Loss of Speed at Sea. Möckel's collection of observations from ships in service seems to indicate that the speed loss is roughly proportional to the square of the wind force. For ships trading between Germany and South America the speed reduction at wind force 4, Beaufort scale, was 10 per cent for a 10-knot ship, 2·7 per cent for a 12-knot ship, and 1·5 per cent for a 19-knot ship. In all trade routes, at wind force 4, the loss of speed increased heavily when the shaft horsepower per ton displacement was less than 0·4, but with s.h.p./Δ greater than 0·6 the speed loss in per cent was very nearly constant and independent of the power. Fig. 1 gives the speed loss in the North Atlantic at wind force 6 as a function of

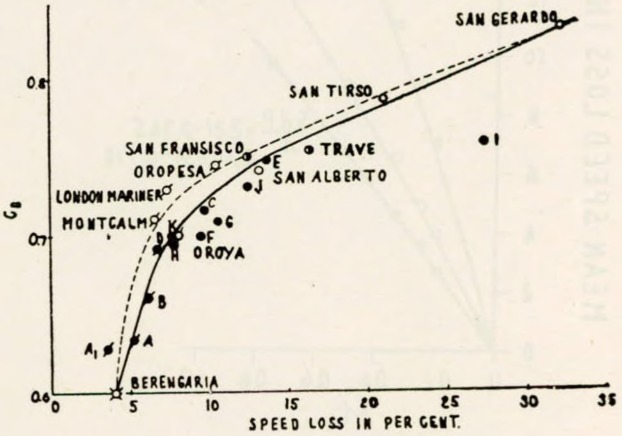


FIG. 1

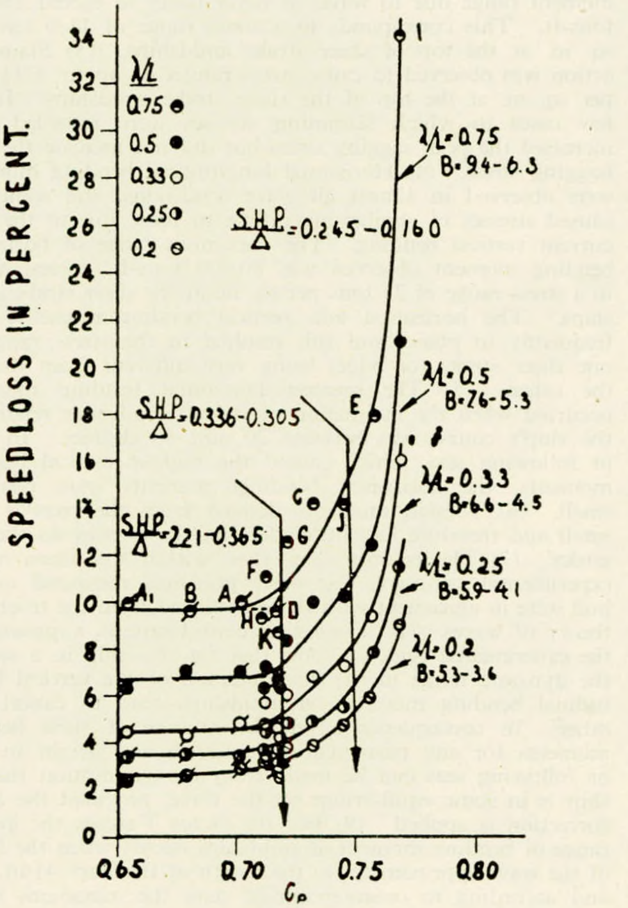


FIG. 2

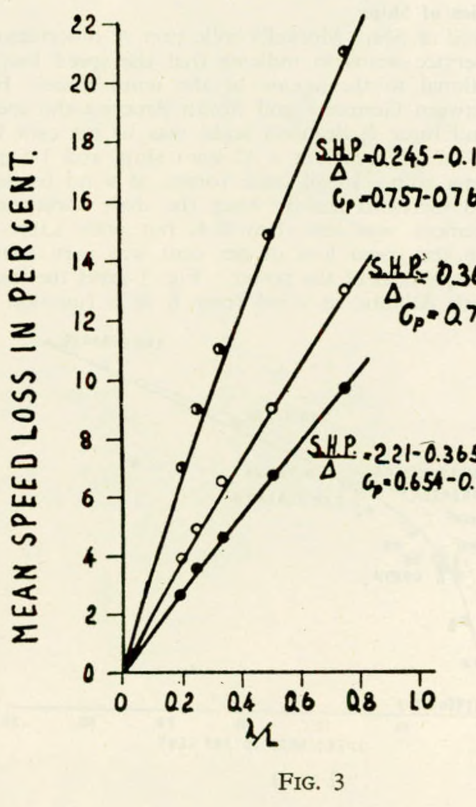


FIG. 3

the block coefficient C_B for German ships compared with the observations published by Kent for British ships in the same ocean with waves 6 feet high. The mean curve has a critical knee separating moderate losses from heavy losses at a block coefficient of about 0·72. The position of this knee will be somewhat higher in lower waves or more moderate wind. In Fig. 2, which gives the speed reduction in the North Atlantic as a function of the prismatic coefficient, C_P, the figures have been subdivided according to the ratio between wave length λ and ship length L. Mean curves for different constant values of this ratio have been drawn. The curves can be subdivided into three characteristic parts, a horizontal, an intermediate, and a third part where the speed losses rise rapidly. Fig. 3 gives the cross-curves obtained by vertical sections through the three groups in Fig. 2. They seem to indicate a nearly linear rise of speed reduction with ratio of wave length to ship length. The ships studied have all been fairly large vessels, which is probably the reason why the greatest value given for λ/L in Fig. 2 is 0·75. One would expect the curves of Fig. 3 to reach a maximum at λ/L somewhat greater than unity and then to drop again. Allan's tank tests with drifters at $V/\sqrt{L} = 0{\cdot}98$ indicate a maximum speed reduction at λ/L between 1·3 and 1·4 and considerably less reduction at larger wave length. At $\lambda/L = 2{\cdot}2$ the speed loss has been reduced to about the same as for $\lambda/L = 0{\cdot}32$. From the reference to λ/L in Figs. 2 and 3 it will be seen that small ships suffer a much heavier speed loss than large ones when running in the same sea.—*G. Vedeler, European Shipbuilding, Vol. 2, No. 2, 1953; pp. 28-36.*

New Cargo Liner

The combined passenger and cargo liner *Braunschweig* is the latest addition to the fleet of the Hamburg-Amerika Linie and is one of a number of 10,000 tons d.w. vessels delivered for the Far Eastern service operated jointly with the Norddeutscher Lloyd. Length of the vessel is 159·06 m. o.a. and 146·50 m. b.p. Breadth moulded is 19·10 m. Service speed is 17·5 knots. The vessel is driven by a two-casing A.E.G. steam turbine of 9,000 s.h.p. connected with the single screw through a double reduction gearing. Steam is supplied by two La Mont boilers at 42 atm. g. and 450 deg. C. steam temperature. Each boiler has an evaporative capacity of 18 tons per hour. There are two electrically operated Saacke oil burners per boiler giving an infinitely variable firing rate of 80 to 840 kg. of oil per hour.—*Schiff und Hafen, Vol. 5, July 1953; pp. 310-314.*

Sulphur Trioxide Content of Flue Gases

An investigation into the effect of fuel, load and CO_2 on the SO_2 content of flue gases from an oil-fired boiler was planned, and the results analysed, statistically. Five oils, the sulphur contents of which ranged from 0·75 to 3·55 per cent, were tested over a wide range of boiler operating conditions. The SO_2 produced under each set of conditions was measured at various stages through the boiler, (a) with the B.C.U.R.A. dewpoint meter, and (b) by chemical analysis. The results, based on 161 dew-point readings and 72 chemical analyses under 44 different conditions, show that SO_2 can be produced in oil-fired boilers in amounts sufficient to cause some trouble from low-temperature surface corrosion, if arrangements are not made either to prevent its formation or to remove its effect. The statistical survey has indicated relationships between certain variables, but with other variables no relationship has been found on the basis of the data available. In particular, it has not been possible to establish a direct relationship between the SO_2 (as measured either analytically or by the dew-point meter) and the total sulphur in the oil. The results show that with all the oils tested, from 1 per cent to 3 per cent of the SO_2 is oxidized to SO_3, giving rise to the dew points of from 250 deg. to 300 deg. F. The work has confirmed the results of previous laboratory investigations.—*P. F. Corbett, Journal of The Institute of Fuel, Vol. 26, August 1953; pp. 92-106.*

ENGINEERING ABSTRACTS

Section 3. SHIPBUILDING AND MARINE ENGINEERING

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First 18,000-ton Standard Tanker for Shell Fleet

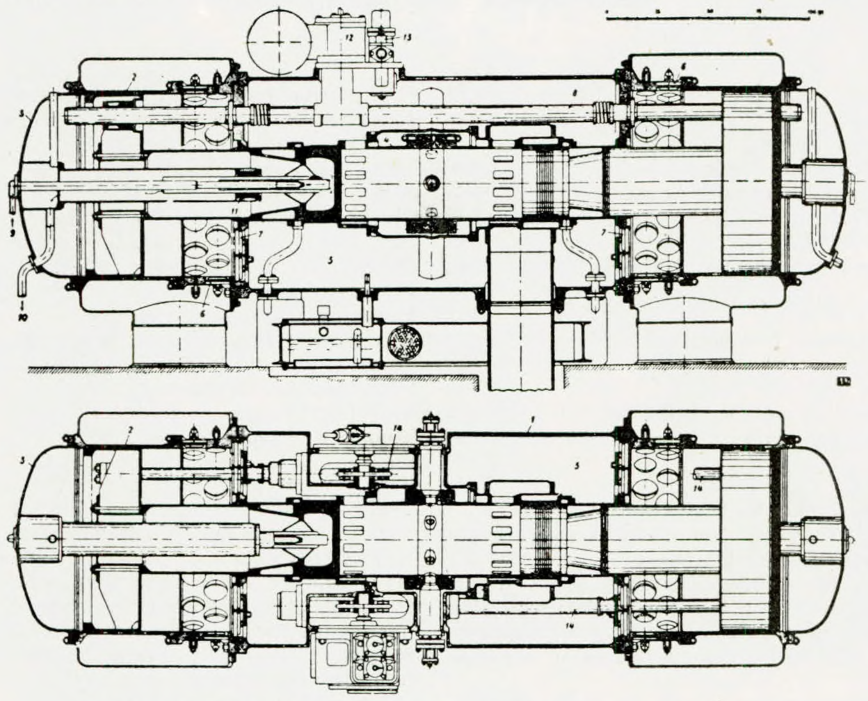
Harland & Wolff, Ltd., have completed at their Belfast Shipyard for the Anglo-Saxon Petroleum Co., Ltd., the 18,000 ton d.w. tanker *Harpa,* first of a series of 58 similar vessels for general purpose duties in the Shell fleet. The design incorporates many new features including a permanent tiled swimming pool. The *Harpa* has the following principal dimensions: —

Length between perpendiculars	530ft. 0in.
Moulded breadth	69ft. 3in.
Moulded depth to upper deck	39ft. 0in.

The vessel is of usual tanker design with machinery aft, poop, bridge and forecastle decks connected by fore and aft gangways, boat deck aft, and upper bridge and navigating bridge amidships. The hull is constructed on the combined transverse and longitudinal system of framing and is divided by bulkheads into 33 cargo oil-carrying compartments. There is one main cargo pumproom and a forehold pumproom, a cofferdam, a gastight cargo hold forward, and under this space a deep tank for oil fuel. Wing bunkers forward of the engine room and the double bottom under the motor room are also arranged for the carriage of oil fuel. The peak tanks are arranged for water ballast. Welding construction has been adopted to a considerable extent. Comfortable accommodation is provided amidships for the captain and officers, and on and under the poop deck for the engineers, officers, petty officers and crew. A large dining saloon, petty officers' mess with adjoining smokeroom, and mess rooms for crew and catering staff are arranged on the poop deck convenient to the galley and pantries, the crew's recreation room being situated under the poop deck and the officers' smokeroom on the boat deck. The bulkheading in the P.O.'s and crew's accommodation is constructed of Formica-clad Marinite incombustible panels. There are four Birmabright lifeboats, one of which is powered. The propelling machinery consists of a single-screw arrangement of Harland and Wolff-Pametrada double-reduction geared turbines having a total power in service of 7,500 s.h.p. Steam at 500lb. per sq. in. and 800 deg. F. is generated in two Harland-Babcock and Wilcox integral-furnace watertube boilers, fitted with superheaters and air heaters. The auxiliaries are in general electrically driven, but the standby feed pump, the tank cleaning booster pump and the main cargo pumps are all independently driven by small steam turbines. The electrical installation is subdivided as follows: power circuits, including ventilation and cooking gear, 440 volts A.C., lighting circuits 115 volts A.C. Electric and energy is supplied normally by two 550 kW. 450 volt, 3-phase, 60-cycle, 8 power factor, Peter Brotherhood-Harland and Wolff turbo alternators. As an alternative means of supply for use in port, a 200 kW. 450 volt 3-phase, 60-cycle, National-Campbell and Isherwood Diesel alternator is fitted.—*The Marine Engineer and Naval Architect, Vol. 76, August 1953; p. 350.*

Performance of Free-piston Generators

The free-piston gas generator turbine power combines the high thermal efficiency of the Diesel-engine combustion cycle with the simplicity and flexibility of the turbine power take-off. It consists of a gas generator producing high-temperature and pressure gas which is used to drive a turbine. The gas generator has two opposing pistons in a single-power cylinder. These pistons compress their own air for scavenging and charging the power cylinder. They also store the energy required to halt their outward movement and return them for the next compression stroke. Combustion proceeds on the two-stroke Diesel cycle and power is produced by the exhaust gas which drives the turbine. The crankshaft, connecting rods, and bearings of the Diesel engine are eliminated. In comparison to a gas-turbine power plant, the free-piston gas generator supplants the compressor, compressor turbine, combustion chamber, and heat exchanger. Ducting is much less. The use of a free-piston gas generator-turbine combination for producing power has long passed the theoretical stage. Test data are available on two designs of free-piston units which have been developed to an operating condition and which provide factual information for their evaluation. A third unit is now in the test phase and information will be available at a later date. The paper notes that the free-piston gas generator-turbine power plant has already shown its commercial suitability and acceptability by the variety of installations made with the S.I.G.M.A. units in France. Developments in this country have shown that several concepts of the same principle of operation can be perfected. Operational data have shown that the thermal efficiency of modern Diesel engines can be surpassed with a power plant of no greater bulk or weight, fewer working parts, and reduced cost and time of construction. The free-piston engine has shown that it has many peculiar advantages, such as absence of mechanical vibration. It provides an independent high-speed drive well-suited to various types of applications. The free-piston unit as presently available in this country is the result of comparatively little development. A total of only seven gas generators has been constructed. Two models now appear ready for commercial exploitation. These models have not reached the stage of perfection which will be possible with continued development, but they are commercially competitive. One line of development certain to be continued, especially for uses where low specific weight and mass are required, is supercharging of the power cylinder. Outputs approaching twice the present output with little increase in severity of operation will be obtained. The weight of the power plant will be decreased accordingly. Another line of development is after-burning. This possibility is attractive because the exhaust gases from the gas generator contain approximately 75 per cent of the original oxygen, and by after-burning the power output can be increased about 30 per cent with only an approximate 10 per cent increase in specific fuel consumption. When one considers that this increase in power is obtained by the addition of a comparatively simple combustion chamber of low weight and yet maintaining the same operating conditions in the gas generator, the possibilities are apparent. A parallel line of development will be perfection of heavy-duty units for installation where low initial cost, easy maintenance, economy of operation, and long life and reliability are the more important factors. Low exhaust pressures and temperatures will reduce cost and increase life of the turbine. The power plants will still have a specific weight and mass superior to that of comparable Diesel-power plants.—*Paper by J. J. McMullen and W. G. Payne, read at the 1953 A.S.M.E. Spring Meeting. Paper No. 53-S-18.*



Cross section of Sigma model GS-34 free-piston gas generator

1.—Engine case. 2.—Compressor cylinder. 3.—Dished ends. 4.—Motor cylinder. 5.—Scavenge air receiver. 6.—Suction valve. 7.—Delivery valve. 8.—Balance pipe. 9.—Coolant inlet. 10.—Coolant outlet. 11.—Gland. 12.—Starter, 13.—Stabilizer. 14.—Synchronizing rod.

New 33,000-ton d.w. Tanker

The first of two 33,000-ton vessels to be built at the Walker Naval Yard of Vickers-Armstrongs Ltd., for the World Tankers Corporation, of Monrovia, Liberia, has successfully completed her trials. The *World Enterprise* is the largest tanker so far launched in the United Kingdom, and has been built to the highest class of Lloyd's Register, the American Bureau of Shipping, and the Ministry of Transport, and complies with all the latest requirements for the carriage of petroleum in bulk. She has been chartered by the Anglo-Saxon Petroleum Co., Ltd., and makes her maiden voyage to Mena al Ahmadi to load crude oil for the U.K. or the Continent. The principal dimensions and particulars of the *World Enterprise* are as follows:—

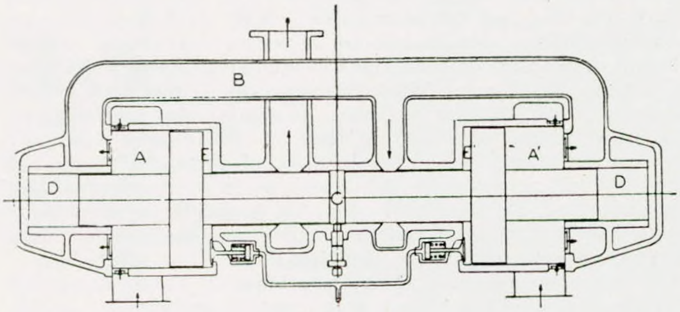
Length overall	663ft. 0in.
Length b.p.	635ft. 0in.
Breadth moulded	86ft. 0in.
Depth moulded	45ft. 9in.
Loaded draught	34ft. 5in.
Deadweight	33,000 tons
Service speed	16 knots

The *World Enterprise* is powered by geared turbines of Pametrada design, manufactured by the Parsons Marine Steam Turbine Co., Ltd. The turbines were designed to develop a normal service power of 12,500 s.h.p. at 100 r.p.m., and they are capable of developing a maximum of 13,750 s.h.p. at

103 r.p.m. continuously, if required. The astern turbine will develop about 45 per cent of the ahead service power at 50 per cent of the service revolutions per minute. The machinery has been constructed to the requirements of Lloyd's Register of Shipping and the American Bureau of Shipping. The set consists of an h.p. and l.p. turbine working in series, operating with superheated steam at the exceptionally high pressure of 850lb. per sq. in. and a total temperature of 850 deg. F. The h.p. turbine is of the all-impulse type and comprises 15 single row stages, the solid rotor forging being gashed to form the discs to which the blading is attached. The l.p. turbine is of the single flow type, comprising 21 rows of reaction blading which is of the segmental type throughout the cylinder, and in 17 rows in the rotor, the last four rows being of the integral type. The astern turbine is situated within the exhaust end of the l.p. ahead turbine and consists of two 3-row impulse wheels which are forged solid with the l.p. ahead rotor. The astern casing is separate from the main casing and supported in such a way that no distortion can be transferred to the main casing. The impulse and reaction blading throughout the turbines is of stainless iron. The h.p. turbine casing and rotor, and the astern casing, are made of 0·5 per cent molybdenum steel. Double reduction articulated type gearing is employed, the primary gears of the h.p. and l.p. turbines each being enclosed in a fabricated steel case; the main gearing is similarly enclosed in a fabricated case to which the primary gear cases are secured. Each turbine drives through a flexible coupling to a pinion which engages with its own primary wheel. The secondary pinions which engage with the main gear wheel are driven through flexible couplings and quill shafts from the primary wheels. A single collar Michell thrust block is fitted immediately aft of the gear case. A Siemens-Ford type torsionmeter, with the indicator conveniently located, is installed for measuring shaft horse power. Two Foster Wheeler "D" type watertube marine boilers supply steam at a pressure of 850lb. per sq. in. and a temperature of 850 deg. F. at the superheater outlet. The boilers complete with superheaters, desuperheaters, economizers and uptakes, were constructed by Richardsons, Westgarth and Co., Ltd., and operate in conjunction with bled steam Weldex air heaters, made by the Wellington Tube Works. Howden forced draught fans supply air to the boilers.—*The Shipping World, Vol. 129, 8th July 1953; pp. 36-37.*

High-output Free-piston Gas Generators

The development of free-piston gas generators suitable for naval propulsion purposes was started in 1943 at the Baldwin-Lima-Hamilton Corporation, then the General Machinery Corporation, under a contract with the U.S. Navy Department. For certain naval propulsion purposes, the free-piston gas generator-gas turbine or "gasifier-turbine" power plant has potential advantages. The developments at Baldwin-Lima-Hamilton have been concerned exclusively with gasifiers of high specific output, suitable for naval propulsion, or other applications which require lightweight machines. The weight and volume of a gasifier-turbine plant of a given total output power is largely fixed by the weight and volume of the gasifier part



Gasifier with outward compression, outer bounce cylinder

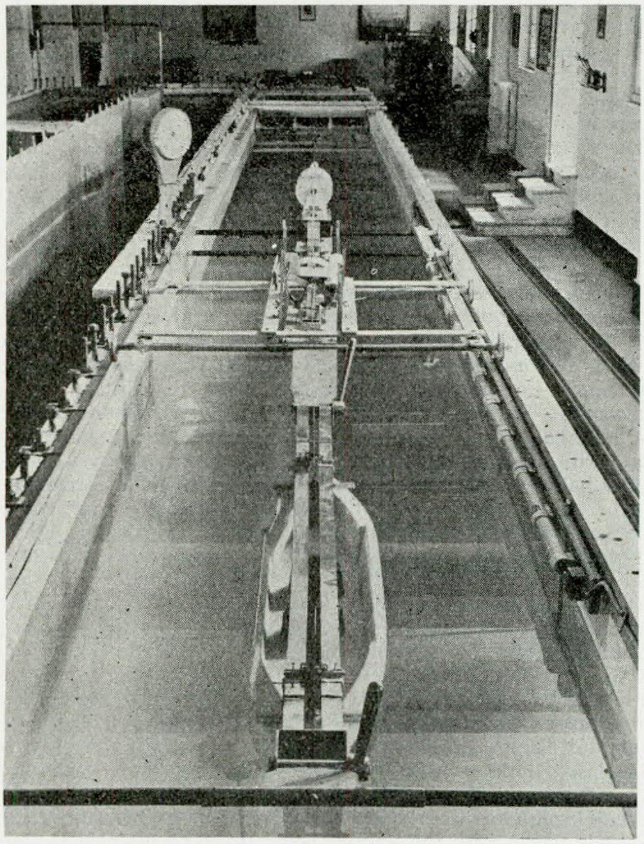
of the plant. This is influenced by the following factors: The size and number of the gasifier units used for a given total power; the arrangement of the parts of the gasifier; the pressure and temperature of the cycle; the piston speed of the gasifier; and the design as regards material used, thickness of stressed parts, and other similar factors. Of the gasifiers constructed by Baldwin-Lima-Hamilton, the Model A is of 7-in. bore power cylinder, 21-in. bore compressor cylinder × 10-in. full power stroke and the Model B of 8¼-in. bore power cylinder × 23-in. bore compressor cylinder × 11-in. full power stroke. All arrangements utilize a pair of opposed pistons connected by some type of synchronizing mechanism to ensure their operation in exactly opposite phase. This opposed-piston arrangement gives a balance of inertia forces, so that any vibratory reactions between the gasifier and its foundations are limited to forces produced by the pulsating flow of the discharge and scavenge gases.—*Paper by R. A. Lasley and F. M. Lewis, read at the 1953 A.S.M.E. Meeting. Paper No. 53-S-34.*

A Modern Tanker Using Alternating Current

The use of alternating current on ships is still a controversial subject but its adoption has increased considerably in the last two or three years, and building programmes now in hand for the major tanker companies will show an even greater rate of growth in this form of electricity supply. The major oil companies, through their tanker-operating associates, have in hand an extended programme of medium and large tankers in which alternating current is used, probably more completely than in any other existing British tonnage. One of the early ships in this programme is the 18,000-ton *San Florentino,* built by Cammell Laird and Co., Ltd., for the Eagle Oil and Shipping Co. Ltd., and the major part of the electrical equipment has been supplied by The General Electric Co., Ltd. The main propulsion machinery in the *San Florentino* consists of double reduction geared turbines manufactured by the shipbuilders and supplied with steam at 485lb. per sq. in. superheated to 790 deg. F. and generated by John Thompson "La Mont" oil-fired boilers. Electric power is obtained from two turbo-alternators each comprising a Brotherhood multistage impulse turbine operating under the same steam conditions and driving, through 6,000/1,200 r.p.m. reduction gears, a G.E.C. alternator of 687 kva., 0·8 p.f., 440 volt, 3-phase, 60 cycles. The alternator is of the salient pole type and has a direct coupled main exciter and "V" belt-driven pilot exciter. Also, for emergency and port use, there is a G.E.C. 250 kva., 600 r.p.m. alternator driven by a British Polar six-cylinder Diesel engine; this machine also has main and pilot exciters. The pilot exciters generate at a constant voltage of 115 volts, and those on the turbo-alternators also supply current for the speed-control motor on the turbine and for the circuit breaker closing and tripping coils. For this reason they are rated at 4 kW., and are over-compounded so that the peak current of the circuit breaker solenoid does not affect the voltage of the alternator. Each pilot exciter supplies only the circuit breaker for its own set and there are no cross-connexions. The alternators are drip-proof self-ventilated with terminal boxes on the upper half conveniently disposed for overhead cable connexions to the main switchboard. The normal sea load is carried on one large alternator and the port load mainly on the Diesel set which has an output sufficient to start from cold ship, including running the forced draught fans, boiler fuel and circulating pumps, as well as other essential auxiliaries. All three alternators can be synchronized and paralleled, and although it will be normal practice to parallel the steam sets to the Diesel set before shutting down the latter, it will not normally be necessary to carry out the reverse procedure. For this reason the main alternators have remote-control speeder gear and electrically-operated circuit breakers, whereas, for the Diesel set, both speed and circuit breaker are hand-controlled. All alternators have Newton-Derby carbon pile automatic voltage regulators in the field of the main exciter and these, in conjunction with the appropriately designed alternators, aim at minimum voltage dip and rapid voltage recovery on peak loads, especially in respect of the starting currents of the larger motors when switched direct on the line. The engine room motors are all 3-phase squirrel cage induction motors ranging from the 4½ h.p. extraction pump to the 178 h.p. boiler feed pump, and are all single-speed machines except those driving the forced draught fans, transfer pump, fire and bilge pump and engine room ventilation fans which have double-wound stators for different speed ratios to suit each individual case. All the motors are started by switching direct to line except the 178 h.p. feed pump which has an air-cooled auto-transformer. An all-insulated 440 volt, 3-phase system has been adopted on this vessel, employing the spur system of distribution.—*C. P. Harrison, G.E.C. Journal, Vol. 20, July 1953; pp. 168-183.*

Closed Circuit Model Tank

A tank, with a closed circuit for experiments with ships' hulls and propellers, has been erected at the University of Genoa. This tank differs from the original insofar as the model is now static while the water is in motion, whereas formerly the model was moving and the water was static. In the illustration, the apparatus for measuring the model's resistance and the speed of the water is shown. This consists of a 5 cm. dia. Pitot tube, in which is placed a float that controls, by means of a lever, a stylus operating on a revolving drum. The resistance of the model is measured by a balance which also records on a revolving drum. This new tank possesses the following advantages over the older type: it costs less to construct and requires less time to carry out experiments. In addition to allowing the resistance of the model to different speeds to be measured, the new tank makes it possible to measure the pressure of the water on different types of hull, to follow the track of the threads used for observing the flow round the hull, to estimate the efficiency of the model, test propellers either on their own or fitted to the hull, and study generally the behaviour of solids in liquids. An air tunnel has been fitted to the tank and, by regulating the speed of the water with the speed of the wind, it is possible to test yachts fitted with sails. The tank is about 75ft. 6in. long and the channel section is about 4ft. by 4ft.—*The Shipping World, Vol. 128, 17th June 1953; p. 561.*

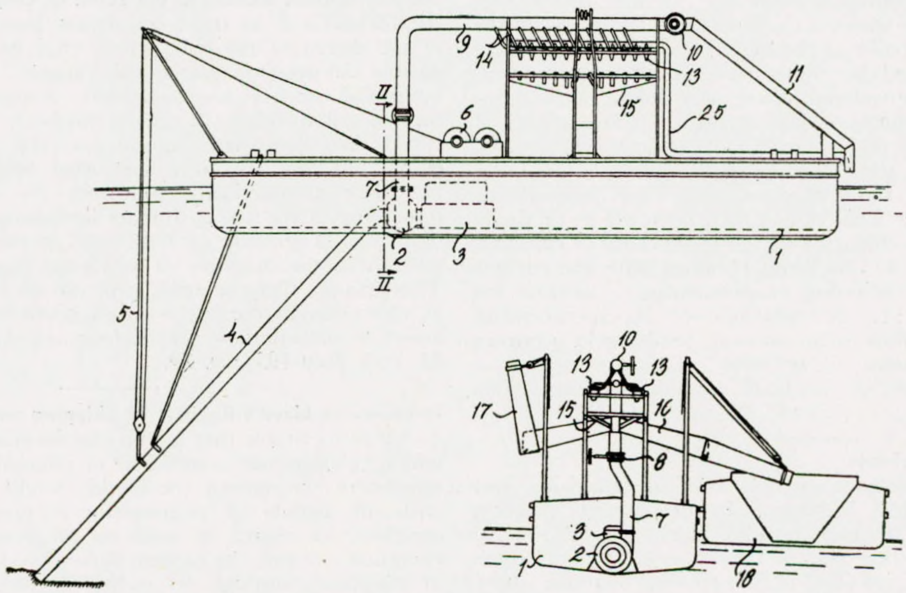


Vanadium Ash Problems

Vanadium bearing, residual fuels have been a continuous source of fireside problems in boilers at the U.S. Naval Experiment Station during the past decade. They have produced a variety of troublesome deposits in saturated tube nests, played havoc with refractory and caused serious wasting of superheater tubes. While these furnace problems resemble those of fleet boilers, sea water contamination played no part in them. Analyses show that while the major constituents in all of the furnace deposits are the same as the major constituents of the heavy fuel ash, their proportions vary markedly from one furnace area to another and rarely agree with the typical proportions of the ash. This variation in proportion of major constituents in different parts of the furnace suggests selective deposition related to furnace temperature gradients, combustion characteristics, or ash properties. Neither fuel impingement nor uniform ash deposition would explain these variations, since both should yield approximately the same ratio of vanadium to sodium from front to rear. It is improbable that a mechanism exists for concentrating sodium in some areas and vanadium in others once a uniform ash has been deposited. The selective deposition of vanadium compounds on the hot, rear tubes may mean that the vanadium is derived from organic complexes which burn while passing through the furnace and finally deposit as ash or molten droplets on the last tubes. On the other hand it could mean that low melting vanadium compounds, travelling as molten droplets in the gas stream, select the hottest tube surfaces because they wet more readily. Such wetting is most apt to occur when the tube skin temperature approaches the melting points of the compounds involved. While the rear row tubes of 800lb. per sq. in. generating nests probably are several hundred degrees below the melting points of ordinary vanadium compounds like V_2O_3, they could easily approach melting points of unusual vanadium compounds like the hydrotrisulfonates. The black slag which recurs in this area melts at about 1,000 deg. F. These hypotheses by no means exhaust the possible mechanisms. It is conceivable that selective deposition is based on ash classification due to relationships between particle size and gas velocities. The high concentrations of sulphate and acidity in the granular deposits show that sulphur oxides influence deposition. The general character of the deposits in the central, transition zone would appear to indicate that deposits can be altered considerably in composition and appearance after they have been laid down. The greater thickness of the black slags several tubes behind the fire row may mean that this is another selectivity point based on skin temperature, gas pattern or particle size, or simply that the fire row deposits are thinner because of continuous dripping under the intense heat. Even though powdery deposits have been found on forward tubes, there is no proof that they are a necessary primary stage of the other deposits. Steaming periods as short as eight hours have developed scattered meandering islands of fused slag on the forward tubes and uniform, thin, dark blue slags on the after tubes. Both the islands and the uniform coating contained about 25 per cent vanadium. Either fusion of powders or coalescence of molten droplets could have produced the islands. Whatever their source, this early occurrence of fused deposits on clean tubes shows that true slagging can start immediately on lighting off and that it need not involve an "induction" period. The phenomenon of precoat and secondary slagging continues to be a prominent and puzzling one. Long period operation generally yields deposits with ⅛ inch or more of vanadium tube-side slags, overlaid with heavier deposits of sodium sulphate slags. The reverse structure never has been found in these boilers. Variation in fuel composition does not explain this stratification because both the heavy fuel and the tube-side layers are consistently high in ratio of vanadium to sodium. The structure suggests that vanadium compounds deposit first because of lower melting points, better wetting properties, or other characteristics and then instigate deposition of other slags by mechanical trapping or thermal insulation effects. This does not explain why the proportion of sodium is higher in the secondary slags than in the fuel. It may be that vanadium compounds tend to separate from mixed deposits and collect on the tube surfaces after deposition has occurred.—*F. E. Clarke, Journal of the American Society of Naval Engineers, Vol. 65, May 1953; pp. 253-270.*

Cyclone Separator for Suction Dredger

This patent describes a suction dredger in which the delivery side of the pump is connected to a series of cyclones disposed in parallel. The cyclones may be installed on an auxiliary vessel alongside the main dredging vessel. The vessel shown in Figs. 2a and 2b is equipped with a centrifugal pump (2) operated by an engine (3). The suction pipe (4) is suspended in tackles (5) which are operated by winches (6). The delivery pipe (7) rises vertically, passing through a valve (8), and discharges into a horizontal distributing conduit (9) at a considerable height above the deck of the vessel and connected through a valve (10) with an overflow pipe (11). Slightly lower and to either side of the distributing conduit are arranged two series, each of twelve cyclones (13). Beneath these cyclones is placed a collecting tank (15) which receives the thickened suspension leaving the cyclones. From this tank the suspension can be transferred in the usual manner through tip chutes (16 or 17) into barges (18). The water separated off in the cyclones which has been freed of solid material is discharged via another pipe (25).—*Patentees: N.V. Aannemersbedrijf, Berg en Dal. (British Patent No. 692,668.) World Shipbuilding, Vol. 3, August 1953; p. 119.*



FIGS. 2a *and* 2b

Metallic Elements in Fuel Oils

Deposition of ash and corrosion, both major difficulties in the use of residual fuels in gas turbines, are due mainly to vanadium and sodium in the oil. Iron, sodium, vanadium and nickel are the chief metals present in an Iraq residue examined, and methods have been developed for the determination of these elements in oils. Iron and sodium compounds can be removed by filtration and water-washing respectively, but vanadium and nickel compounds cannot be removed by simple means, and are present as organic compounds dispersed in the oil. Adsorbent and solvent extraction have been found to concentrate the vanadium compounds, but by both methods the maximum for an Iraq oil was only fourteen times that of the original fuel oil, viz. 0·07 per cent vanadium. Both vanadium and nickel compounds could be concentrated in a fraction representing about 5 per cent of the Iraq crude used and in about 20 per cent of a Venezuelan crude, and they are combined with a significant part of the asphaltenes. Previous work suggesting that all the vanadium is present as porphyrins is criticized. No method was found of preparing fuel oils free from vanadium which would be cheaper than distillation.—*F. H. Garner et alii, Journal, The Institute of Petroleum, Vol. 39, May 1953; pp. 278-293.*

Research into Oil Burning

A new organization has been set up by White's Marine Engineering Co., Ltd., to deal with the many and varied problems that arise in connexion with the combustion of fuel oils. Over the past few years the rate of research into the combustion of fuel oils has increased steadily and it has become necessary to erect a test furnace designed to absorb 42×10^6 B.Th.U's per hour as well as a shell boiler capable of absorbing 19×10^6 B.Th.U's per hour. There is a large amount of data available on the spray characteristics of the various types of sprayer. The problem of design and development of air registers, however, is a big one and no large and comprehensive programme of research has ever been undertaken by any authority as far as is known. Consequently little data exists and air registers are normally designed by rule of thumb, the design being perfected after a considerable amount of trial and error. Due to this lack of knowledge, it was decided that the research facilities at Hebburn should be developed mostly on the air flow side of oil burner design and the testhouse was laid out accordingly. The test furnace is at present capable of taking burners up to 800lb. per hour capacity and is supplied with a pumping and heating set capable of delivering 1½ tons of oil at 450lb. per sq. in. and 300 deg. F. When extended and fitted with water cooling coils the furnace will be suitable for testing burners of one ton of oil per hour capacity. Pumping is done electrically and in two stages, first to 200lb. per sq. in., then to 450lb. per sq. in. Thermostatically controlled heating, either by steam or electricity, is employed. Combustion air is delivered to the furnace by a Davidson fan at a pressure of 20in. W.G., almost all of which is available for discharge through the register. Velocities of over 17,500ft. per min. can be obtained for mixing with the oil spray. With these facilities it is anticipated that pressure jet burners obtaining CO_2 figures of over 14 per cent from full to half load can be developed and information now available suggests that these anticipations will be realized. The boiler is of the natural draught Scotch marine type, and is used for the development of natural draught registers and for the rigorous testing of blast (i.e., air or steam atomizing) burners. Results obtained so far on natural draught are interesting, as with the latest development tests 13 per cent CO_2 has been obtained without difficulty with a draught loss through the register of only 0·25 W.G. The work on oil burners is now proceeding extensively and covers the following items: (a) High pressure oil burning equipment: A detailed study has been going on for some time into the aerodynamics affecting air register design in an endeavour to determine the governing characteristics for optimum combustion efficiency. (b) Wide range spill type burners: The register design in this case is extremely complex as, due to the wide variation in capacity, high efficiency over the working range is difficult to obtain. This work is running parallel with the high pressure equipment. (c) Remote controlled oil burners for lighting up P.F. flames in power station boilers: This type of burner is very specialized and is of the constant capacity type. White's have designed several control systems, the one now undergoing test being of the type where an operator on the floor of the station can flash up a boiler at the touch of a button and where the sprayer is scavenged by compressed air after shut down. A scheme is being considered where this takes place automatically if the P.F. flame is extinguished. Three forms of ignition equipment are under survey, viz.: the resistance coil, 10,000-volts spark and the "High Energy" ignitor. The latter is expected to be the most effective. (d) Low pressure air atomizing oil burners: This burner is controlled by a single lever and is thus highly efficient over its whole range of capacity. A turn down ratio of 4:1 is being obtained with the present burner and tests are continuing in an attempt to increase the ratio to as high as 6:1. A large amount of experimenting has been done on the flow of air and oil, resulting in a burner having a sensible constant oil/air ratio. This feature enables the burner to be handled efficiently by unskilled firemen.—*The Shipping World, Vol. 129, August 1953; pp. 147-148.*

Launching Gear for Lifeboats

This invention relates to a new type of launching and lowering gear for ships' lifeboats. In the normal position with the boat stowed, the two threaded blocks (21) in Fig. 10 are close together on either side of the stand (15). To launch the boat the threaded drive shaft (13) is allowed to rotate under the control of a brake or by operators at one or both of the handles (20). The resulting movement of the blocks (21) gradually pays out the fall ropes (26) as the distance between the blocks and the sheaves (25) on the stands (10a and 11a) is decreased. The boat may be hoisted by the reverse process, turning the shaft (13) by means of the handles (20) so as to move the blocks (21) back again. As this will be a slow process, it may be preferable to take a rope or wire (29) from a convenient winch and attach it to the bight of the fall ropes (28) which is temporarily disconnected from the centre stand. The winch is then operated so as to wind the fall ropes on to its drum.—*Patentees: Marepa Trust, Ltd., Whitstable (British Patent No. 691, 676.) World Shipbuilding, Vol. 3, August 1953; p. 121.*

Anti-vibration Nut

The temporary fastening of machinery parts by means of bolts and nuts presents certain problems to engineers, particularly when vibration is likely to be present. Many ingenious devices have been tried such as special nuts and washers, locking plates and so on, but improvements to ensure bolt tightness are still being studied. A recent fastening which appears to solve the problem, is the Lester anti-vibration nut, manufactured by the Lester Lock Nut and Washer Company. This nut consists of an element of conical discs firmly assembled in a hexagon body, and then drilled and tapped. It is free-running when first screwed on to the bolt, but on tightening the nut against the work, the stack of coned discs flattens until the threads cut in them contact at great pressure both sides of the thread of the bolt, which thus become fully supported against deformation and thread-fatigue. Radial, bending and tangential stresses are uniformly distributed and vibration, transmitted through the elastic nut body and periphery of the coned disc laminated element, is very considerably reduced before reaching the fully supported bolt threads. Vibration thereafter automatically continues the effect of the initial tightening of the nut by further increasing the elements' wedge-like inwards pressure on both sides of the bolt threads in proportion to the intensity of vibration present in the assembly. This ensures that the nut's grip on the threads of the bolt at all times exceeds the unscrewing tendency of the nut when subjected to vibration.—*Shipbuilding and Shipping Record, Vol. 82, 16th July 1953; p. 79.*

Influence of Lloyd's Register of Shipping on Welding Problems

It is inevitable that a ship classification society whose rules and regulations set a standard of strength for ships and their machinery throughout the world should be closely associated with all aspects of engineering progress and development, especially in regard to methods of construction, design and operation. From the earliest days of welding Lloyd's Register of Shipping, through its technical officers, played an active part by conducting tests to establish the strength properties of welds, and by regulating the application of welding in ship construction. A feature of the Society's Rules for Ship Construction which has had considerable influence on welding development is the control of the quality of electrodes. Electrodes are required to be of an approved type, and approval depends not only upon compliance with a series of preliminary

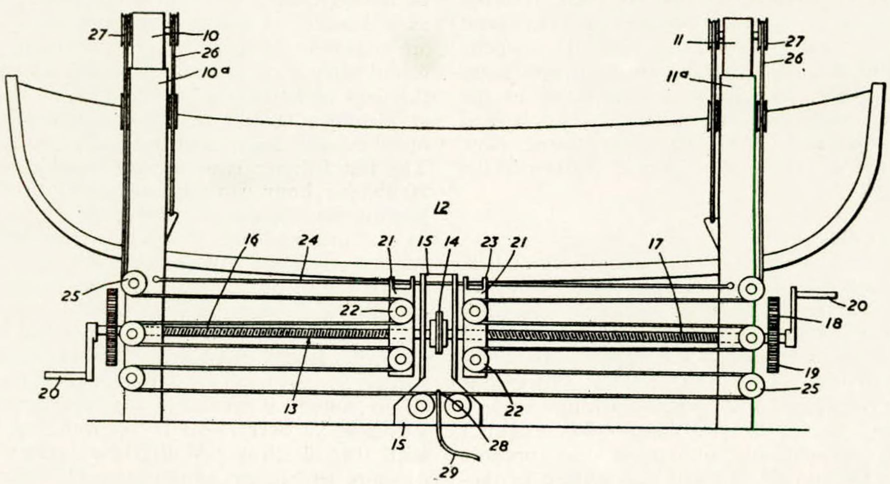


FIG. 10

tests to prove the quality of the weld deposit but also upon the quality of the weld deposit and also upon periodical inspection by Lloyd's Register surveyors of the establishments where the electrodes are manufactured. Further efforts to improve the standard of shipyard welding include the encouragement which Lloyd's Register has given to the use of radiography as a means of quality control and in the training and testing of welders. In regard to the welding of boilers and other pressure vessels, Lloyd's Register of Shipping was the first authority in Europe to issue regulations governing standards of design and construction. These first rules were primarily intended to apply to pressure vessels for land plant. In consequence, numerous welded pressure vessels, including water tube boiler drums, steam receivers, condensers, evaporators and other vessels forming part of chemical, oil-refining and refrigerating plants, were constructed under the Society's inspection, and on the basis of their experience the Rules for Welded Pressure Vessels were extended to cover marine installations in 1938. The welding of Class I pressure vessels is a specialized branch of the welding industry and involves 100 per cent radiographic examination of butt welded joints. So far as Lloyd's Register of Shipping is concerned, this class of work is accepted only provided the vessels are made by approved firms who are properly equipped for this type of construction and who have demonstrated that their welding work is of the highest standard, both as regards quality and consistency.—*H. N. Pemberton, The Welder, Vol. 22, June 1953; pp. 38-39.*

Crane for Unloading Tankers

This patent relates to a crane for use in loading and unloading tankers. The invention is intended to give a means whereby a group of pipes can be connected between the shore pipe lines and the manifold on board a tanker in such a manner that tankers of varying sizes and at varying conditions of loading can be automatically accommodated at all states of the tide, sea and weather. In Fig. 7 the jib (10) is formed from six 8-in. diameter mild steel oil pipes (11) braced together by means of strut members (12) to form a rigid structure. The jib is secured to a pair of outer rings (13) which are free to rotate about inner rings (14), roller bearings (15) being provided between the two sets of rings. The jib can therefore rotate about a horizontal axis (16). The inner rings (14) are secured to supports (17) which rest on a cradle (18). The cradle is mounted on roller bearings (19) on a table (20) so as to be free to rotate about a vertical axis (21). The table has four braced leg members (22) which rest on the jetty (23). The jib has a rearward extension (24) supporting a counterweight (25) which ensures that the jib head (26) tends to rise even when all the oilpipes are full. The length of the jib from the luffing axis to the jib head is, for example, 50ft., which is sufficient for tankers of all sizes to be accommodated under all conditions of mooring, loading, etc. Oil connexions are maintained through swivelling joints (33, 37), arranged in the horizontal and vertical axes of rotation.—*Patentees: Anglo-Iranian Oil Co., Ltd., and G. S. Barrass, London. (British Patent No. 692,892.) World Shipbuilding, Vol. 3, August 1953; p. 121.*

Carbon Paste as Protection During Welding

On many welding jobs some sections of the part need to be protected against the heat of the blowpipe. On other jobs, sections must be rebuilt to specific contours or dimensions. Fire-resistant carbon products are virtually the only materials that are completely satisfactory for these purposes. Carbon does not shrink or crack when heated, and does not stick to metal. That is why carbon is so widely used as a welding aid. Many forms of carbon are available. Carbon paste possesses all of the moulding qualities of plaster of Paris, clay or other moulding substances with the added fire-resistant qualities of carbon. In solid form, carbon can be machined to practically any shape desired. Some uses for carbon paste in welding are to protect a threaded or irregularly shaped hole near the break to be welded and to prevent distortion of a part due to excess heat. Solid carbon can be threaded to fit a threaded hole which has broken. Such a carbon bolt will act as a mould for rebuilding the threaded hole. Carbon in bar shape forms a mould for rebuilding a plain hole in the same way.—*E. E. Olson, The Welding Journal, Vol. 32, July 1953; p. 632.*

Automatic Welding Process

The Argonaut welding process uses a shielded arc with direct current (electrode positive) of relatively high amperage (50,000 amps. per sq. in. minimum) on a continuously-fed bare-wire electrode of small diameter. The high current gives a very fast deposition rate, and the manner of metal transfer (by projecting across the arc) permits welding in all positions. These advantages, naturally, give the process a wide field of application on all types of fabrication where aluminium of relatively heavy section is welded *in situ*. No flux is needed for the process, and the welds are of good quality, free from slag and inclusions, and also, of course, from post-weld corrosion problems. With multi-pass techniques, there is no practical thickness limit, and comparatively few passes are needed because of the large amount of filler metal deposited. Thin metal (down to $\frac{3}{16}$in. in thickness) may also be welded by using a smaller-diameter electrode wire and lower current. The efficiency of the Argonaut process of this type of work is comparable with that on heavy sections. The arc is self-adjusting, and the equipment includes a reel and feed motor

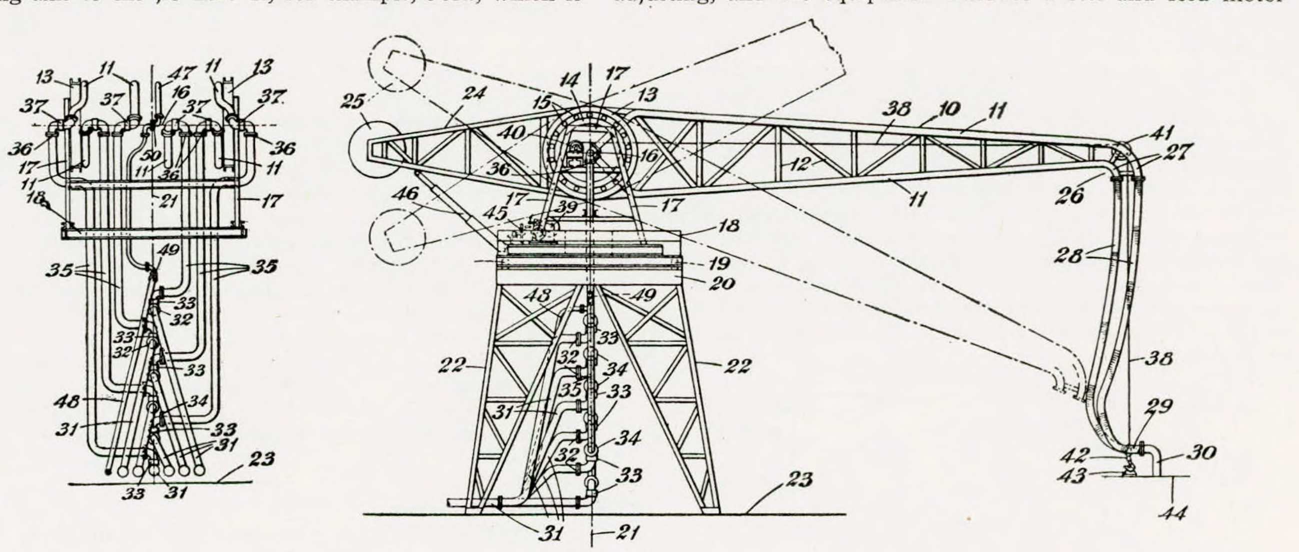


FIG. 7

to supply the filler wire to the torch or gun. The feed to the gun is via a special cable, which also conducts the argon gas to provide the air-excluding shield around the arc and over the weld pool. The automatic adjustment of the arc is governed by the burn-off rate of the wire, which is fed at a constant speed through the gun. In the event of the operator lengthening the arc by drawing the gun away from the work, the burn-off rate is reduced, the wire feeds to the work, and the arc is, therefore, shortened. If the gun is brought too close to the work, the process is reversed, and the operator therefore has a reasonable degree of latitude, without any adverse effect on the weld. To commence the welding operation, the electrode or filler wire is brought forward so that it projects about ½in. from the nozzle of the gun. With the depression of the trigger, the welding circuit is switched on and the argon starts to flow. The tip of the wire is then brought down to the face of the joint and the arc is struck. At this stage, a relay starts the wire-feed motor, the wire is brought forward, and the burn-off action continues. The fusion rate is very much higher than that with conventional arc welding, and, consequently, a higher weld speed is obtained. The Argonaut welding process is not confined to light alloys. It may also be used on stainless steel and copper-base alloys. Certain types of welds between dissimilar metals, such as copper to steel, can also be effected successfully by employing an aluminium-bronze wire as the filler material. The Argonaut process is very easy to use in the downhand position, for both fillet and butt welding. It is, however, in the vertical and overhead welding of fillets and butts that most advantages can be claimed for the Argonaut process, since operation in the more difficult positions is much easier than with the comparative methods for normal manual arc welding. In this connexion, the most notable improvement is that, when welding in the overhead position, the effect of gravity on the weld metal crossing the arc is hardly noticeable. This allows an operator of only average welding skill to carry out work in positions which would normally require a highly-skilled operator. The main difference in technique between the Argonaut process and the more conventional methods of welding is, principally, the necessity to move along the joint far more quickly. Once this technique has been mastered, the Argonaut process can be used in all positions, and, it is claimed, will produce satisfactory welds.—*The Shipbuilder and Marine Engine-Builder, Vol. 60, August 1953; pp. 496-497.*

Control of Distortion

A problem of distortion, which presents itself regularly, occurs when a heavy bar is welded round a cylindrical vessel to form a flange. It is usually found that distortion takes place in flanges which have to be used for making joints for covers on top of vessels. It is possible to avoid a great deal of this distortion by means of step and sequence welding, but this is not necessarily an economical method and the results indicated by the dotted lines on Fig. 9 show what happens in the majority of cases. The method of correction is simple. Where the flanges are rolled into shape, stresses are set up in the material itself which are partly released by welding, and therefore it is as a rule impossible to predict the reaction of the flanges to the welding operation. Application of heat as indicated in the sketch will do all that is necessary to bring the flanges straight and flat, if care is taken to see that it is applied quickly and that the full thickness is not heated to an appreciable extent, since the skin shrinkage creates the stress or force which flattens the flange.—*Transactions of the Institute of Welding, Vol. 15, December 1952; pp. 151-160.*

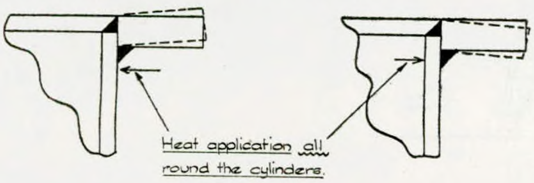


FIG. 9.

Very High Frequency Radio

Very high frequency radio, which has a range limited to visibility distance, is inherently a form of localized communication and the establishment and growth of such networks falls naturally into the working of a well run port. Existing medium frequency radio telephone services are unsuitable to port working and port control and in most ports their use is forbidden, as also is the use of morse on 500 kc/s. Consequently, until the development and exploitation of V.H.F. communications in a port, the operations of berthing, docking, tugging, radar control and administration of traffic was conducted by cumbersome visual signalling methods or by hand messages. After World War II, when V.H.F. had successfully been used for convoy communications on the limited scale permitted by the rules of radio silence, a radio convention was held at Atlantic City in 1947 to decide upon the international use of radio communication as a whole and, for the first time, a substantial band of V.H.F. frequencies was allocated for the world wide use of maritime mobile services. At first, adoption of this new medium was slow, although it was in Great Britain that the first V.H.F. licence in the world was granted to a marine user. This user was, and still is, France Fenwick Tyne and Wear Ltd., owners of the Anchor Line tugs, 26 of which are equipped with V.H.F. radio telephones for communication to their two dock offices, whence instructions and orders are passed to the tugs.—*R. I. T. Falkner, Cargo Handling, Vol. 1, No. 1, 1953; pp. 3-7.*

Refrigeration Equipment on Board Ships

The design of refrigerating installations for ships usually reflects a compromise between safety and reliability on the one hand, and space economy on the other. A completely enclosed room for the refrigerating machinery is desirable and sometimes essential; only for Freon installations does this not always apply. In this machinery room there are, *inter alia,* the compressors, condensers, main valves and possibly also the cooling-water pumps. It is also advisable to have a complete emergency unit, which may consist of a compressor, an electric motor, and a condenser. A spare cooling-water pump is not always necessary, since this can be temporarily replaced by the deck-washing or fire pumps. When brine is used as the cooling medium, brine coolers and pumps are usually also installed in the machinery room. With ammonia compressors, it is desirable that a water-sprinkler system should also be provided. The gravity system of cooling cargo holds, in which the cooling coils are mounted against the walls and ceiling of the hold, has now been abandoned because of the danger of mechanical damage to the coils by the cargo during loading or at sea. Instead, forced-air cooling is used. The cooling element lies flat against one of the walls and has one or two ventilators to provide the air flow. It is often specified that this cooling element shall consist of two independent sections. Ventilator-speed control is advantageous, and it is desirable to have access to the ventilators without entering the hold. Sometimes a separate ventilator room is provided. As regards the cooling agent, there are not rigid rules applying in all cases, but, in general, ammonia is preferred for the larger installations. The construction of an ammonia system is simpler than that for Freon, so that the initial cost is lower. Also, Freon is about five times as expensive as ammonia and twice as heavy. For smaller installations, Freon is often preferred. Brine cooling is most suitable where there are many holds to be cooled and long pipes are necessary. The disadvantages of brine cooling as compared with cooling by direct evaporation are the high cost of installation, larger motor capacity, greater space requirements, and heavier construction.—*Dutch Shipbuilding, Vol. 2, March 1953; p. 16. Journal, The British Shipbuilding Research Association, Vol. 8, July 1953; Abstract No. 7765.*

Voltage Regulators

Probably the most important type of automatic regulator found in the marine industry is the voltage regulator used for auxiliary power systems. Every ship of any consequence has an auxiliary power system for ship's service. The generating plant for these systems may be as much as 12,000 kW. for one of the Navy's large aircraft carriers. The voltage must be regulated within relatively close limits. An example of modern voltage-regulating methods in the marine industry is the type WRN-11 voltage regulator. This regulator has found wide use in the U.S. Navy for ship's service generators, varying in size from 60 to 1,500 kW. It is of the rotary-amplifier type and consists of (1) a Rototrol exciter directly connected to the generator and (2) a static electrical measuring circuit. It is shown diagrammatically in Fig. 1. The output voltage of the generator is fed through a static measuring circuit whose output is a D.C. potential that varies in polarity, according to a high or low condition of the generator voltage, to energize the control field of the Rototrol exciter. The terminal voltage of the exciter is regulated by this low-energy controlled field to effect the required change in the generator excitation and consequently the generator output voltage. Physically, the equipment involved in the WRN-11 regulator consists primarily of an automatic-control unit and a so-called potential unit, together with a manual-control unit (not shown in the figure), a voltage-adjusting rheostat, and a current transformer. The potential unit is energized by the A.C. generator voltage and current. Its output is a single-phase A.C. voltage that is used to energize the automatic-control unit through the voltage-adjusting rheostat. This automatic-control unit is a voltage-sensitive device, the output of which is a D.C. voltage; this output is then impressed upon the Rototrol exciter control field. When the generator output voltage is at the desired value, the output voltage of the automatic-control unit is zero. If the generator A.C. output voltage increases above the regulated value, the D.C. output voltage of the regulator will be in the direction to decrease excitation voltage through the Rototrol exciter. When generator voltage falls below regulated value, the opposite is true. The voltage-sensitive device consists of two parallel circuits, one containing a capacitor and the other a saturating reactor. The volt-ampere characteristics of these two elements are shown in Fig. 2. The point where the two curves intersect is the balance point of the two impedances. The operation of the voltage regulator depends on the fact that, if the voltage decreases below the balance point, the capacitor current is greater. When any unbalance occurs, a current flows in the Rototrol control field. Since the voltage-

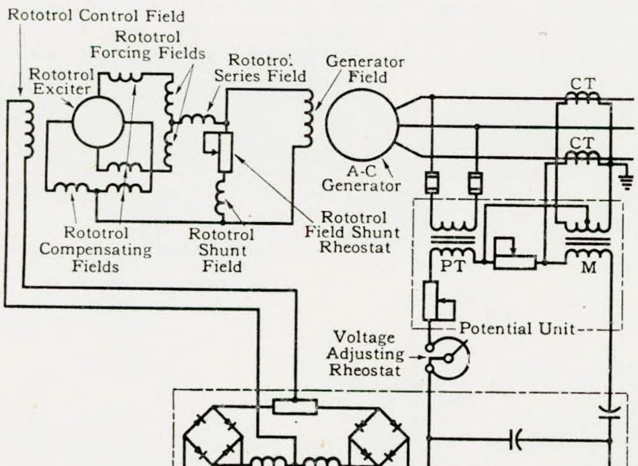


FIG. 1—*Diagram of the WRN-11 voltage regulator, widely used for U.S. Navy ship's service generators of many different sizes*

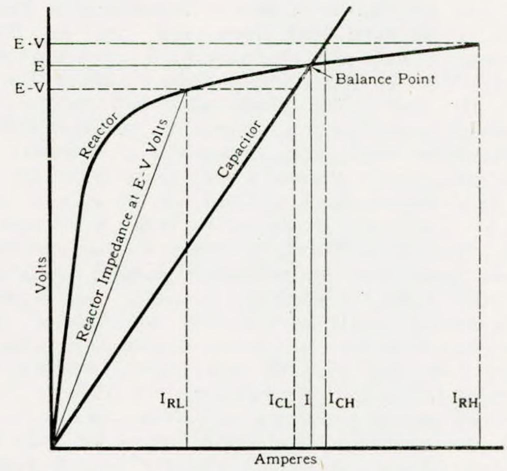


FIG. 2—*Volt-ampere characteristics of the capacitor and reactor circuits for the WRN-11's voltage-sensitive device*

I_{RL} = reactor current with low voltage
I_{RH} = reactor current with high voltage
I_{CL} = capacitor current with low voltage
I_{CH} = capacitor current with high voltage
I = reactor or capacitor current at normal or E A-C line volts

sensitive circuit consists of elements responsive to frequency, some means of frequency compensation must be provided; this is introduced in the WRN-11 regulator by a circuit between the generator and the voltage-sensitive circuit. The effect of an increase in frequency when the voltage-sensitive circuit is operating at the balance point is to increase the current in the capacitor branch and decrease that in the reactor branch. This causes control current to flow in the "raise" direction, which is the same effect that would be obtained by a decrease in voltage. In order to compensate for this effect, the voltage across the voltage-sensitive circuit must be increased as the frequency increases, or, conversely, it must be decreased as the frequency is decreased below normal. This response is obtained by means of a series condenser of the proper value such that it, together with the reactance of the circuit, will accurately compensate for frequency variation. The automatic-control unit used in the regulator responds to a single-phase voltage. Since the three phases of the A.C. generator load are not always balanced, the voltage drop in the windings may be different. Correction for this condition is made by the positive-sequence filter circuit of the potential unit. This filter circuit involves the two current transformers (CT), which energize a mutual reactor (M) and resistor in series with the secondary of the potential transformer (PT). The voltage output of the unit is proportional, as a result of the filter, to a balanced three-phase voltage (the positive-sequence voltage) of the machine. The performance of the WRN-11 regulator is approximately ±1 per cent at rated frequency and at unity power factor. Its frequency compensation holds voltage within ±1·5 per cent for a ±5 per cent frequency variation.—*S. A. Haverstick, Westinghouse Engineer, Vol. 13, July 1953; pp. 130-134.*

Radiation Suppressing Coatings

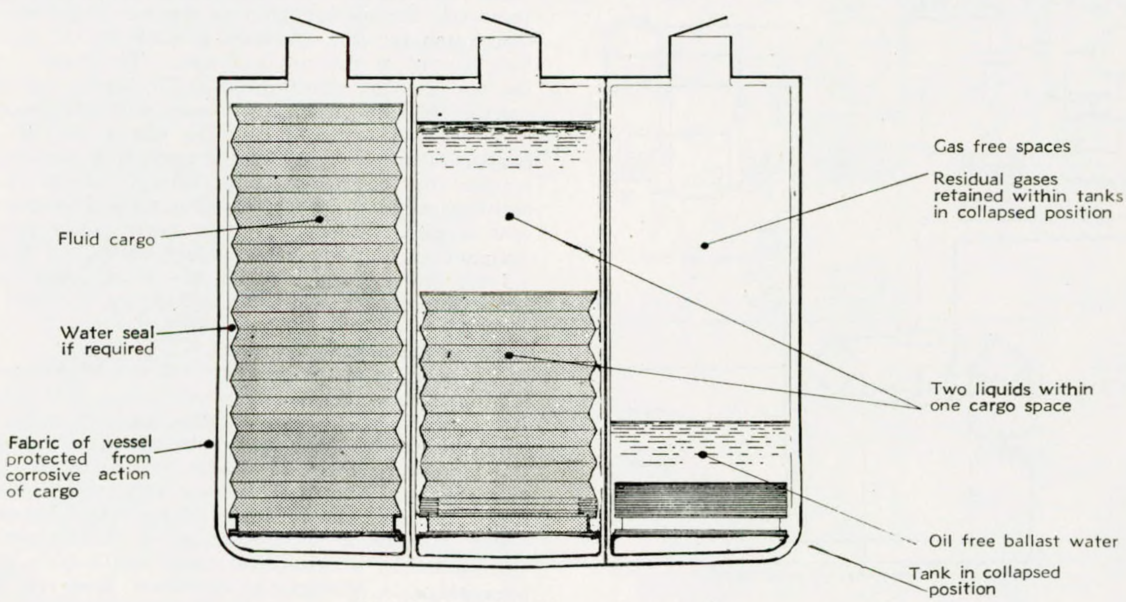
The object of the present work was to develop a refractory coating of low thermal emissivity which could be applied to alloys used in gas turbine construction and which would withstand the severe operating conditions involving thermal shock and vibration which are encountered in certain parts of such mechanisms. The most obvious application of such coatings is for the internal surface of flame tubes in which a flame, which may be at a temperature of 1,800-2,000 deg. C., is surrounded by a cylindrical metal wall which, for reasons of strength or of resistance to oxidation, must not be heated to a temperature much in excess of 800-900 deg. C. For highly luminous flames, such as result from the combustion of marginal fuels, the transfer of heat by radiation is a significant proportion of the total heat transferred from the flame to the surrounding wall and the maximum advantage accrues from a reduction of the thermal absorptivity of the inner surface of the wall. The results described show that the coatings which have been formulated are capable of reflecting a high proportion of incident radiation and moreover, when correctly applied, can withstand a very considerable amount of vibration and thermal shock without loss of adhesion to the basis metal. They can be applied to nimonic 75 and, with the use of a protective enamel undercoat, also to stainless steel and to mild steel. They are potentially suitable to application to any metal parts of engines, furnaces, etc., which are undesirably heated largely by radiation, and in such applications their use will result in a substantial reduction in the metal temperature compared with the temperature attained by an oxidized metal in the same application. The data which have been presented on the emissivity of polished metals, however, show that the latter are excellent reflectors of radiation as long as they remain unoxidized and the use of refractory coatings is not to be particularly recommended for application to metals which remain unoxidized at service temperatures. The main application of refractory coatings of the type described will be for parts which normally operate in the range of temperature 600-900 deg. C. The coatings are not to be recommended for temperatures higher than the firing temperature of the coatings due to physical changes which occur at these temperatures and which may result in an increase in thermal emissivity and a loss of adhesion of the coating to the basis metal. It is not considered that these coatings will afford any protection to the metal against attack by volatile constituents of fuel such as vanadium or lead compounds. Indeed, it is possible that the adhesion of the coatings may be adversely affected by such attack. There is, however, not yet sufficient data from practical trials with the coatings to enable their behaviour with fuels of this type fully to be assessed.—*Fulmer Research Institute, Special Report No. 1, 1953.*

Collapsible Tanks for Fluid Cargo

A projected scheme for a possible means of transporting bulk liquids while also providing facilities for carrying dry cargo without having to remove the containers for the fluid, was on view recently at the International Oil Bunkering Exhibition held by the Anglo-Iranian Oil Co., Ltd., in London. The idea originates from the firm of R. and H. Green and Silley Weir, Ltd., who hold patents on the scheme. The tanks in question are square in section and so constructed that only upward and downward movement is possible. The flexible inner containers are like a concertina and are encompassed by square tubular section frames, each connected to its immediate neighbour by equal sets of hinged links offset by stops to prevent them from opening too far and so failing to close again. These hinged links are made with a simple bearing surface at the pivot in order to prevent side wracking. Strong fabric, suitably impregnated with synthetic rubber, plastic or pure rubber, and impervious to the fluids which may be carried, is used for the flexible containers. When these containers are filled with fluid, all air is driven out so that there is no room for swashing at the top of the tank and all movement of the fluid within is thus prevented. The greater the head applied to the fluid in the tank, the more rigid the tank becomes, thereby increasing its self-supporting characteristics. The weight of the dry cargo carried on top of the tank when it is collapsed is transferred through the top cover to a supporting frame built within the tank, thus preventing damage to the container. The partial side frames situated between the main square frames are, on closing, forced inwards by means of the hinged links, and automatically control the folding of the container, as fluid is discharged. Delivery, suction and air vent connexions are all brought to the bottom channel frame. Each tank is a separate but standard unit and is secured at the base by quick release clips so that a tank in need of repair or overhaul can be quickly replaced without delaying the ship. Aluminium alloy will be used almost entirely in the construction of the tank bottom and plates, frames, hinges and all metal parts, which will be standard and interchangeable. Cleaning the containers is simple, and the work may be undertaken on the spot, or exchanged for a set of clean tanks, thus avoiding delay in sailing. While the tank is collapsed, a tank washing machine can be passed through the manhole in the centre of the top cover and, as cleaning progresses, the tank top is hoisted by a tackle round the machine and hose, while the drainings are drawn off at the bottom. Alternatively, if hand cleaning is desired, one man only with equipment is needed.—*The Shipping World, Vol. 129, 22nd July 1953; p. 78.*

Cathodic Protection in Sea Water with Graphite Anodes

This paper discusses in detail the experience gained with the use of graphite anodes at the Naval Research Establishment, Dartmouth, N.S., of the Royal Canadian Navy. The trials discussed show that active ships can be cathodically protected by impressed current systems using graphite anodes provided a current shield is placed around the anode to ensure a fairly uniform current distribution out from the anode.



Sketch of collapsible tanks in tanker

The current shield must be a high duty insulating material, such as a polymet material, must be much thicker than the usual thickness obtained with paint films and it must be bonded more firmly to the underlying steel than a paint film. Vulcanized rubber has proved satisfactory but is costly. Some success was obtained with flame sprayed Polythene but the coating applied was not thick enough to give this material a fair trial. The results and experience obtained with these tugs is applicable to larger ships and an impressed current system using graphite anodes has been designed for an active ship of 16,000 sq. ft. underwater area and will be fitted shortly. The trials also indicate that if alternating current is available on the ships, A.C.-D.C. rectifier units would be preferable to motor generators with the resultant interruption of cathodic protection. If motor generators are used, it is imperative to have a spare on board at all times.—*K. N. Barnard, G. L. Christie and J. H. Greenblatt, Corrosion, Vol. 9, August 1953; pp. 246-250.*

Cargo Liners for Mediterranean Trade

In 1951 an order was placed by the Cunard Steam-Ship Co., Ltd., for three 14-knot motorships which are to serve the Mediterranean and Levant trades of the Company. The first of these was delivered recently and sailed on her maiden voyage from Liverpool to Greece and Turkey. The *Pavia,* as the ship is named, has the following principal particulars, which also apply to the sister vessels *Lycia* and *Phrygia.*

Length b.p.	320ft. 0in.
Length o.a.	348ft. 3in.
Moulded breadth	49ft. 6in.
Draught	23ft. 4in.
Deadweight capacity	4,400 tons (approx.)
Gross tonnage	3,410 tons
Nett tonnage	1,828 tons
Service speed	14 knots

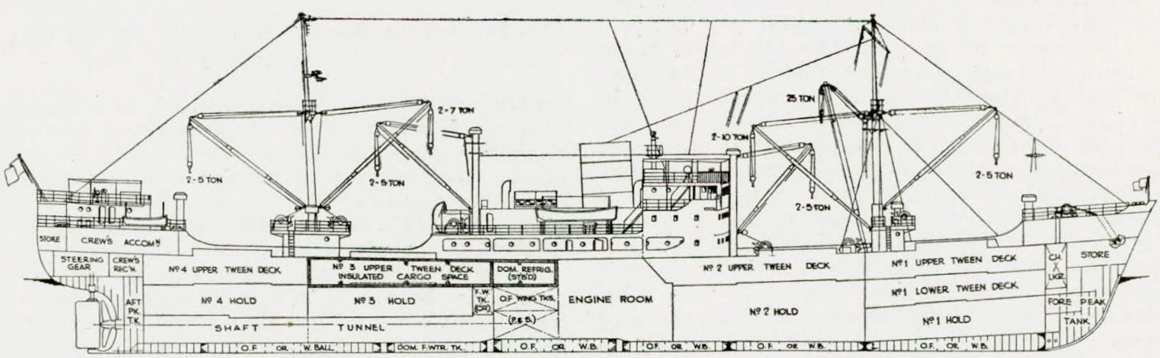
There are four holds with 'tween-deck compartments above each. The insulated cargo spaces, situated in No. 3 'tween deck are well subdivided for the carriage of different commodities, being cooled by four independent cross-current air coolers designed to maintain a temperature of 0 deg. F. in one and 10 deg. F. in the remaining three compartments. The main and auxiliary machinery was supplied and installed by David Rowan and Company Ltd. The main engine is a four-cylinder Rowan-Doxford unit with a bore of 600 mm. and combined piston stroke of 2,320 mm. There are two side-lever-driven scavenging air pumps attached to the centre crossheads of Nos. 1 and 2 cylinders, the beam from the latter also driving a forced lubrication pump, a cylinder-jacket and piston cooling-water pump, and a salt-water pump. The cylinder jackets and pistons are cooled by fresh water and a fresh water cooler is installed with sea-water circulation from the engine driven sea-water pump. Independent pumps for standby duties are a Simplex steam-driven forced lubrication pump and a Duplex steam-driven jacket and piston cooling pump. The ballast pump can be arranged to circulate salt water through the fresh water and lubricating oil coolers. For valve cooling, by fresh water, there are two pumps, one motor-driven and one steam-driven, each capable of full duty. The distilled fresh water storage tank is installed at a sufficient height to supply the separate service tanks by gravity. The engine-driven and independent fresh water cooling pumps draw from one tank and the valve cooling pumps draw from the other. The main engine can burn either Diesel or heavy fuel oils. Separate gravity tanks and connexions are provided for each class of fuel with connexions to two centrifugal oil fuel purifiers.—*The Marine Engineer and Naval Architect, Vol. 76, August 1953; pp. 321-323.*

Filiform Corrosion

Filiform corrosion is a thread-like type of corrosion which develops under protective coatings on certain metals, usually in humid atmospheres. It is characterized by directional growth and definite structure. Controlling factors are the physical nature of the coating and the exposure conditions. Normally each growing thread consists of an active head of an unstable corrosion product and an inactive body of a stable corrosion product. Filiform corrosion usually appears in an apparently haphazard or disorderly pattern. However, its growth is uniform and orderly in direction, rate and dimensions when developed under controlled conditions. The tracks do not cross one another but deflect or join in a predictable manner. To date no direct relationship has been found between filiform corrosion and the metallurgical pattern of the surface, the presence or absence of light, biological activity or the presence of inhibitive pigments. It has been observed under both non-metallic and metallic coatings, provided that the coatings are semi-permeable and are sufficiently elastic to yield without rupture to the volume of the oxide formed. A theoretical explanation is offered for filiform corrosion, involving an initiating force, a driving force and a directing force. It is suggested that corrosion anodes are initiated by an electrolytic mechanism as in ordinary corrosion. The driving force is the diffusion of the corrosive atmosphere into the active head, resulting in further metallic corrosion. The directing force is explained on the basis of concentration cells.—*M. Van Loo, D. D. Laiderman and R. R. Bruhn, Corrosion, Vol. 9, August 1953; pp. 277-283.*

Propeller Manufacture

During the past twenty-five years there have been very considerable changes in the design of marine propeller blades. In the merchant-ship field, for example, most propellers now have a variable pitch from root to tip, the shape of the blade sections varies from radius to radius, and it is quite common to find that while the inner blade-sections have a rounded or positively cambered face, the outer sections will have a flat face, or may even have a hollow or negatively cambered face. The adoption of these new sections has led to an entire change of attitude towards the degree of precision with which the blades of marine propellers require to be defined and finished. Improvements in design technique have called for improved methods of manufacture, and the propeller industry has not been slow in providing the necessary means for achieving the desired objective. In the first place, there has been the introduction of the sand-cement process for the construction of



Inboard profile of the new Cunard cargo liner propeller moulds to replace the earlier brick and loam construction. This has led to improved rigidity of the moulds and consequently to an improvement in the accuracy of the initial propeller castings. Then there has been the development of new machines enabling the whole pitch face of the blades to be planed with a single-point tool to a uniform or variable pitch, thus providing an accurately machined surface, or base line, for the subsequent definition of the shape of the blades. And, finally, there have been developed new and improved instruments for marking off and measuring the blades, and there is also the more general use of templates to define the shape of the ends of the blade sections. The position reached at the present time is that the largest marine propellers, up to 22 feet or so in diameter, are finished to standards of dimensional accuracy which, when viewed in relation to size, are not possible for the corresponding models used in connexion with self-propelled ship model tests. For example, the thickness tolerance of 1 mm. applied to the outer parts of the blades of a propeller 20 feet in diameter would be represented by 0·0015 inch on a model 9 inches in diameter.—*Paper by Professor L. C. Burrill, read at the Autumn Meeting of the Institution of Naval Architects, 16th September 1953.*

Fuel Consumption of 10,000-ton Motorship

The m.s. *La Hacienda* returned to London after the completion of a maiden voyage of 37,647 miles at a mean speed of 13·09 knots for the whole trip on an average daily consumption for all purposes of 12·23 tons of fuel, of which the main engine accounted for 11·5 tons of boiler oil, the auxiliary engines operating on Diesel fuel. The propelling engine output was 3,600 i.h.p., the speed 104·9 r.p.m. and the specific fuel consumption (for all purposes) 0·299lb. per i.h.p. hr. The cylinder lubrication amounted to 10·7 gallons per day and that for the engine 6·8 gallons, the total, including other requirements, being 18·66 gallons daily. There was a negative slip of 1·2 per cent; these figures are averaged for the complete voyage. The engine did not have a single stop at sea and there were no machinery troubles, apart from the usual small incidents associated with the maiden voyage of every ship. The *La Hacienda* is a vessel of just over 10,000 tons total deadweight and is equipped with a standard four-cylinder Swan Hunter-Doxford engine having cylinders 670 mm. in diameter with a piston stroke of 2,320 mm., the designed speed being 115 r.p.m. The length is 465ft. 9in. (435ft. b.p.) the beam 60ft., and the depth to the shelter deck 39ft.—*The Motor Ship, Vol. 34, October 1953; pp. 284-285.*

Combination Propulsion Plants for Naval Vessels

A major consideration in the design of a combination plant is the division of power between steam and gas-turbine booster power, this paper states. The optimum division of power must be made on the basis of operational requirements and will depend on the type of vessel, maximum desired cruising speed, the relative machinery weight, and the astern power requirement. At first it would appear that use of the lowest possible cruising power would give the maximum gain in weight of machinery plus fuel, but this is not quite true for several reasons. First, the weight of auxiliary machinery is not proportional to the cruising power. Secondly, the astern power requirement usually fixes the minimum capacity of the steam base load or cruising plant because boiler capacity must be provided. Thirdly, the operating time of the gas turbines should be kept to a minimum. In determining the division of power, the maximum speed required with the cruising plant is important from an operational point of view because it is desirable, when cruising in formation, not to have to change from one plant to the other. Calculations indicate that a steam-to-gas turbine ratio of about 50 per cent is probably desirable for most light-displacement high-speed combatant vessels. This higher percentage of steam power reduces the maintenance of the gas-turbine portion of the plant and adds appreciably to its life. The advantages of combination machinery plants are as follows: 1. A decrease in overall weight and space, resulting in more speed for the same displacement, increased endurance, of greater installed power. 2. A decrease in maintenance and repair; also easier handling during maintenance of the smaller, lighter machinery. 3. Quick-starting. The ship can get under way on the gas turbines alone. This is a limited advantage because the ship cannot be backed down unless controllable pitch propellers or reverse gears are installed. 4. Fast acceleration from maximum cruising power to full power. The gas turbines require a very short warm-up period and will provide quick power. The main disadvantages of combined plants are as follows: 1. The necessity for selecting a fuel acceptable to the gas-turbine booster units. When the base-load plant does not burn the same fuel as the gas turbines, the percentage of gas-turbine fuel to base-load plant fuel must be decided by operational requirements. 2. The arrangement requires a rather special gear configuration to accommodate the larger number of pinions. 3. Some extra ingenuity is required in disposing the various turbines around the gear. There are several different types of combination plants, such as: (1) Steam and gas turbine; (2) gas turbine and gas turbine; (3) free piston and gas turbine; (4) Diesel and gas turbine. Each of these fundamental types has several possibilities, and are outlined briefly in the paper.—*Paper by J. J. McMullen, Bureau of Ships, U.S. Navy Department, Washington, D.C., given before the 1953 A.S.M.E. Semi-Annual Meeting; Paper No. 53-SA-71.*

Diesel Engine Repair

Those who have had much to do with the running or maintenance of Diesel engines will sooner or later have experienced the cracking of piston crowns or cylinder heads, and the rapid wear of air and exhaust valves and their seatings. In some larger engines the piston crown is separate from the piston skirt. When cracks occur in the head it is easy to unbolt it, place in a lathe and bore out the cracked portion. Next drill a hole right through the head, as shown in Fig. 1, which illustrates a 20-in. diameter piston head. After removing the head from the lathe, turn and fit a mild steel washer into the bored-out recess as shown in the same sketch. Grind the

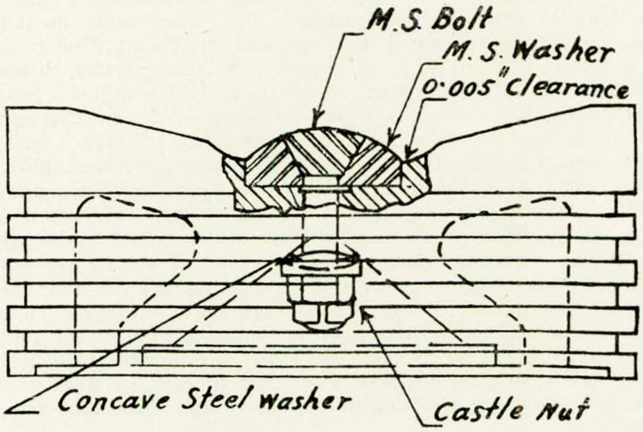


FIG. 1—*Showing the method adopted for repairing a cracked 20-in. diameter piston head*

washer into the head to ensure that the joint is gastight. Next turn and fit a bolt, as shown, grinding the bolt head into the recess provided in the washer. Leave a clearance of 0·005 inch between the periphery of the washer and the bored-out part of the piston head to allow for any unequal expansion between them. Now fit a concave mild steel washer at the bottom of the bolt as shown, screw up the castellated nut and insert a split pin. The bolt passed through the washer and piston head should not be too tight a fit in either. Piston heads thus treated have remained in service for at least ten years. In the particular instance sketched, two four-cylinder engines were involved, so the economy effected will be appreciated.—*I. E. K. Peterson, Gas and Oil Power, Vol. 48, July 1953, pp. 169-170.*

ENGINEERING ABSTRACTS

Section 3. Shipbuilding and Marine Engineering

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Kinetic Study of Deterioration of Crankcase Oils

Mathematical equations are derived which describe the kinetics of the deterioration of I.C. engine crankcase oils in its various aspects, such as the accumulation of oil-insolubles and soluble oil oxidation products and the depletion of additives. By the application of these equations it is shown how a closer understanding can be obtained of the mechanisms of oil deterioration. When applied to additive depletion the kinetic approach is shown to be useful in throwing light on the causes of additive depletion. In addition, the kinetic approach makes possible the calculation of the optimum additive concentrations required for specified engine operating conditions and of the maximum oil change periods. The kinetic approach to oil deterioration phenomena suggests how engine tests can be designed to give the maximum information.—*J. H. Hughes and J. B. Matthews, Journal of The Institute of Petroleum, Vol. 39, July 1953; pp. 463-475.*

Deterioration of I.C. Engine Crankcase Oils

The significance of lubricating oil oxidation tests is discussed with particular reference to the catalytic conditions employed. From experimental data it is deduced that two lacquer-forming reactions may occur in engines. The first is a slow reaction, the effects of which can be predicted from oxygen absorption measurements on the oil, provided that the appropriate catalytic conditions are chosen for the test. The second is a much more rapid reaction, which only occurs above a critical temperature dependent on the oil, is controlled by an adsorption mechanism, and is mainly responsible for ring-sticking caused by oil oxidation. Apart from the deleterious results produced by oil oxidation, some beneficial results can be obtained if a certain class of oxidation product is produced in service in sufficient quantity. In engine tests this oxidation product prevents sludge deposition and improves ring-sticking properties.—*J. H. T. Brook, J. B. Matthews and R. P. Taylor, Journal of The Institute of Petroleum, Vol. 39, July 1953; pp. 454-462.*

Radiation and Furnace Design

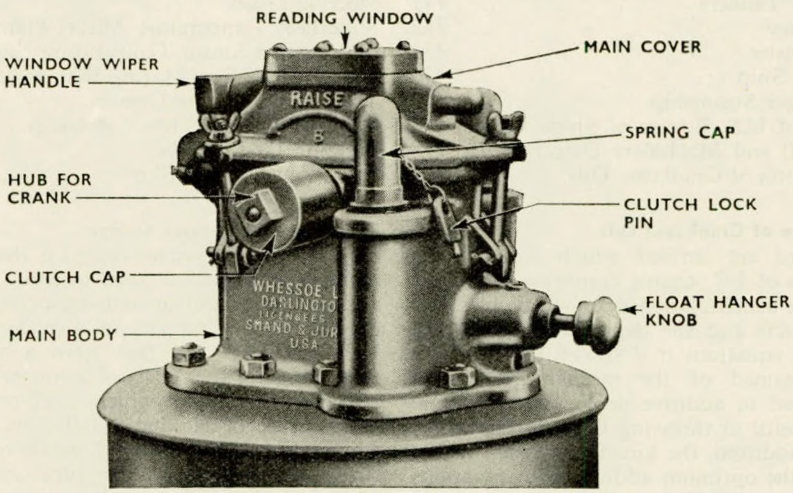
The art of furnace design is that of transferring heat from flame or combustion gases to solids by conduction, convection and radiation, and in high-temperature furnaces one must rely on radiation. Comparison is made between the radiation from CO_2 and H_2O and that from a black body. The radiation from burning fuel oil is of a higher order owing to the carbon particles, the size of which is of primary importance. As the temperature of a flame falls, its emissivity falls, but the emissivity of H_2O and CO_2 tends to rise with falling temperature. In low-temperature furnaces the material on one side of the metal wall frequently cannot remove the heat transferred by the flame to the other side. This problem is tackled by reducing the combustion intensity, by screening and by lowering the combustion-chamber temperature by recirculation of the products of combustion. In shell boilers the heating surface must be correctly divided between that receiving radiation and that heated by convection. In watertube boilers the high emissivities may lead to trouble in the exposed tube banks if water treatment is neglected. In high-temperature furnaces the flame must be stretched to cover the receiving surfaces, but when using a flame of emissivity of 0·5-1, gaps must be left round or between flames to keep the roof at a reasonable temperature. The initial burner momentum must be such as to enable the flame to cover the desired distance in approximately 0·2 seconds.—*G. J. Gollin, Journal of The Institute of Fuel, Vol. 26, September 1953; pp. 151-162.*

Effect of Metal Oxide Smoke on Combustion Gases

A previous paper (Engineering Abstracts, September 1953, p. 128) has described the results of systematic tests on an oil-fired watertube boiler to determine the effects of (i) sulphur content of fuel oil, (ii) rate of firing, and (iii) combustion conditions, on the production of SO_3. The present work was undertaken to provide the quantitative data on the influence of various additives to the fuel burnt, as a method of decreasing the quantity of SO_3. Measurements were made with a dewpoint meter on three relatively small oil-fired appliances. Two of these were sectional boilers and the third, on which most of the work was done, was a refractory furnace. Five oils, varying from a heavy fuel oil (3·5 per cent sulphur) to a gas oil (0·75 per cent sulphur) were burnt in the refractory furnace and the dew-point of the combustion products was in the range of 250 to 300 deg. F. Measurements on the gases from the heavy fuel oil, treated with (i) soda residue (AS), (ii) calcium residue (AC), (iii) commercial zinc naphthenate (AZ), showed that the latter alone was successful in decreasing greatly the amount of SO_3. Under good combustion conditions there was no acid dew-point when burning oil containing 0·14 per cent zinc (by weight). With 0·07 per cent zinc the amount of SO_3 was still very low, a dew-point of 160 deg. F. being measured.—*D. Flint, A. W. Lindsay and R. F. Littlejohn, Journal of The Institute of Fuel, Vol. 26, September 1953; pp. 122-127.*

Automatic Marine Tank Gauge

The Shand and Jurs automatic marine tank gauge, which is now being manufactured in the United Kingdom by Whessoe, Ltd., is claimed to permit accurate gauging without loss of vapour and without the explosion and fire hazards attendant on open hatch gauging. The gauge consists of a float-operated tape which can be lowered into the tank by means of a crank, the tape being read through a glass sighting window on the top of the cover. When not in use the float is held at rest on a hanger. This prevents damage to the float during voyage or, should a pressure cleaning system be installed, when the tank is being cleaned. The heavy glass reading window is equipped with a wiper. The gaugehead may be mounted either flush with the deck or on a trunk to elevate it above the wash. This type of gauge has been effectively used in many tankers built in the United States, particularly by the U.S. Navy and the Standard Oil Company of California. Recent orders have included equipment for tankers of 32,000 tons d.w. building in Italy for Fratelli d'Amico, of Rome.—*The Shipping World, Vol. 129, 16th September 1953; p. 237.*



The Shand and Jurs automatic marine tank gauge manufactured by Whessoe, Ltd.

New German Tanker

For the time being, the motor tanker *Ernst G. Russ,* built by the Deutsche Werft for Mr. Ernst Russ, is the largest vessel in the German mercantile fleet, and the first of a series with similar characteristics ordered from the same builders by different owners. The vessel has a well-raked stem and a cruiser stern and is constructed to Lloyd's highest classification. There are 10 centre tanks and 12 side tanks, the total capacity being 871,262 cu. ft., allowing 2 per cent for expansion. A cargo hold is arranged forward, the capacity being 29,360 cu. ft. (grain). In the following table are the main details:—

Length o.a.	581ft. 4in.
Length b.p.	540 feet
Breadth, moulded	72 feet
Depth, moulded	37ft. 10in.
Deadweight capacity	18,290 tons
Corresponding summer draught ...	30ft. 3in.
Gross register	12,984 tons
Net register	7,488 tons
Service speed, loaded	14 knots

There are two main pump rooms, one between Nos. 4 and 5 tanks, and the other between Nos. 7 and 8, in each compartment being two Atlas-Werke duplex steam pumps. Each pump has a capacity of about 400 tons an hour. In the bilge and ballast pump room forward are two duplex steam pumps for fuel and water. The *Ernst G. Russ* is propelled by an eight-cylinder two-stroke single-acting M.A.N. engine of the cross-head type, developing 7,200 b.h.p. at 115 r.p.m. The cylinder diameter is 780 mm. and the piston stroke 1,400 mm. It is calculated that the fuel consumption will be approximately 28 tons daily.—*The Motor Ship, Vol. 34, October 1953; pp. 290-293.*

French Twin-screw Passenger Steamship

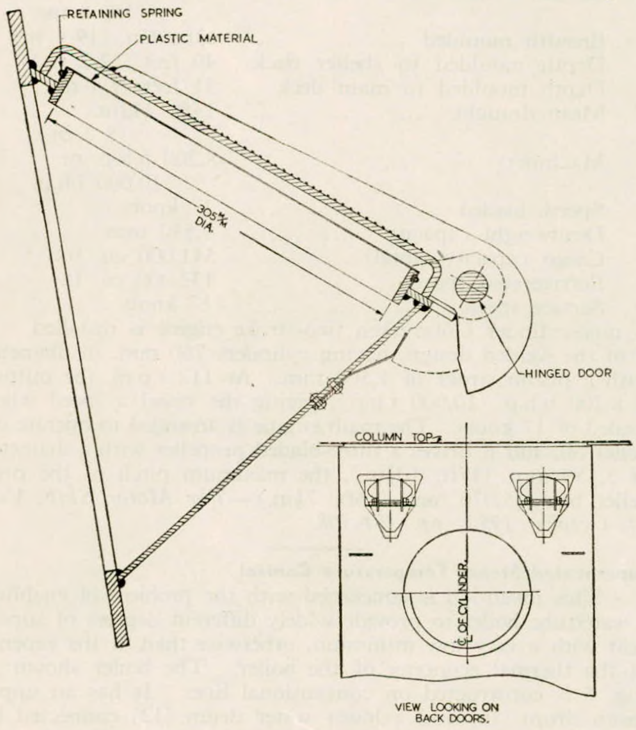
The latest addition to the fleet of the Compagnie Générale Transatlantique, Ltd., is the twin-screw passenger steamship *Antilles,* built at the Naval Dockyard, Brest, to plans prepared by Soc. des Ateliers et Chantiers de France, of Dunkirk. The *Antilles* is, of course, a sister ship of the *Flandre,* which entered service towards the end of last year. The principal dimensions and other leading particulars of the *Antilles* are given in the accompanying table:—

Length overall...	597ft. 0in.
Breadth moulded	80ft. 0in.
Breadth extreme	80ft. 3in.
Depth moulded to promenade deck...	56ft. 1in.
Gross tonnage	20,464
Number of passengers...	777
Speed in service, knots	22
Corresponding s.h.p.	36,000
Maximum s.h.p.	44,000
Steam conditions:—	
Pressure, lb. per sq. in.	900
Temperature, deg. C.	480

The *Antilles* has a cargo-carrying capacity of 5,035 tons, of which 590 tons are insulated. The cargo is carried in four holds, three forward and one aft. The cargo hatchways are served by a comprehensive outfit of electric deck cranes, which are disposed as follows:—Four, each of 5 tons, serving Nos. 1 and 2 holds; two, each of 3½ tons, serving No. 3 hold; and two each of 2½ tons, serving No. 4 hold. These cranes, which are operated by high-speed electric winches, have a working radius of 16 feet. The main propelling machinery consists of two sets of Rateau-Bretagne double-reduction, geared turbines developing 36,000 s.h.p. in normal operation, and capable of developing a maximum of 44,000 s.h.p. Steam is supplied at a pressure of 900lb. per sq. in., and a temperature of 480 deg. C., by four (type P-41) forced-draught, rapid-firing, watertube boilers, constructed by the Chantier et Ateliers de Saint-Nazaire-Penhöet, Saint Nazaire-sur-Loire. Electric power is provided by two generating plants, which are accommodated in separate watertight compartments. The main plant consists of two groups of turbo-alternators, each of which has an output of 2,000 kW. The secondary plant, which is for use when the ship is at anchor, consists of two groups of Diesel alternators, each capable of developing 750 kW.—*The Shipbuilder and Marine Engine-Builder, Vol. 60, October 1953; pp. 582-588.*

Crankcase Explosion Door

The accompanying illustration shows the crankcase relief valve developed by William Doxford and Sons, Ltd., and now being fitted on the Doxford oil engines built under licence by John Brown and Co., Ltd., Clydebank. Crankcase explosion doors have also been fitted to Fairfield-Doxford engines. None has yet been fitted to the engines built by Doxford and Sons themselves, approval being awaited on a modified design which has been tested at their Pallion Works, Sunderland, before representatives of Lloyd's Register and the Ministry of Transport. It will be seen that the door is simple in construction and meets the main requirements for this purpose:—(a) That the gases can be released at a rate which ensures that the pressure is kept within the safe working pressure of any part of the crankcase, and (b) that immediately the excess pressure has been released, the valve is instantly closed to prevent the ingress of air, and a second explosion. The weighted door is hinged at the bottom and lifts at a pressure of 2½-3lb. per sq. in. A plastic cover to retain the oil mist is fitted across the port and secured by a spring.—*The Motor Ship, Vol. 34, September 1953; p. 248.*



Details of crankcase relief door

Tests of Launching Greases

An extensive investigation into the frictional properties of launching greases has been carried out for the British Shipbuilding Research Association by John Brown and Co., Ltd., Clydebank. Four series of tests were made, three of which were concerned with Tallene grease, the fourth with mineral launching grease. Details of tests, testing procedure and records are given; the principal results are presented in a series of diagrams and discussed in the text. An important feature was observed, namely the variation in frictional values over the first two feet of travel, and the much greater uniformity for travels of slider beyond two feet. Valuable results were obtained from the third series of tests in which the force acting on the launching trigger was measured by weighing the force required to "hold" the slider. These forces were found to be of considerable magnitude. This aspect of the tests could be of great interest to the practical shipbuilder. The results of these tests indicated a striking disparity between the coefficient of static friction and the coefficient of initial sliding friction. A hypothetical explanation of this phenomenon is offered. The fourth series of tests on mineral launching grease produced important results when compared with those for Tallene.—*Paper by J. Brown and J. M. Ferguson, read at the Autumn Meeting of the Institution of Naval Architects, 15th September 1953.*

Investigation of Ships' Hull and Machinery Defects

This paper presents a number of examples of techniques employed by the Research Department of Lloyd's Register of Shipping in investigating machinery defects. Strain gauges have been used with satisfactory results in the measurement of surface strains on submerged parts such as propeller blades, rudders, pressure vessels, hydraulic cylinders internally, ships' underwater shell plating, etc. Strain gauges were used in an investigation of the tightening and running stresses in large bolts as installed in the running gear of large oil engines. Where ample access is available, the normal method of slogging up using a heavy hammer and ring spanner can produce initial tightening stresses in a 3-in. diameter bolt greatly in excess of those which are necessary. It is, however, problematical as to what the initial tightening stresses might be with large coupling, bottom end, crosshead or piston rod bolts which have to be hammered up in the cramped confines of the crankcase. Experience of a number of such failures which have almost invariably indicated that the soundness of the material was never in question, leads one to the conclusion from an examination of the fractured surfaces that it is under-tightening which has been responsible. This suggests that unless a regular and conscientious crankcase inspection involving sounding and hardening up of running gear bolts is carried out, there is always the possibility of the pre-stress in the bolt being reduced under running conditions to such proportions as would permit dynamic stresses to be imposed, which at points of stress concentration, and aggravated by local bending, might reach such proportions as would quickly cause failure. Heavy explosions at starting, racing in heavy weather are two factors which make an early inspection of such bolts a prudent safeguard. Much can still be done towards improving certain designs of bolted assemblies. Shank relief, generous fillets, controlled and positive pre-stress and adequate locking are all factors which increase the safety margin. The coupling bolt which theoretically has merely to transmit shaft torque as a shear strain, for which it is adequate, in practice may be subject to cyclical bending stresses of large magnitude due to malalignment which may be produced in lining out or in service resulting from unequal wear down of bearings. It is under exceptional circumstances that the hogging, sagging or vibratory movements of a ship's hull will, in themselves, cause bending strains of serious proportions in the shafting. It is well proven and now generally accepted that unless the elasticity of the bolt is greatly in excess of that of the bolted assembly in a dynamic system, the bolt may be subjected to appreciable dynamic stress for which it is not a suitable design. Thus it would be clear that in crankshaft or lineshaft couplings which are liable to be subject to bending, the design should ensure that the requirements as to relative elasticity have been met. Therefore, particularly under these conditions, it is not good practice to drive or force coupling bolts through a fit that is so hard as to ensure that the effective elasticity of the bolt in tension is practically nil, nor is it good practice to have coupling faces butting over their whole area of contact; adequate relief should be given to ensure that the bolting load is concentrated in way of an annular fitting strip in radial depth about one and a half times the bolt diameter above and below the pitch circle of the bolts. As in the case of the bolts fitted in the running gear of the engine, adequate shank relief with generous fillets should be provided and the fitted portion of the shank at the butting surfaces of the coupling should be reduced to a minimum, say one-half the reduced shank diameter on either side of the abutment. Holding-down bolts can be another source of constant trouble, but here again, observing the same principles already discussed, failures can be avoided even under severe conditions of engine unbalance or transverse rocking modes of vibration of the engine, it being assumed that the tank top plating or seating is adequate for the job. The size of chocks should be kept down to a minimum and the fitting surfaces should preferably be annular around the bolt. Studs are much inferior to through bolts and should be avoided wherever possible. Adequate pre-stress of about 8 tons per sq. in. should be provided and a positive locking of the nuts arranged. It is much more difficult to make a sound job of rechocking an engine which has fretted into its chocks than to start off with a sound, well-designed and fitted, bolted arrangement.—*Paper by T. W. Bunyan, read at the Autumn Meeting of the Institution of Naval Architects on 16th September 1953.*

New Method for Removing Slag Deposits

An article in Power Engineering describes a specially designed pellet gun for removing slag deposits in very high duty furnaces and in furnaces using particularly low quality fuel developed recently by engineers of the Diamond Power Specialty Corp. Tests, in which use of 0·22 calibre rifle bullets, various bore shotguns (previously used off and on for more than fifty years to remove slag) and even small calibre machine guns resulted in the adoption of compressed air as a propelling medium on account of the high cost of explosive cartridges. Small steel shot of B-B calibre were first selected as pellets, and during development, care was taken to perfect a feeder with no moving parts, low power requirements, uniformity of

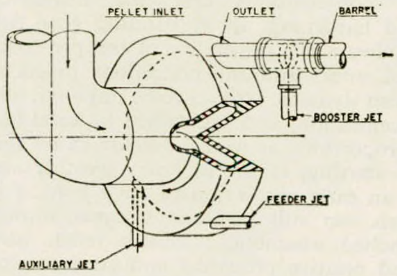


FIG. 4—*Diagrammatic arrangement of pellet feeder unit of latest design*

feed rate and portability. The first model gun resulting from this investigation, using $\frac{9}{16}$-in. diameter glass marbles, was operated successfully to remove slag from local areas on boilers while on load. It proved in principle that pellets could be used for this purpose. The final design of the pellet feeder mechanism is outlined in Fig. 4. Pellets are fed into the centre of a hollow disc by gravity. A feeder jet of air enters the disc at the bottom and tangentially, and produces circulation of pellets in the feeder. The outlet, placed 180 degrees from the jet inlet, allows pellets to pass one at a time and discharges them into the barrel, the feed rate having been successfully varied from 50 to 250 pellets per second. Use of compressed air is comparatively economical. Supplies at pressures between 80 and 250lb. per sq. in. have been successfully used, giving air consumption rates varying from 100 to 340 c.f.m. free air per feeder. A booster jet further downstream admits compressed air at a pressure higher than that in the feeder. This pressure governs the force with which the pellets are discharged. Striking power of the pellets varies with barrel length, mass of the pellet, booster air pressure and diametrical clearance between the pellet and the barrel. Velocities as high as 1,400ft. per sec. have been recorded. A gun with a 24-inch barrel and 70lb. per sq. in. tank pressure in one test shot pellets at a velocity of 571ft. per sec. Feed rate and velocity of pellets are completely divorced in this model. Further study is also being carried out to determine a suitable and economical material for all types of pellet guns. Glass marbles, wooden beads, plastic beads, screened gravel, compressed slat, asphalt and asbestos mixtures have been considered, and at present $\frac{1}{2}$-in. diameter asphalt-asbestos pellet is considered the best.—*Engineering and Boiler House Review, Vol. 68, September 1953; pp. 286-287.*

9,350-ton 17-knot Ship

For their service between Norwegian and other North European ports to the Pacific coast of North America (Los Angeles, San Francisco, Seattle, Vancouver and other ports), Fred Olsen and Company recently took delivery of the motor vessel *Bonanza,* which was constructed by Götaverken at Gothenburg. She is, in general details, practically a sister ship of the *Buffalo,* except that the machinery is of a different type. The *Buffalo* has been in satisfactory service for some months, operating on boiler oil and Diesel oil alternately. The main particulars of the *Bonanza,* which was constructed to the highest class of Det Norske Veritas, are as under:—

Gross register	7,147·74 tons
Net register	4,071 tons
Length overall	511ft. $1\frac{1}{2}$in. (155·8 m.)
Breadth moulded	63ft. 6in. (19·4 m.)
Depth moulded to shelter deck	40 feet (12·2 m.)
Depth moulded to main deck...	31 feet (9·4 m.)
Mean draught	26ft. $11\frac{3}{4}$in. (8·2 m.).
Machinery	8,200 b.h.p. or 10,000 i.h.p.
Speed, loaded	17 knots
Deadweight capacity	9,530 tons
Cargo capacity (total)	541,000 cu. ft.
Refrigerated space	132,000 cu. ft.
Service speed	17 knots

A nine-cylinder Götaverken two-stroke engine is installed. It is of the welded design, having cylinders 760 mm. in diameter with a piston stroke of 1,500 mm. At 112 r.p.m. the output is 8,200 b.h.p. (10,000 i.h.p.), giving the vessel a speed when loaded of 17 knots. The main engine is arranged to operate on boiler oil, and it drives a three-bladed propeller with a diameter of 5,450 mm. (17ft. $10\frac{1}{2}$in.), the maximum pitch of the propeller being 5,070 mm. (16ft. $7\frac{1}{2}$in.).—*The Motor Ship, Vol. 34, October 1953; pp. 304-308.*

Superheated Steam Temperature Control

This invention is concerned with the problem of enabling a watertube boiler to provide widely different degrees of superheat with a very low minimum, otherwise than at the expense of the thermal economy of the boiler. The boiler shown in Fig. 5 is constructed on conventional lines. It has an upper steam drum (10) and a lower water drum (12) connected by steam generating tubes (14). The tubes are arranged in a furnace (18) fired by a number of burners (20) supplied with fuel and with preheated air. At the outlet end of the furnace, a duct (22) is provided which contains a two-stage superheater. Steam from the drum (10) passes through the line (24) and through two banks of tubes (26) which are connected in series and form the "earlier superheater stage". The steam then passes through the line (28) to a heat exchanger or desuperheater (30) and thence through the line (32) to a further bank of tubes (34) in the superheater duct (22), which bank comprises the "later superheater stage". The superheater duct (22) is in communication with the furnace at the bottom so that the furnace gases which pass through it flow countercurrent to the steam. It also has a side inlet (38) between the two superheater stages. Dampers (40) and (42) are provided, and they can be set in such a position as to close the side inlet (38) and allow free passage for the furnace gases through the two superheater sections as shown in Fig. 5, or to open the side inlet and close the superheater duct (22) between the two stages and thus bypass the later superheater

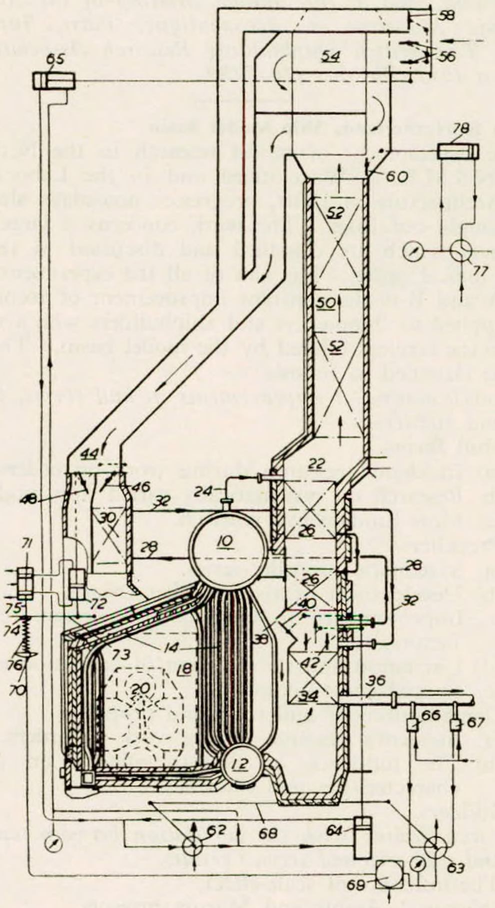


FIG. 5

stage. The desuperheater (30), through which the steam passes between the two superheater stages, is arranged in a duct (44) through which air is passed to the burners (20). Dampers (46) and (48) are provided, and these can be set so as to cause a portion or all of the air to flow through the desuperheater or to bypass the desuperheater. On leaving the superheater duct (22), the heating gases pass through a heat recovery section (50) including an economizer (52) and an air preheater (54). Dampers (56) and (58) are provided which can be set to cause the air to flow through the heater (54) as shown in Fig. 5, or to bypass the heater. When a high degree of superheat is required, the various dampers are set as shown in Fig. 5. When a low degree of superheat is required the dampers (40), (42), (56) and (58) are set in the opposite positions to those shown in Fig. 5. In that case the furnace gases enter the superheater duct through the side inlet (38) and bypass the later superheater stage (34), and the air bypasses the preheater (54). As the air which reaches the desuperheater (30) is not preheated, it extracts a relatively large amount of heat from the steam which is superheated in the earlier superheater stage (26), and as the steam is not further superheated in the later superheater stage (34), it issues from the outlet pipe (36) at a relatively low temperature. The extent to which the steam is desuperheated depends on the setting of the baffles (46) and (48). Although the air extracts a relatively large amount of heat from the steam, it is not overheated, because it is caused to bypass the preheater (54). Instead of, or in addition to, causing the air to bypass the preheater (54), the furnace gases can be caused to bypass that heater by control of the damper (60).—*Brit. Pat. No. 693,910 issued to Foster Wheeler, Ltd. Complete specification published 8th July, 1953. Engineering and Boiler House Review, Vol. 68, September 1953; pp. 289-290.*

Electrical Characteristics of Tankers

The tanker presents quite a departure from passenger liners and the cargo ship with regard to the electrical load distribution. Whereas the loadings in the liner and cargo ship are broadly speaking well spread throughout the midships and after sections with lighter loading at the fore end of the ship, the electrical loading in the tanker is almost wholly concentrated in the engine room and its vicinity, with about 95 per cent of the total connected load, leaving only 5 per cent for the bridge and vicinity, which is usually situated slightly forward of amidships. A steam-driven windlass is invariably installed in tankers, rendering the forward electrical loading negligible. In the largest tanker subsequently reviewed the loading is 96 per cent for engine room and adjacent spaces and 4 per cent only for bridge distribution. Should 75,000-ton tankers be constructed, these proportions would probably advance to 97½ and 2½ per cent respectively. Thus the substantial credits in favour of the A.C. system with 440-volts, three-phase distribution due to the saving of long lengths of considerable numbers of smaller cables, tray plates, bulkhead glands, deck tubes and labour set out in previous articles on passenger liners cannot be realized in tankers, and the saving in A.C. credits on generator cables to switchboard and big h.p. engine room feeders is more than offset by the cumulative economy of the D.C. cabling for other connexions, even when approval is obtained from the classification societies to install 440-volts galley equipment. An even more prominent tanker characteristic is the very high percentage of cost due to the generating plant, with its erection piping, seatings and trials. The author has reviewed ten tankers ranging from 8,000 tons to 32,000 tons deadweight, and with one exception the (16,500-tons tanker with steam-driven auxiliaries) the percentage of total electrical costs of the generating plant ranges from 34 per cent to 62 per cent of the entire electrical cost excluding radio, radar and equipment supplied by the owners. The larger the tanker the higher the percentage cost of this item, and it seems possible that this percentage would progress to as much as 70 per cent in a 75,000-tons tanker.—*H. J. D. Thompson, The Shipping World, Vol. 129, 9th September 1953; pp. 211-212.*

Recent Canadian Icebreakers

The Canadian Department of Transport's new icebreaker *d'Iberville* is the largest and most powerful icebreaker on the North Atlantic Continent and in the Commonwealth. She has been designed specially for work in Arctic waters and equipped with up-to-date navigational equipment and has a range of 12,000 miles without refuelling. She is unlike many other icebreakers in having two screws aft and none forward. This is because her duty is partly that of a passenger and cargo carrying ship and only during certain seasons of the year is she employed as an icebreaker proper. The *d'Iberville* is 310 feet overall with a moulded beam of 66·5 feet, a depth to the forecastle of 40·25 feet and an extreme draught of 30ft. 5in. She has a dry cargo capacity of 47,300 cu. ft. and a refrigerated space of 5,150 cu. ft. She was built by the Davie Shipbuilding and Ship Repairing Company, a subsidiary of Canada Steamship Lines, Ltd., at Lauzon, P.Q. Her two Skinner Unaflow engines were built under licence by Canadian Vickers, Ltd., and each has six cylinders, developing a total of 10,800 i.h.p., taking steam from watertube boilers. A contrast to the steam-driven *d'Iberville* is the 10,000-s.h.p. icebreaker, H.M.C.S. *Labrador,* built at Sorel for the Royal Canadian Navy. This ship, which has an overall length of 269 feet, is a twin-screw Diesel-electric unit with main propelling machinery consisting of six Fairbanks Morse 10-cylinder opposed-piston engines, rated at 2,000 h.p. at 810 r.p.m., each driving a Canadian Westinghouse 1,375-kW. generator. There are also four seven-cylinder opposed-piston Diesel engines of the same make, each coupled to a 250-kW. alternator, to supply power for all the electrically-driven equipment in the ship, which includes an electro-hydraulic Denny-Brown stabilizer, heeling and trimming pumps, in addition to the ordinary ship's pumps. Another ship for Arctic service is the *C. D. Howe,* which, like the *d'Iberville,* is owned by the Canadian Department of Transport. This ship is designed for the Eastern Arctic Patrol Service during the summer months and for servicing and maintaining navigational aids in the Gulf of St. Lawrence and adjoining water in the winter. The ship is propelled by two Skinner Unaflow engines of 4,000 total i.h.p. taking steam from two watertube boilers. She is 276 feet in length between perpendiculars with a beam moulded of 50 feet and a depth moulded of 26 feet, and a deadweight on a salt water draught of 18ft. 6in. of 2,673 tons. Her loaded speed is 13½ knots. A further ship of icebreaking type for the Canadian Department of Transport, of all-welded construction and specially strengthened for navigation in ice, is the *Edward Cornwallis,* constructed to carry out lighthouse supply and buoy service on the Canadian Atlantic coast. The main buoy-handling derrick has a 25-tons lift and the ship is fitted up with special mechanical aids. Here again, two Skinner Unaflow engines are employed with an output totalling 2,800 i.h.p. They take their steam from three Scotch boilers. The loaded speed is 13½ knots. Length between perpendiculars is 240 feet, with a beam moulded of 43 feet, a depth to upper deck of 20ft. 6in. and deadweight with an 18ft. 6in. draught in salt water at 1,800 tons.—*A. C. Hardy, The Shipping World, Vol. 129, 2nd September 1953; pp. 182-183.*

Optical Marking-off System

The application of the optical marking-off system for plates and sections in shipbuilding during the last five years has proved that there are certain advantages over the classical method. An analysis of this classical marking-off method is given and, further, the new method is described in full. A comparison is made between the two systems, showing clearly the advantages of the optical method. Special attention is given to the advantages regarding man-hours, cost and space. A few examples of man-hour calculation are given. Reference is made to the application of the method in several shipyards. The arguments which have led to the introduction of the system into the reorganization of the yard in which the author is engaged are dealt with. A few remarks are given in connexion with the expected further development of the system, especially in face of the new full automatic marking-off and oxy-acetylene flame-cutting machine. A list of references is added.—*Paper by J. H. Krietemeijer, read at the Autumn Meeting of the Institution of Naval Architects, 15th September 1953.*

Criticism of Sea Trials

It is considered that sea trials in their present form give results which are subject to a number of inaccuracies, and that it should be possible to improve the techniques employed so that more valuable results are obtained. The various causes of error are first studied. The factors considered in this section of the paper include the surface roughness of the hull, the temperature of the water, the depth of water, inaccuracies in the actual measurements made, the current or tidal stream, the effect of waves, the wind, and movements of the rudder. The dispersion of the results obtained is then considered, and statistical methods of analysing them are discussed. These methods are illustrated by reference to the results of a large number of observations specially carried out on certain ships. The use made of the results obtained from sea trials is then discussed. The method normally employed for determining the fuel consumption is given and certain improvements are suggested. The laws governing the propulsion of ships are considered, and a method of evaluating the trial results is derived from these laws. In conclusion, the authors state that if full advantage is taken of the accuracy of existing instruments and methods of measurement, and if the execution of the trials is organized so that the best use may be made of the data obtained, it is possible to obtain results which will define the performance of the ship completely, and which will be applicable not only on trials but also in service.—*Paper by R. Brard and M. Jourdain, read at the Annual Meeting of the Association Technique Maritime et Aéronautique, Paris, June 1953. Journal, The British Shipbuilding Research Association, Vol. 8, August 1953, Abstract No. 7789.*

Research in Netherlands Ship Model Basin

The development of model research in the Netherlands, in the N.S.M.B. at Wageningen and in the Laboratory for Naval Architecture at Delft, progresses nowadays along carefully thought-out lines. The work concerns a large number of subjects which are classified and discussed in this paper in their logical order. The aim of all the experiments headed under A and B is the constant improvement of recommendations supplied to shipowners and shipbuilders who avail themselves of the services offered by the model basin. The subject matter is classified as follows:—

A. Research aiming at improvements in hull forms, propellers and rudders.

I. Hull forms.
   (a) Incidental research during work on orders.
   (b) Research on systematically varied ship models.
   (c) More fundamental research.
II. Propellers.
   (a) Systematic propeller-series.
   (b) Development of the propeller theory.
   (c) Improvement in methods of design (correction factors).
   (d) Cavitation and the problem of cavitation erosion.
   (e) Special constructions.
III. Combinations of hull-form and propeller.
   (a) Incidental research during work on orders.
   (b) The influence of flow-irregularity on cavitation characteristics and efficiency.
IV. Rudders.

B. Research dealing with the correlation between tank results and trial-trip and service results.

I. The influence of scale-effect.
   "Victory", Arabia and Marnix projects.
II. The influence of roughness of the hull form.
   D. C. Endert, Jr., project.
III. The influence of flowing and restricted water.
   Arabia project.
IV. The influence of wind and seaway.
   "Victory" project.

—*Paper by Professor W. P. A. van Lammeren, read at the Autumn Meeting of the Institution of Naval Architects, 15th September 1953.*

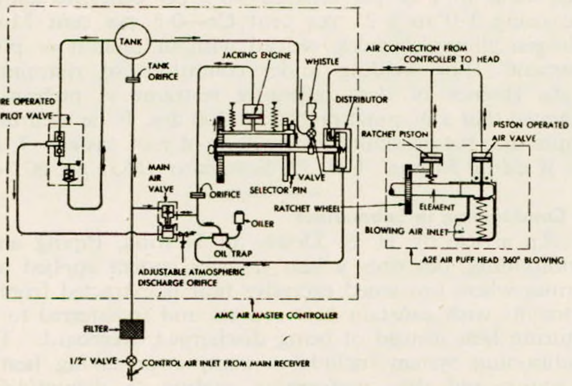
Hydromechanics Research of U.S. Bureau of Ships

This paper gives some account of the organization of naval research in the United States, and in particular describes the scope and methods of administration of the Bureau of Ships Programme of Fundamental Hydromechanics Research This programme is planned in close association with other agencies engaged in this field, such as the Office of Naval

Research, the Society of Naval Architects and Marine Engineers, the American Towing Tank Conference and various universities and laboratories. The research work under the programme is carried out partly in the Navy's own laboratories, partly in the universities, the responsibility for the planning and administration of all the various projects being vested in the Commanding Officer and Director of the David Taylor Model Basin. The paper describes these projects and some of the results obtained in different fields. An account is also given of recent developments in facilities and instrumentation.—*Paper by F. H. Todd, read at the Autumn Meetings of the Institution of Naval Architects, 16th September 1953.*

Boiler Cleaning by Air

The Babcock-Diamond air-puff soot-blower is now being manufactured in Great Britain for Babcock and Wilcox by Dewrance, Ltd., and a subsidiary company. The use of air instead of steam for blowing affords a valuable saving of make-up water and shows also a definite thermal gain over a steam system. Operation is extremely simple, the blowing sequence being initiated by opening two valves, after which the operation is entirely automatic. The boiler is thoroughly cleaned and, because the blowing cycle is spread over a relatively long period —usually 3 to 4 hours, depending on the number of blowers— there is no soot nuisance on deck or disturbance of boiler operating conditions. Air for the system is supplied by a compressor discharging into a receiver of approximately 100 cu. ft. capacity at 125lb. per sq. in., and it is common practice to provide a standby compressor. The system is operated by a master controller using air from the receiver, supplied via a



Arrangement of master control and soot-blower unit

small pressure control tank. During the blowing, each soot-blower element produces a series of "puffs" of one second duration, during each of which the element is rotated automatically through an arc of 17½ degrees by means of an air engine operating a ratchet mechanism. After each "puff", the element remains stationary for one minute, while the air compressor replenishes the receiver, and is then rotated through a further 17½ degrees, and so on, until it has completed a blowing arc of up to 360 degrees. The master controller then shuts off the air supply to that blower and transfers it to the next blower in the sequence. After the final blower in the sequence has been operated, a whistle blows continuously until the operator shuts off the air supply to the master controller. Although operation of the air-puff system is automatic, the normal sequence can be interrupted at any time during the cycle to enable the blowing of a particular element to be repeated or to bypass the blowing of an element—by turning the knob of the indicator dial to the number of the element which it is desired to blow. While it is possible to arrange the control mechanism to cover more than one boiler unit, it is common practice to have one master controller for each boiler. The compressor is automatically controlled to shut off when the receiver pressure is slightly higher than the working pressure and the setting of the master controller is such that the compressor is working at a steady load throughout the blowing period. The blower elements used in the air-puff system follow the design of the well-known Babcock-Diamond steam soot-blowers. While the vessel is in port the air compressors of the air-puff soot-blower system can be used for the operation of boiler-tube cleaners or air-operated tools and for general scavenging purposes. They can also be used as a standby to the compressors of an automatic combustion control system.—*The Shipping World, Vol. 129, 26th August 1953; p. 165.*

Special Service Vessel

The twin-screw steamship *Vigilant* is a buoyage, salvage, survey and fire-fighting vessel built by John I. Thornycroft & Co., Ltd., for the Mersey Docks and Harbour Board, and intended for service in Liverpool Bay and the Mersey Estuary. The vessel will be engaged in the regular periodical servicing of over 100 automatically illuminated buoys and one lightship in the River Mersey and its environs. The leading particulars are as follows: —

Length overall	172ft. 6in.
Length b.p.	160ft. 0in.
Breadth moulded...	35ft. 0in.
Depth moulded to main deck	16ft. 6in.
Mean load draught	11ft. 1in.
I.H.P. at full power	1,450
Corresponding r.p.m.	198
I.H.P. at service power	1,300
Corresponding r.p.m.	188
Speed at full power, fully laden, knots	12½

The compressed-air equipment for diving purposes is designed to deliver 75 cu. ft. of free air per minute at a pressure of 120lb. per sq. in. This will permit the use of one pneumatic tool, taking 50 cu. ft. of air per minute, and supplying two divers at a depth of 100 feet, or one diver at a depth of 220 feet. The *Vigilant* is fully equipped with the most up-to-date fire-fighting apparatus, capable of dealing with major outbreaks of fire, including oil fires. The principal items of equipment comprise two Dixon 6-in. monitors, one of which is fitted on the wheelhouse top and the other above the boat deck, aft. Each unit has a capacity of 1,600 gal. per minute at a pressure of 230lb. per sq. in., and is provided with a wide range of nozzles in order to meet varying conditions. In addition, there are fourteen portable rail guns for operation from the bulwarks, or from the platforms on the fore mast. Twenty 100-ft. lengths of 3½-in. diameter, and eighteen 100-ft. lengths of 2½-in. diameter, high pressure canvas delivery hose have also been provided. A powerful electrically-driven fire and salvage pump 10in. by 10in., with a capacity of 440 tons of sea-water per hour, at a pressure of 230lb. per sq. in. is installed in the auxiliary-machinery space. The pump is driven by an electric motor rated to develop 435 b.h.p. at 2,100 r.p.m. Water is impelled *via* a 10-in. steel-tube fire main, which is gradually reduced to 6 inches as the various branches are taken off. Distribution boxes and hose connexion valves are provided throughout the ship. The *Vigilant* is capable of undertaking any type of salvage work, and much careful thought has been exercised in the design and arrangement of the salvage equipment. In addition to the usual arrangement of bollards, fairleads and mooring pipes, provided in connexion with the normal handling of the vessel, a specially designed steel bow casting, capable of taking a load of 100 tons, is arranged forward. The unit operates in conjunction with four 100-ton roller bollards, each of which has an 18-in. diameter roller. An anchorage to take a 100-ton purchase is arranged on each side of the deck, to line up with the bow casting. The deck arrangements are augmented by suitably arranged bollards, fairleads, samson posts, eyeplates and mooring pipes of special design, in order to assist in heavy salvage operations. The *Vigilant* is propelled by twin screws, directly driven by triple-expansion, surface-condensing, open-type, reciprocating steam engines, capable of developing, collectively, a maximum of 1,450 i.h.p. at 198 r.p.m. and 1,300 i.h.p. at 188 r.p.m.—*The Shipbuilder and Marine Engine-Builder, Vol. 60, October 1953; pp. 589-595.*

Arc Welding of Aluminium in Shipbuilding

Practical experience proves that arc welding with coated electrodes is a convenient joining method for aluminium and its alloys, for material in thicknesses from 2 mm. and up. This is the case as far as strength, corrosion, resistance and economy is concerned. The condition for this is that, when choosing base materials, one should consider the weldability. The presence and amount of the different elements is of great importance. When selecting electrodes one should have an open eye to the important factors. Considering the various standard electrodes, the type containing approximately 1·3 per cent Mn seems to be the best with regard to mechanical-corrosion-cracking properties. As for the porosity, which may be a great inconvenience, one should try to find and eliminate the causes. The most dangerous point here is the moisture entering the melting bath. Distortions caused by shrinking stresses may be avoided or reduced by simple precautions. The results of experiments here mentioned are in accordance with practical experience gained, especially in connexion with the shipbuilding industry, this being the field in Scandinavia where arc welding of aluminium has found its most extensive application. The conditions and experiences here may of course be transferred to other industries where aluminium and its alloys are of great importance.—*Paper by G. Aronsen, read at the 5th International Mechanical Engineering Congress, Turin, 1953.*

Distortion and Shrinkage Stresses in Welded and Riveted Hulls

Careful measurements were made on several butt welds and riveted joints during the construction of a cargo vessel (9,300 tons dead weight) and an oil tanker (17,200 tons dead weight). Movable mechanical extensometers and resistance strain gauges were used to measure distortions and shrinkage stresses. Fixed mechanical extensometers and acoustic extensometers were not applicable under shipyard conditions. Shrinkage stresses were estimated by drilling holes at strain gauge rosettes. The measurements showed that shrinkage stresses reach and, in some places, exceed the normal service stresses. The shrinkage stresses, due to welding, were higher than those due to riveting. Near welded joints, inelastic deformation reaching 5 per cent at a joggled joint was observed. In several locations near butt and fillet welds, biaxial tension was observed. On the contrary, in riveted joints, the shrinkage stresses are compressive, which explains the value of riveted crack arresters. —*R. Spronck and J. J. L. van Maanen, The Welding Journal, Vol. 32, September 1953; p. 474-s.*

Effect of Heat Treatment on Transition Temperature

In a previous investigation, it had been found that the steels A and C considered by the American Ship Structure Committee exhibited a zone of minimum ductility immediately adjacent to the weld area in a region which was not heated above the lower critical temperature (A_1) at any time. The occurrence of the region of highest transition temperature outside the so-called heat-affected region has been observed by other investigators. Although no weld failures have been found to originate in this embrittled zone, it may play an important role in the brittle failure of welded structures by allowing a crack, once it has been initiated, to propagate at a high rate with the absorption of only a small amount of energy. In this investigation the embrittling characteristics of base plate of a ship plate steel (C steel) were studied under more closely controlled subcritical thermal cycles than can possibly exist in weldments. Isothermal heat treatment in the 700-1,200 deg. F. range resulted in little or no embrittlement when employing an air or furnace cool; water quenching resulted in a significant decrease in ductility. Transition temperature-isothermal time curves indicated that time and temperature had little effect on the ductility. Room temperature ageing studies of quenched specimens revealed that this steel was susceptible to embrittlement by the quench-ageing mechanism. The degree of embrittlement increased with an increase in solution temperature from 650 to 1,300 deg. F. Characteristic quench-ageing curves were obtained after quenching from 1,300 deg. F. and employing ageing temperatures up to 350 deg. F., i.e. the peak embrittlement reached and the time to attain this peak decreased with increasing ageing temperature. Specimens aged above 250 deg. F. "overaged" so rapidly that no peak in the ageing curve was detected. Metallographic examination of quench-aged specimens indicated that a two-stage precipitation reaction was operative. This study suggests that a low-temperature post-heat at about 650 deg. F. would lead to rapid recovery of ductility, by overageing, in the subcritically embrittled zone previously found in ship plate weldments.—*E. B. Evans and L. J. Klingler, The Welding Journal, Vol. 32, September 1953; pp. 417-s-431-s.*

Welding Cr-Mo Pipe

The arc welding of 1·25 per cent Mo alloy steel pipe, using 1·0 to 1·25 per cent Cr—0·5 per cent Mo and 2·25 per cent Cr—1·0 per cent Mo low-hydrogen alloy electrodes, has been investigated. It has been determined that a significant correlation exists between hardness and ductility and between hardness and tensile strength, for heat-treated 1·25 per cent Cr—0·5 per cent Mo alloy steel pipe material and for 1·0 to 1·25 Cr—0·5 per cent Mo alloy weld deposits. On these bases the properties of the various zones in the welds have been more realistically evaluated. Similarly, longitudinal bend tests have provided confirming evidence for weld metal and heat-affected zone ductilities which are not apparent from results of the usual reduced section tensile tests of butt-welded specimens. The results of this investigation indicate that satisfactory static and impact physical properties are obtained for unrestrained welds made in 1·25 per cent Cr—0·5 per cent Mo alloy steel pipe using 1·0 to 1·25 per cent Cr—0·5 per cent Mo low-hydrogen alloy electrodes, welded without preheat or postheat treatment. For welding under conditions of restraint, and in the absence of data reflecting restraint v. preheat, it is suggested that a nominal preheat of 300 deg. F. be used in order to minimize the possibility of cracking of root passes.—*J. Bland, The Welding Journal, Vol. 32, September 1953; pp. 803-814.*

Air Conditioning in Submarines

An article by H. S. Dewey in Heating, Piping and Air Conditioning, describes a heat transfer system applied to submarines where unwanted excessive heat is extracted from compartments with constant heat sources and transferred to spaces requiring heat instead of being discharged overboard. The air conditioning system includes a compressor taking heat from its source and thus performing cooling or dehumidification duty and raising its level through compression. The heat is transferred to fresh water heaters via refrigerant gas and retained for use inside the submarine. The refrigerant is always routed through the chiller regardless of whether the unit is performing heating, cooling, or both.—*Bulletin de l'Institut International du Froid, Vol. 33, 1953, No. 3; pp. 540, 542.*

Corrosion in the Moisture Region of Large Steam Turbines

Tests to compare the resistance of various actual and possible steam-turbine materials to corrosion-erosion were run, using wet steam extracted from an operating turbine. To determine the cause of the corrosion, chemical studies were made of the liquid phase, herein called initial condensate, at the temperature and pressure of the dew point in the turbine. It was learned that the pH of the fully condensed steam from the same point and the sulphur compounds from boiler sulphite were a principal cause of the lowered pH with its attendant corrosion. A programme was laid out whereby actual and prospective turbine materials could be compared under corrosion conditions duplicating turbine service as closely as possible. Testing was done in wet steam extracted from a turbine where corrosion was known to exist. A prominent New England utility company permitted the tests to be run in one of their stations. The turbines in this station are 40,000-kW. 3,600 r.p.m., 21 stage, tandem-compound, double-flow turbines, operating with initial conditions of 1,200lb. per sq. in. and

900 deg. F. The dew point occurs in the 14th-stage extraction line through stainless-steel pipes. The test apparatus inlet valve was run wide open and most of the pressure drop occurred through the restriction in the specimens. The inlet pressure to the specimens was, therefore, very near the 14th-stage pressure. A short section on uninsulated pipe supplied steam to each tester from the top of a drained header to ensure that equal amounts of fresh condensate would be supplied to all metal specimens. According to the paper, good pH equipment was used and the chemical determinations were made by the methods of microchemistry. A statistical study of the relationship of various boiler-water treatments to corrosion in representative power stations is being carried on. Plans are being made for additional chemical studies in other power stations.—*Paper by T. W. Bigger, J. F. Quinlan, and C. C. Carson, given before the 1953 A.S.M.E. Semi-Annual Meeting; Paper No. 53-SA-12.*

Steering Gears

The steam steering engine with its long steam pipes to and from the engine room is now fast becoming a thing of the past. Today the hydraulic-ram type of steering gear has become standard in most naval and merchant vessels. This type is unlikely to be further developed since little, if any, saving in weight results from the use of higher pressures. An improvement could be made by the use of servo-controls. The present trend in control systems is to reduce the effort at the wheel. The hydraulic telemotor has predominated for the last fifty years, but it is now giving place to hand-power electrical synchronous systems. In merchant ships, the gyro-pilot, a true servo-mechanism, is becoming popular. It is now possible to produce a ball-bearing screw with an efficiency of over 90 per cent. A steering-gear design could be based on such a screw using a single one-handed screw with a single nut. The end or ends of the screw would be connected to the tiller or tillers by motion gear. The nut would be rotated by a hydraulic motor through gearing, suitable duplication of drive to ensure safety being arranged. Hand emergency steering would be easy but it would be necessary to ensure that the high-efficiency screw was prevented from over-hauling. Power to the hydraulic motors would be provided from two variable-delivery pumps controlled from the wheelhouse through an electrical system and servo-motors. A steering gear on the lines described would weigh no more than half the weight of the equivalent hydraulic-ram type, and would occupy considerably less space. It would cost more, however. A new design of hydraulic gear could be developed in which a cylinder of fairly large diameter is fitted concentric with the rudder head. The tiller arms, shaped like vanes, and two diaphragms would divide the interior of the cylinder into four pressure-tight compartments. By closing the cylinder top and bottom, and admitting pressure to two diametrically opposite compartments and opening the other two to exhaust, it would be possible to move the rudder as required. This may be described as the "vane type" of steering gear fitted in some foreign vessels. Sealing the pressure chambers would be a difficulty, but the advent of new seals should allow this problem to be solved. In order to reduce the size of such a gear, high pressures would be necessary. The possibility of using differential linkage connexion in steering gears to increase the leverage at large rudder angles is discussed.—*Paper by A. M. Butterfield, read before the New England section of the Society of Naval Architects and Marine Engineers, June 1953; Journal, The British Shipbuilding Research Association, September 1953, Vol. 8, Abtract No. 7,933.*

Ship Model Correlation and Tank Wall-effect

The paper first emphasizes the insufficiently appreciated fact that model tests in shallow water tanks give very exaggerated resistances compared with ship results in water of the same relative depth but of unrestricted width. A re-presentation of Comstock and Hancock's 1942 work clarifies the extent of the exaggeration and suggests a new principle in model testing technique. This states that for constant speed-length ratio, specific resistance is increased linearly with blockage, where blockage is measured by the ratio of model cross-sectional area to tank net cross-sectional area. The use of this principle eliminates previous errors in shallow-water prediction work. The problem is shown to appear in analogous form in the analysis of a "geosim" series, since all constituent resistance observations require to be corrected for blockage before the series can be extrapolated to the ship. Accepting the linearity of the blockage resistance differentials it becomes statistically possible to determine their numerical value at each speed-length ratio. These determine the real geosim extrapolator freed from the wall-effect which distorts the apparent extrapolator and prevents extrapolation to the ship. The approach is shown to be particularly useful when the same series is tested in two tanks differing only in width. This practice is also recommended for commercial testing with a single model. The application of the methods developed to the recently published geosim tests and ship trials of the *Lucy Ashton* is illustrated in full detail; and it is significant that the derived real extrapolator agrees exactly with that previously found from fully turbulent planks, pontoons and pipes of the same length-girth ratio as the *Lucy Ashton*. The ship model correlation shows the paint roughness to take the expected viscous form, but unexpectedly a further residual difference between model and ship is shown to exist. This is either a wave-making scale-effect or a ship shallow-water effect and is a challenge to the Froude principle. The paper concludes with a review of the possibilities of other geosim series. Further considerations of application detail are discussed.—*Paper by Professor E. V. Telfer, read at a meeting of the North-East Coast Institution of Engineers and Shipbuilders, 16th October 1953.*

Superheat Temperature Affects Plant Design

The possible increase of operating temperature, which is one of the best methods of increasing the efficiency of steam-propulsion plant, is limited by the reduction in allowable stresses in the materials used that occurs as the temperature rises. For example, the hoop stress of chromium-molybdenum pipes at 1,000 deg. F. is reduced to 58·5 per cent of its value at 900 deg. F.; whereas with carbon-molybdenum pipes the reduction is only 7 per cent between 800 deg. F. and 900 deg. F. Allowance must be made for the components to withstand short operational periods at 25 deg. F. in excess of design temperature with very short periods in excess of that margin. Superheaters primarily of convection design, but placed so as to receive a portion of heat by radiation from the furnace, have been installed in destroyers, with a maximum design temperature of 750 deg. F. A similar arrangement operating at 850 deg. F. is being used in new naval and merchant ships, whilst a few go up to 950 deg. F. Control of the superheated steam temperature is essential and should (a) prevent the temperature from exceeding the maximum at all times, except for short periods which would cause negligible increases in metal temperature; (b) maintain the superheat temperature at the allowable maximum over a wide range of load, to maintain cycle efficiency and to prevent excessive moisture in the last turbine stages; (c) have adequate range and speed to maintain this constant superheat temperature under fluctuating conditions of feedwater temperature, excess combustion air, and quantities of saturated or cooled steam flow to auxiliaries. For highly-rated naval boilers, three methods of approaching the subject are indicated and illustrated. The first comprises a two-furnace three-drum single-uptake superheat-control boiler, with automatic combustion control for both pressure and temperature. This is heavier and more complicated than the plant used in the second method —a single-furnace boiler with spray cooling for control of the superheat. Finally, there is a single-furnace boiler without superheat control but with pipes and turbines designed for the maximum temperature anticipated under adverse operating conditions of excess air, reduced feed temperature, and rapid changes in load.—*G. M. Boatright, Bur.Ships.J., Vol. 2, 1953; p. 19. Journal, The British Shipbuilding Research Association, September 1953, Vol. 8, Abstract No. 7,986.*

Experimental Freezing Trawler

The purposes behind the freezing-fish-at-sea project now under way at the Boston Technological Laboratory of the United States of America Fish and Wild Life Service's branch of Commercial Fisheries are:—1. The development of handling freezing and storage facilities which can be installed and used successfully on existing New England vessels. 2. The investigation of the technical and economic feasibility of freezing fish "in the round" at sea for later processing, that is to say, thawing, filleting and refreezing ashore. Emphasis has been placed on methods easily adaptable to present New England vessels, in preference to comprehensive systems requiring extensive conversions. The *Delaware* is a typical present-day large New England trawler with the following particulars:—

Length o.a.	147ft. 6in.
Beam	25ft. 0in.
Depth	14ft. 8in.
Deadweight tonnage ...	544 tons
Cruising range... ...	8,000 nautical miles
Speed	10 knots approximately
Crew	20 persons

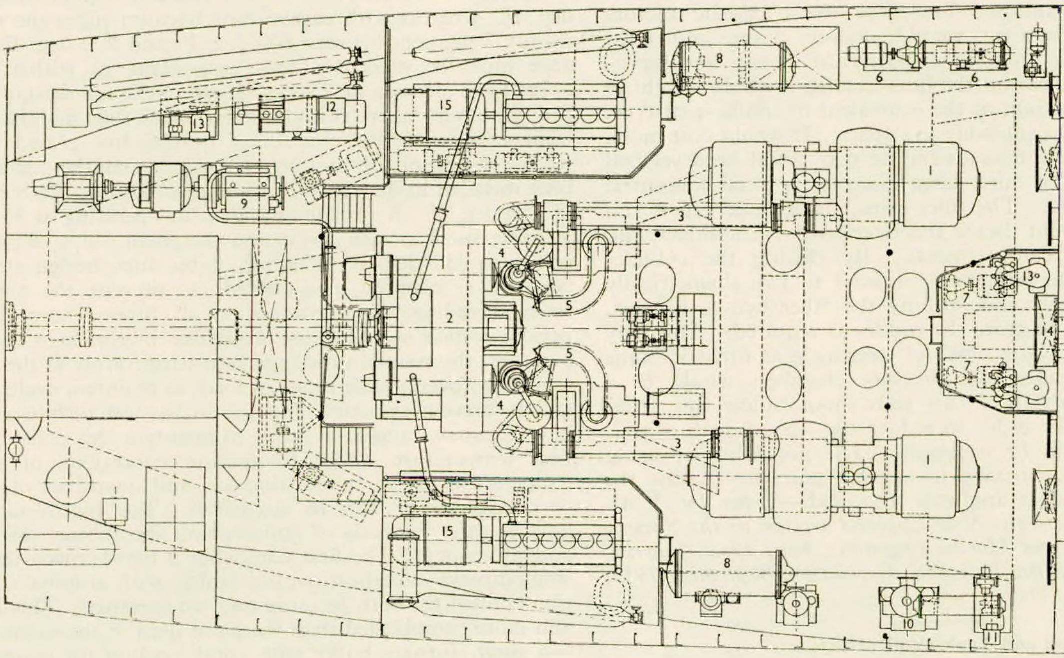
She is propelled by a 7-cylinder 2-cycle engine of 735 h.p. at 300 r.p.m. The two units supplying the 115 v. D.C. electrical system are a 40-kW. Diesel generator and a 25-kW. Diesel generator standby set. Trawl winch power is provided by one 80-kW. Diesel generator set connected to the 100 h.p. electric winch motor. She has a fresh water capacity of 11·9 tons, fuel oil capacity of 63·2 tons and a lubricating oil tank capacity of 400 gallons. Since delivery to the East Boston laboratory the vessel has been altered mainly in the fish hold and equipment added to fit her out for experimental freezing and storing of the round fish. She had originally a fish hold of about 8,000 cu. ft. capacity, the 36½-ft. length of which had been divided into seven sections. The new partitioning arrangement by cork insulated bulkheads are as follows:—One pen-section right forward for storing iced gutted fish, two pen-sections insulated and refrigerated for storing round frozen fish, three pen-sections insulated and refrigerated for storing round frozen fish and housing the fish freezing tank, and one pen right aft for housing the refrigerating machinery. The *Delaware* was originally insulated with 4 inches of cork laid in hot asphalt and sheathed with 1½ inches wood sheeting. The refrigeration equipment now on the *Delaware* consists of:—1. Brine tank and fish freezing mechanism. This method of freezing the round fish by immersion in refrigerated brine was chosen over other methods because it appears to offer such advantages as economy of space and manpower, short freezing time, and simplicity of machinery. 2. Hold cooling equipment. The frozen fish storage holds, wherein the design temperature is 0 deg. F. (− 17·8 deg. C.), are refrigerated by banks of 1¼-in. (3·2 cm.) iron pipe coils mounted on bulkheads, deckhead, and port and starboard surfaces. The refrigerant is a solution of ethanol and water. 3. Refrigeration plant. A 25-ton ammonia absorption refrigeration machine is used for the freezing and cold storage of the round fish.—*Shipbuilding and Shipping Record, 24th September 1953, Vol. 82; pp. 425-426.*

Free-piston Engined Cargo Ship

In view of the increasing interest which is being shown throughout the world in the free-piston engine, both for industrial and marine purposes, the completion of the first cargo vessel to be equipped with such machinery is of importance. The ship in question, the *Cantenac*, is one of two coasters ordered from the Ch. et At. Augustin Normand, at Havre, by Worms et Cie, and the general arrangement and engine room plans are now given, also illustrations of the machinery under test in the manufacturers' works. Several similar engines have been fitted in French minesweepers but no installation has been made in a merchant ship. The following are the main particulars of the ship:—

Length b.p.	59·4 m. (185ft. 0in.)
Moulded beam ...	9·3 m. (30ft. 6in.)
Moulded depth ...	5·6 m. (14ft. 5in.)
Mean draught ...	4·113 m. (13ft. 6in.)
Machinery	1,200 b.h.p.

The vessel has three holds and the machinery is aft. MacGregor



Engine room arrangements of the Cantenac

1.—Free piston gas generators. 2.—Geared gas turbine. 3.—Exhaust pipe between generators and turbine. 4.—Servo motors and valve for reversing machinery. 5.—Pipes between reversing valves and inlets (astern running). 6.—25-ton fresh water circulating pumps. 7.—60-ton sea water cooling pumps. 8.—Fresh water coolers. 9.—Emergency generating unit. 10.—60-ton ballast pump. 11.—Fuel transfer pumps. 12.—Air compressors. 13.—Fuel oil purifiers. 14.—Fuel oil heaters. 15.—80-kW. generators.

steel hatch covers are employed. In a free-piston engine there is a horizonal cylinder in which are two free pistons, one of which may be termed the combustion piston and the other the compressor piston. The latter supplies the air to the combustion cylinders for scavenging and supercharging, and the mixture of excess air and products of combustion passes out through exhaust ports to a gas collector and thence to a gas turbine (there are two in the *Cantenac*) which drives the propeller shaft. The diameter of the combustion cylinder is 340 mm. and the total output of the two units and the turbine is 1,200 b.h.p. It is to be recorded, however, that the plant is capable of developing 1,800 b.h.p. but the power is limited owing to the characteristics of the hull, as it was intended in the first place to have a smaller engine. The two gas turbines run at 9,000 r.p.m. and can operate independently if desired. Each turbine has three stages ahead and one astern, the maximum power astern being about 800 b.h.p. The double reduction gear reduces the speed to 220 r.p.m. The free-piston generators are started on Diesel fuel but operate on light fuel oil. The weight of the complete engine, including the piping, is 71·5 tons, which is equivalent to 89lb. per b.h.p. at the rating of 1,800 b.h.p.—*The Motor Ship, November 1953, Vol. 34; pp. 322-323.*

Synthesis of Two Marine Watertube Boilers

In this paper a brief résumé is given of the important lessons learnt from the operation of oil-fired watertube boilers at sea since 1925. These are linked with the shipowners' requirements—particularly those pertaining to the post-war merchant service—and the development of two designs of boiler from these requirements is shown. The first design is the main boiler fitted in the s.s. *Nestor,* and the second is the auxiliary boiler fitted in the same ship and in motorships building at the same time. The main boiler is designed for a normal output of 25,000lb. per hr. and a maximum output of 30,000lb. per hr., the feedwater inlet temperature being 240-230 deg. F. and the boiler efficiency to be 88·5 per cent at overload. The specified steam conditions at the superheater outlet are 625lb. per sq. in. pressure and 950-750 deg. F. steam temperature. It

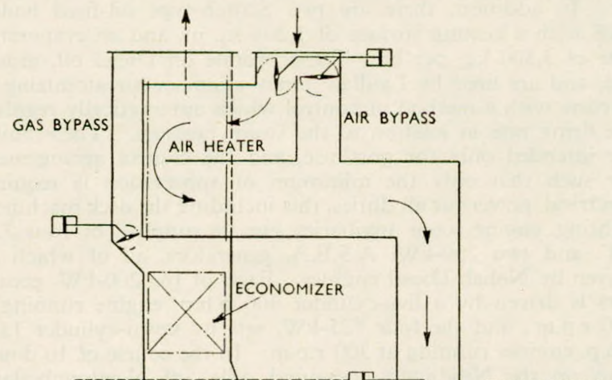


FIG. 26—*Automatic steam damper control system*

was considered desirable to ensure that creep phenomena could not adversely affect the safe operative life. It was, therefore, essential to provide a design of temperature control such that the steam temperature would be limited automatically to a maximum of 950 deg. F., any necessary tolerance being taken as negative. In addition, to provide as much safety as possible, the high temperature end of the superheater should also be arranged so that unbalanced gas or steam flow could not occur. As a result of war-time experiences with slagged superheaters, it was specified that the superheaters should be designed so that they could be washed with fresh water with minimum interference and the least possible risk of damage to the furnace refractory. Similarly, as a result of the very satisfactory experiences with superheaters of the MeLeSco type in Admiralty boilers, it was decided to use the Concen tube joint for up to a maximum temperature of 850 deg. F., which is approximately the top limit for this type of fitting. The design produced by the Superheater Company was the neatest for this section of the superheater, as intermediate headers were eliminated between the first two sections. The turbine manufacturers required the steam to the astern turbines not to exceed 750 deg. F. at the maximum boiler output, so that the superheater was to provide for means of control over a narrow range of temperatures at 950 deg. F., and of reducing the temperature to 750 deg. F., and it had to be physically remote from the furnace. To prevent overheating by mal-operation, it was considered essential to have all the fuel fired into one furnace, as there is always a risk of the two-furnace design of boiler being incorrectly fired. With one furnace, a new method of superheater control had to be devised and, by integrating this requirement with the physical separation of the superheater and the furnace, it was concluded that damper control provided the best solution. Examination of the details showed that this decision resulted in a reduction in weight. Similarly, the physical separation of the two components was found to have the merit of allowing the superheater to be designed as a heat exchanger without regard for the geometrical limitations imposed upon it by other sections of the boiler. The requirement for good steam distribution meant a long steam path and relatively few paths in parallel, which also suited the geometry of the space limitations in the ship. The long steam path meant a considerable number of "U" tubes or, and this was preferred, a welded design, with the length of elements kept within reasonable proportions for handling by separation into three sections. The maximum temperature under the astern conditions was found not to exceed 800 deg. F. at the outlet of the second section, so that the first two sections were of the MeLeSco design. The final section of the superheater was of all-welded design, to eliminate possible tube end leakage due to fluctuations in temperature. The material for the headers and tubes was 1 per cent chromium - 0·6 per cent molybdenum steel, which could be easily welded. The division of the superheater into three sections also enabled a convenient arrangement of gas dampers to be made, so that the surface of the superheater was greatly reduced for astern operation. This had a secondary advantage, namely, that the high-temperature section of the superheater was out of the gas path when there was greatest risk of the boiler being mishandled during a rapid manœuvre. Finally, it was also simple to arrange for the attemperator to be fitted conveniently into the steam and air circuits. *Adjustment of Steam Temperature.* The attemperator is constructed of finned tube and is damper-controlled. Under normal operating conditions, preheated air is used as the cooling medium, and the damper is operated automatically to adjust the final steam temperature as near as practicable to 950 deg. When, however, a low final temperature is required, the bypass on the air heater is opened and the damper is put in the full-open position, so that cold air is used to reduce the temperature of the steam from the second section of the superheater as far as is practicable. The small amount of heating done in the final section, which is now bypassed, raises the temperature to a maximum of only 750 deg. F. These damper operations are carried out by a system of remote control, and can be controlled by a single lever on the *engine* manœuvring platform. In this way, the control of steam temperature essential for the operation of the turbines is under the control of the turbine operator, instead of being remote from him as has hitherto been marine practice.—*Paper by Com'r(E) L. Baker, D.S.C., R.N.(ret.), read at a General Meeting of The Institution of Mechanical Engineers, 13th November 1953.*

Water-lubricated Thrust Bearings

Water-lubricated carbon thrust bearings, with the correct amount of circumferential and radial taper, can carry loads of at least 110lb. per sq. in. successfully at speeds from 860 to 2,450 r.p.m. Radial tapers perform acceptably at high rotative speeds. At low speeds they may reduce the ultimate load capacity of the bearing. Circumferential tapers of extremely slight angle appear to be associated with the greatest friction. Moderate tapers have low friction loss and are most dependable at low speeds and high loads. From the data on hand, a good all-round design for a water-lubricated 2¾-in. outside diameter, 1¾-in. inside diameter fixed-land, carbon thrust bearing would incorporate: (a) Six lubricant-distributing grooves. (b) Rounded or bevelled leading edges on the six lands. (c) Circumferential taper in ⅔ or ¾ of the land area, 0·0006 to 0·0008 inch deep at the lowest point. (d) Radial taper of zero to 0·0003 inch per inch sloping down toward the inner diameter. (e) Untapered portion of the lands flat and smooth. This research was carried out at the U.S. Naval Engineering Experiment Station, Annapolis, Md.—*M. Levinshon and N. E. Reynolds, Trans. A.S.M.E., August 1953, Vol. 75; pp. 1137-1145.*

Refrigerated Cargo Ship

The single-screw motorship *Penja,* a refrigerated cargo vessel designed for the carriage of bananas, was recently delivered to her owners, L. Martin, S.A.R.L., Paris, by Oresundsvarvet A/B Landskrona. Unusual features include increased thickness shell plating in the way of insulation, the fitting of air-conditioning throughout the accommodation spaces and a general arrangement which gives three cargo holds forward and one aft. Although the contracted speed was 16 knots, a mean speed of 18·28 knots was attained. There are two continuous steel decks, a forecastle deck extended into the boat deck aft to the mainmast and an additional fruit deck in way of the forward holds. The engine room is situated about one-third of the length from the after end. While the *Penja* is transversely framed, longitudinal frames are incorporated in the midship portion of the double bottom. In view of the very high cost of repairs to an insulated shell, the shell plating has been made extra heavy with a minimum thickness of 18 mm. The hull is mainly welded but the frames are riveted. The deckhouse on the navigating bridge is of light alloy. The leading particulars of the *Penja* are:—

Length o.a.	362ft. 5½in.
Length b.p.	336ft. 6½in.
Breadth moulded	49ft. 0in.
Depth moulded to upper deck	28ft. 6in.
Draught	23ft. 1in.
Net tonnage	2,077 tons
Gross tonnage	3,805 metric tons
Contracted speed on 18ft. mean draught corresponding to banana cargo and with 70 per cent engine output	16 knots

The four hatch covers on the weather deck are of the builders' own design, while the light alloy and insulated hatch covers in the 'tween decks are of McGregor-Comarain's design and manufacture. All cargo spaces are completely insulated with granulated cork and they are divided into seven sections with separate refrigerating systems. Temperatures of 33 deg. F. may be maintained throughout the cargo spaces under tropical conditions. At the forward end of No. 4 'tween deck is a small meat room on the port side which may be cooled down to 14 deg. F. The propelling machinery consists of an 8-cylinder direct reversible two-stroke cycle single-acting Götaverken engine. The cylinder bore is 630 mm. and the stroke 1,300 mm. At 105 r.p.m. the engine is rated at 5,000 b.h.p. (metric). Three Götaverken Diesel generators, each consisting of a 3-cylinder, four-stroke cycle oil engine of 300 h.p. direct coupled to a 200-kW. generator, supply current to the refrigerating plant, deck machinery, engine room auxiliaries and lighting.—*Shipbuilding and Shipping Record, 1st October 1953, Vol. 82; pp. 455-456.*

New Swedish Passenger Liner

The new 22,000-ton 19-knot transatlantic liner *Kungsholm,* of the Swedish-America Line, was built by the De Schelde yard at Flushing, Holland. The vessel is of striking appearance, with restrained streamlining, well raked and flared bows and a cruiser stern. There are two funnels, and the masts are arranged so that they can be lowered to permit navigation under bridges in the Kiel Canal and elsewhere. The principal characteristics of the ship are as follows:—

Length o.a.	600ft. 0in.
Length b.p.	530ft. 2½in.
Moulded breadth	77ft. 0in.
Depth to upper deck ...	57ft. 2in.
Depth to main deck ...	48ft. 8in.
Draught on summer freeboard	26ft. 4in.
Displacement on summer freeboard	18,335 tons

The hull is subdivided into eleven vertical sections by means of ten watertight bulkheads, which have a total of thirty-three watertight doors. The two Burmeister and Wain eight-cylinder engines together have a normal output of 17,500 i.h.p. at 115 r.p.m., the cylinder bore being 740 mm. and the piston stroke 1,600 mm. These are of the two-stroke, single-acting type, of standard design and construction. The exhaust gases are utilized in two Schelde-Lamont boilers, each with a heating surface of 266 sq. m. and an evaporation rate of 3,300 kg. per hr. In addition, there are two Scotch-type oil-fired boilers, each with a heating surface of 1,208 sq. m. and an evaporation rate of 3,500 kg. per hr. These operate on Diesel oil, gravity fed, and are fired by Laidlaw Drew steam or air atomizing oil burners with a method of control which automatically regulates the firing rate in relation to the steam pressure. These boilers are intended only for port use, and the control arrangements are such that only the minimum of supervision is required. Electrical power for all duties, this including the deck machinery, lighting, engine room auxiliaries, etc., is supplied by four 725-kW. and two 200-kW. A.S.E.A. generators, all of which are driven by Nohab Diesel engines. Each of the 200-kW. generators is driven by a five-cylinder 300 b.h.p. engine running at 330 r.p.m., and the four 725-kW. sets by seven-cylinder 1,080 b.h.p. engines running at 300 r.p.m. In the course of 10 double runs on the Newbiggin measured mile, off Northumberland, the highest average speed recorded was 21·2 knots at almost full draught; the contract speed was 19 knots. The corresponding r.p.m. were 125 and the machinery output 22,800 i.h.p.—*The Motor Ship, November 1953, Vol. 34; pp. 327-333.*

Engines for Fishing Industry

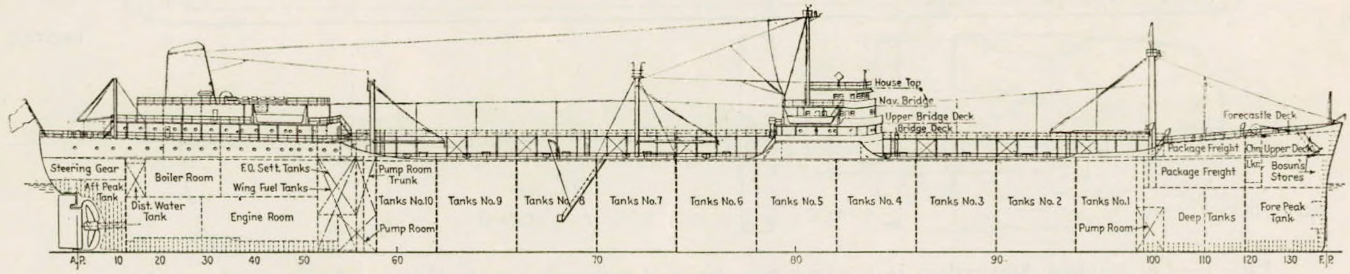
A noticeable feature of the many papers presented at the International Fishing Boat Congress, 1953, in Paris and Miami, is the greater attention being paid by owners and builders of fishing vessels to the advantages of marine Diesel propulsion, and the conflicting claims of high- and medium-speed engine builders. Mr. D. E. Brownlow, in his paper, "The Use of Medium-speed Diesel Engine on Board Fishing Vessels", refers to the variety of engines from which a choice can be made, e.g. four-stroke normally-aspirated, four-stroke turbo-charged, two-stroke engine, and these may be low-speed, medium-speed or high-speed engines. To meet the requirements of the fishing industry, the Mirrlees KS-type engine was designed and developed for speeds from 200 to 450 r.p.m. Up to the present time, the demand and preference for a 120-ft. vessel of 323 gross and 115 net, and with a fish hold capacity of 7,000 cu. ft., has been largely for engines running at speeds of 230 to 300 r.p.m., directly coupled to the propeller shafting. When the engines are required to operate at higher speeds and powers, it is necessary to fit a reduction gearbox between the engine and propeller, in order to maintain propeller efficiency. The Mirrlees KS engine is a four-stroke turbo-charged engine having a bore of 15 inches (381 mm.) and a stroke of 18 inches (457 mm.) and built with six, seven and eight cylinders. It is direct-reversing, or may be unidirectional when coupled to a reverse-reduction gearbox, for the higher engine speed of 300 to 450 r.p.m. These engines are designed and built as pressure-charged units. A Büchi-type of pressure charger is incorporated, without increasing the length of the engine. The advantages in a turbo-charged engine for fishing duty are:—1. The horsepower is increased by 50 per cent without any increase in the size of the engine, and recent developments in turbo-charging are making available increased powers up to 100 per cent. Therefore, the space taken for a given power is used economically, thus allowing a maximum space for the catch. 2. No power is taken from the engine to drive the turbo-charger, which is driven by the exhaust gases, thus eliminating gears or chain drives and couplings. 3. Since there is no fuel required to drive the turbo-charger, the fuel consumption of a modern turbo-charged engine is extremely good. Diesel engine production is classified as being in three stages by William C. Gould in his paper "High-speed Diesels for Use in Fishing Craft", these stages being slow-speed (75-500 r.p.m.) medium-speed (500-1,000 r.p.m.) and high-speed (1,000 r.p.m. and above). It is suggested, however, that too much attention is paid to the convention that piston speeds should not exceed 1,500ft. per min., to ensure lengthier service. Such a limitation disregards the vast development and engineering work on the design, finish and materials utilized in the liners, pistons, and piston rings. Emphasis is laid that piston speeds have little to do with the actual wear rates of liners and piston rings but, despite this, most high-speed engines adhere to piston speeds within the region of 1,500ft. per min. It is pointed out that records show that modern engines running at an average piston speed of 1,750ft. per min, are giving about 32 per cent longer life between servicing than older slower-speed models. In any case, it must be remembered that as smaller cylinder sizes are adopted, so is a shorter stroke, consequently the engine may run considerably faster and still have the same piston speed as a slower running engine with a longer stroke. In many cases, the actual piston speed of high-speed engines has been found to be much less than that of medium- and even slow-speed engines. It is also noted that the present-day high-speed Diesel engines have a frictional horsepower loss, in relation to the maximum b.h.p. rating of the engine, considerably less than the percentage of frictional horsepower loss generally encountered on the medium- and slower-speed engines. It is acknowledged that the high-speed Diesel engines cannot properly burn the lower grade fuels that will operate in the slow-speed and medium-speed Diesels. On the average-sized fishing vessel of 150 feet or less, the use of boiler oil is not practicable. Because of the large use of gas oil throughout the world for all purposes, it is probably the most readily available. While its cost is more than some of the other Diesel fuels, this is offset in high-speed machinery by the reduction in cost of components at overhauls.—*The Motor Ship, November 1953, Vol. 34; pp. 339-340.*

Noise of Ventilating Fans

This paper presents the results of acoustical measurements made on five commercially available ventilating fans. Three vaneaxial fans of different diameters and two centrifugal fans of different sizes were tested. In an attempt to generalize the results of these investigations, an empirical formula was derived that permitted the computation of the overall acoustic power levels expected from fans of known design characteristics. The procedures adopted in using this equipment were arrived at by a process of trial and error, and are felt to offer a simple and efficient approach to the problem of measuring fan noise. Through the co-operation of the Bureau of Ships, Navy Department, three U.S. Navy Standard vaneaxial-type fans were made available for testing, and with the co-operation of the Westinghouse Electric Corporation, Sturtevant Division, two commercial centrifugal-type fans were also tested. To test the various fans to determine their exhaust noise characteristics, each fan was run at about twelve different speeds, at an incremental speed of 200 r.p.m. for the highest speed fans, and 50 or 100 r.p.m. for the slower speed fans. Since most of the fans tested were driven by D.C. motors, the various speeds were regulated by varying the armature current on the smaller fans, and the field current and field voltage on the larger fans. The lowest practicable speed of each fan was first determined, and the readings begun on the next highest multiple of the speed increment chosen. The speeds were measured accurately by means of a strobotac properly calibrated over the desired frequency range. First, the overall level of the exhaust noise at each speed was determined. Then readings were taken successively through the twenty-four one-third octave frequency bands. At the end of each series, the overall level was determined again as a check. In addition, at each speed the electrical input power was determined. From a study of the data obtained, the following conclusions were drawn:—1. The general shape of the exhaust noise spectra is contained in the lower frequencies. 2. The majority of the noise in these outlet spectra is contained in the lower frequencies. 3. The fundamental blade passage frequency of a fan does not have as great an effect as expected, and is more apparent in the inlet noise spectra than in the exhaust noise spectra. 4. Changes in the back pressure on the fans make no significant change in the frequency spectra. 5. In duct systems, the duct dimension in the direction of a bend is a major factor in predicting the attenuation which that duct will provide.—*C. F. Peistrup and J. E. Wesler, Journal of the American Society of Naval Engineers, August 1953, Vol. 65; pp. 605-612.*

American Supertankers

The 30,200 deadweight ton *Delaware Sun,* built by Sun Shipbuilding and Dry Dock Company, is the latest and largest vessel to join the Sun Oil Company tanker fleet. On her first voyage, company records were broken when the vessel made the 1,776-mile trip from Aransas Pass to Overfalls Light in



Inboard plan of the Delaware Sun

4 days 8 hours and 54 minutes with an average speed loaded of 17·88 knots. In port, 213,000 barrels of crude oil were unloaded in twelve hours for an average of 18,500 barrels per hour. The principal characteristics of the *Delaware Sun* and sister ships are as follows:—

Length, overall	641 feet
Length, b.p.	615 feet
Breadth moulded	84 feet
Depth moulded	45 feet
Design draught, keel	34 feet
Deadweight, tons	30,200
Cargo oil capacity, 100 per cent full, bbl.	251,114
Fuel capacity, 100 per cent full, tons	1,345
Reserve fuel, forward, 100 per cent full, tons	1,890
Gross tonnage	18,798
Shaft horsepower, normal	13,500
R.P.M., normal	100
Trial speed, knots	16½

The vessel is of the three-island type—poop, set-in bridge, and forecastle; with raked stem incorporating a bulbous bow, cruiser stern and single continuous steel deck. Propulsion is by a single screw with the conventional location aft for propelling machinery, consisting of geared turbines and two watertube boilers. The main cargo pump room is located at the aft end of the cargo oil space. The principal dimensions and characteristics of the vessel were selected to provide a suitable, sea-kindly, easy driving hull form. Self-propelled model tests in load and ballast conditions were conducted at the Taylor Model Basin, Carderock, Md. It has been observed that the supertankers in service have a tendency to plough through the North Atlantic waves rather than rise up to them and, accordingly, the form forward was given a decided flare in order to throw off the green seas. The vessels have flat sheer line, and the forecastle and poop are raised at the end of the vessel to give a sheer effect, thereby contributing to the seaworthiness. To give added protection, solid bulwarks are fitted for 40 feet aft of the forecastle. To provide adequate longitudinal strength, a length/depth ratio of 13·7 was selected. The vessel is welded throughout, with the following exceptions riveted: Two seams of bottom shell, upper and lower seams of bilge strake, lower edge of sheer strake, seam of stringer strake and two seams of deck plating. In addition, the outboard edge of the deck stringer is welded to a 9½-in. by 1¾-in. gunwale plate which is double riveted to the sheer strake. The vessel has a high degree of fire protection. A CO_2 system is installed for the machinery spaces and cargo pump room, and a flue-gas system supplies inert gas for the blanketing of all cargo tanks. *Cargo Tank-Flue Gas System.* The flue-gas system for the cargo tanks is installed in the upper level of the engine room and consists of a "Sutorbilt" type positive rotary, heavy-duty type blower driven by a 35-h.p. 900 r.p.m., 4-speed motor and is capable of delivering 3,000 cu. ft. of hot damp flue gas per min., at a discharge pressure of 2lb. per sq. in. The flue gas is drawn from each boiler uptake through a water-cooled tray-type scrubber and cooler; the scrubber is capable of cooling the flue gases from 400 deg. F. to 100 deg. F. using sea water as the cooling medium. The inside of the scrubber is neoprene-coated. —*Marine Engineering, September 1953, Vol. 58; pp. 50-59; 64-66.*

Special Self-unloading Timber Ships

This paper discusses the characteristics of a self-unloading timber ship of high capacity of novel design, which was evolved in the Netherlands twenty-two years ago and subsequently put into practice in a modified form in three ships built for the Soviet Union. Some prototypes are also compared. The author's knowledge of what may be called the "Amsterdam" ship began in 1932 as a result of a conversation with Mr. Piet Goedkoop of the Netherlands Dock and Shipbuilding Co., Ltd. This was followed by several visits to the *Walerii Meshlauk,* first of the modified ships for Russia. The information so gained has lain dormant in files until reactivated by I.C.H.C.A's timber symposium. The Amsterdam ship was designed originally for the Baltic trade but could, no doubt, be modified for other duties. At the time the original vessel was designed, a large factory in the city was importing timber for the manufacture of wooden doors and window frames for export. Importing the raw material in large quantity demanded maximum efficiency in unloading. Existing lumber carriers were not considered so perfect that improvement could not be effected and the result was the highly mechanized design shown, Diesel propelled and intended specially for the Baltic trade. The ship was a complete breakaway from current three-island ships of *Frederikstad* and *Ornefjell* type with which Northern Europe ports are familiar. It endeavoured to carry mechanical hand-

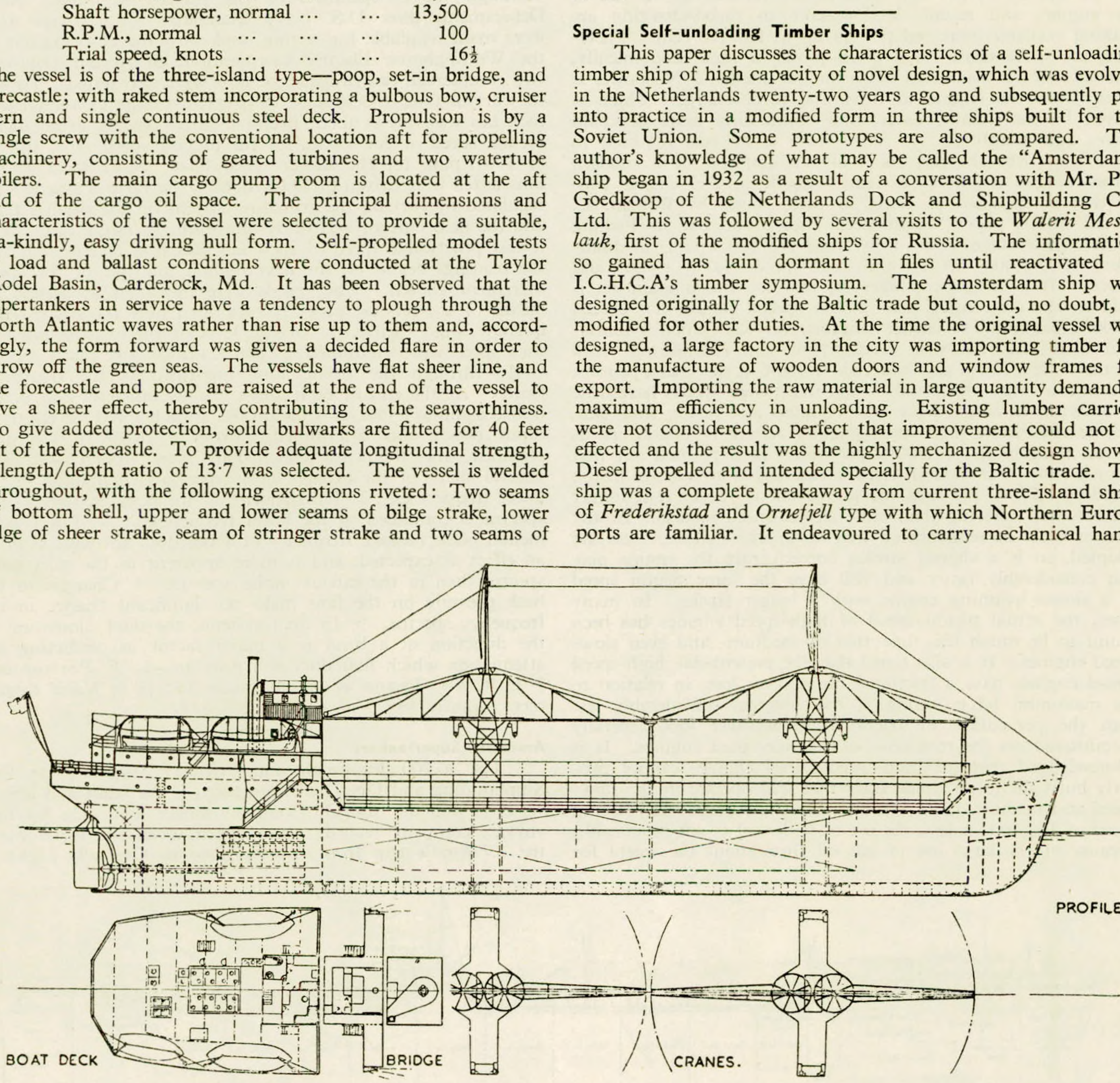


FIG. 1—*The original* Amsterdam *ship which was designed primarily for the Baltic trade. Note the special arrangement of cranes*

ling to its most logical conclusion and combined this with a carefully thought-out hull form, hold shape, and a most complete use of electricity, including distant control of the cargo handling gear. Gear for handling the timber standards is mounted on lattice structures and the "cranes" rotate in a horizontal plane. The cranes themselves are of a gantry type, with the block and cargo hook so arranged that whatever the angle of the gantry to the fore and aft centre line of the ship, the hook can traverse the length of the crane arm from the outer end to the inner end and plumb where required. Fig. 1 shows some details of the cargo working gear, comprising the four cranes working in a horizontal radius from two "bridges" built up on a lattice structure. These cranes are operated from electric winches, the controls being at the ends of the two bridges. The cranes are well clear of the topmost standards of wood and loading and unloading can be carried out under the most unfavourable shore conditions. There is one hold bounded by vertical bulkheads at the sides and ends, and as far as practicable devoid of any internal structure. The hatch opening extends the full width of the cargo hold and its full length, with the exception of the necessary divisions for strength. The large box-shaped hold combined with an equal sized hatch is acceptable for rapid loading and discharge of bulk cargoes which the vessel may be called upon to carry when not carrying timber. The hatch covers were of Macanking (now known as the MacGregor hatch cover) steel type which could be opened or closed in about thirty minutes and which could hinge upwards against the sides of the deep bulwark. The latter is equivalent in height to the poop and fo'c'sle. Those responsible for the Amsterdam ship claimed that, because of its special form and the favourable ratio of standards of timber per ton deadweight, the ship could carry more cargo than its contemporaries. As the net tonnage was low, harbour dues were low also. When in ballast, good immersion of the single screw was possible, because of the large capacity of tanks in the double bottoms, and at the sides of the holds. A small diameter propeller was fitted, working well below the ballast load line, thus assuring a good speed in ballast condition. A speed of ten knots is given by a single 900 h.p., 2-cycle oil engine, driving the screw through an hydraulic coupling; the reason for this was to permit reasonable engine power and revolutions to be maintained at slow propeller revolutions when the ship was navigating in ice conditions. In view of the possibilities existing today for the increased mechanization of cargo handling, it would be interesting to know why greater success has not attended the building, in numbers, of ships of the original Amsterdam type. Each port presents a problem of its own for timber working, but it is suggested that the Amsterdam type with the full gantry arrangement appears to solve the problem of rapid discharge in a unique manner. It would also be interesting to know why the tendency has persisted for so many years to employ three-island ships with deep bulwarks, machinery amidships and derricks and winches on poop, bridge and fo'c'sle respectively clear of the wells. Do such ships possess exclusive virtues? Are they so handy for the carriage of bulk cargoes and particularly those now offering, when not definitely engaged in the timber trade? The Amsterdam ship appears to possess virtues for the carriage of timber and to be reasonably adaptable for handling the bulk cargoes of today and tomorrow.—*A. C. Hardy, Cargo Handling, September 1953, Vol. 1; pp. 134-137.*

The New M.A.N. Engine

The new highly supercharged M.A.N. engine, Type KV, has aroused worldwide attention and deservedly so. It was shown for the first time, and attested by such recognized authorities as Professors Eichelberg and Pflaum that an internal combustion engine can produce power at the unbelievably low fuel consumption of 0·31lb. per b.h.p. per hr., corresponding to a thermal efficiency of 45 per cent, at 231lb. per sq. in. continuous b.m.e.p. Comparing the new M.A.N. engine with another marine engine of similar power output, it is found that the M.A.N. engine weighs 31·5lb. per. h.p., and it puts out 2·03 h.p. per cu. ft. space compared to G.M's 2·2 h.p. per cu. ft. The comparison with smaller cylinder displacement engines is still less favourable which, however, is understandable because the specific output (h.p. per cu. in. displacement) approximately is in inverse ratio to the bore. The new M.A.N. engine is an experimental rather than a production engine. It represents a remarkable accomplishment as it shows the way to very high b.m.e.p's and very low fuel consumptions. With equally refined design and grade of workmanship, similar performance can, no doubt, be achieved in production engines, but whether such high grade of workmanship is practically obtainable in production has not yet been demonstrated. The design features that are obviously responsible for the outstanding performance are high supercharge, high efficiency turbine and blowers, intercooling and crosshead construction. All of these features are applicable to two-stroke cycle engines and most of it with presently available skill and materials. The power output of

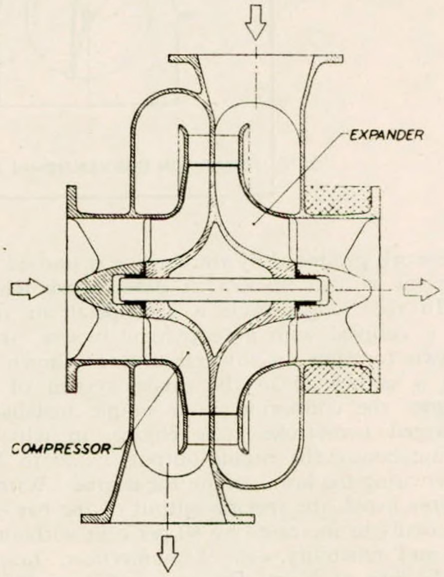


FIG. 1

any listed two-stroke cycle engine could be increased substantially by adopting a more efficient blower, i.e. by improving the blower efficiency from 60 to 70 per cent adiabatic efficiency. In adopting turbo-charging, the practicality of which has been recently demonstrated by the Burmeister and Wain turbo-charged 7,500 h.p. two-stroke cycle engine in the tanker *Dorthe Maersk,* which has completed its shakedown cruise, 35 per cent increase in b.m.e.p. was obtained without reduction in rotative speed. By moderate intercooling, which would reduce the inlet air temperature from 60 deg. F. above ambient to 30 deg. F. above ambient, another 6 per cent could be picked up. A better exhaust system using "aspirator type" exhaust manifolds can easily add another 3 per cent to the power output. By these steps the present 92lb. per sq. in. b.m.e.p. (cont.) of a uniflow engine can be increased to 135lb. per sq. in. and the present 75lb. per sq. in. b.m.e.p. of the loop-scavenged engine to 110lb. per sq. in., without reducing the rotative speed or the engine life. The specific output per pound weight or cubic foot space would increase in almost the same proportion. Therefore, if the two-stroke cycle is ahead now in specific power output it will be still more ahead when these design advancements will be applied. There is, however, another recent development which is more suitable to two-stroke cycle than to the four-stroke cycle turbo-cooling. Turbo-cooling is the name of a cycle in which the engine intake air is cooled by expansion by passing it through an air turbine. It was originated by Mr. H. A. Steiger, now with Sulzer Brothers, Ltd., in Switzerland. Turbo-cooling has already been successfully applied to the cooling of aircraft compartments, where its compactness makes it attractive even

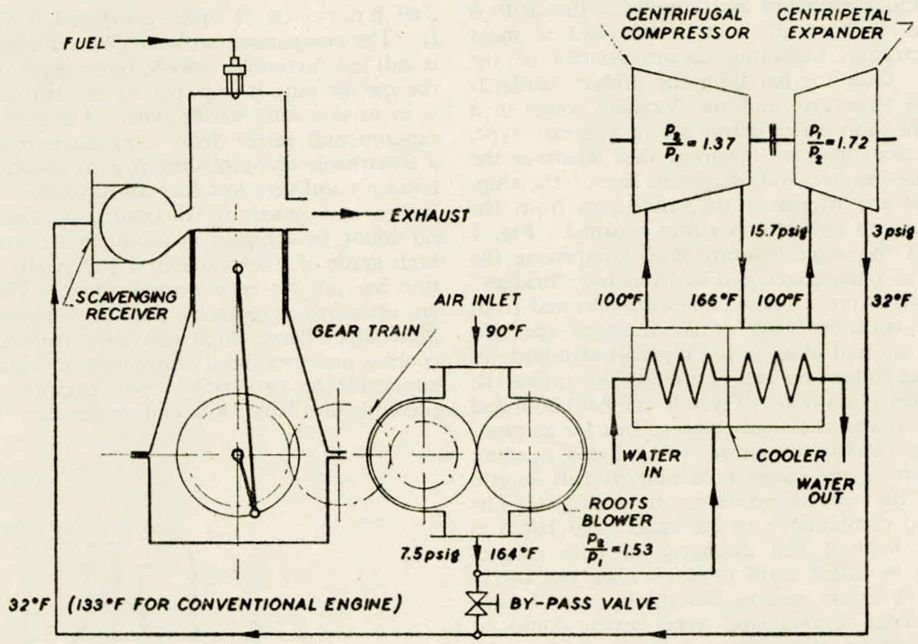


FIG. 2

though the work produced by the turbine is unused. However, the air turbine is also efficient in doing work while it cools the air. In the Steiger cycle a centripetal air turbine (the expander) is coupled with a centrifugal blower (the compressor), the two forming an integral unit as shown in Fig. 1. This unit is connected to the intake system of an engine. Fig. 2 shows the connexion in a simple installation to an unsupercharged two-stroke cycle engine, in which case the turbo-cooling boosts the engine output by 20 to 30 per cent without increasing the heat load on the engine. With the design improvements listed, the specific output of the two-stroke cycle engine can easily be increased by 60 per cent without sacrificing simplicity and reliability.—*P. H. Schweitzer, Journal of the American Society of Naval Engineers, August 1953, Vol. 65; pp. 620-627.*

Steam Trap

This particular invention has a direct reference to steam traps of the type provided with balls or other shaped bodies packed in tubes. As shown in Fig. 5, the steam trap comprises an open-topped container (4) and a removable head (5), an inlet (6) and outlet ports (7) in the head and a needle valve (8) also within the head. A series of packing tubes (9) and pipes (10) are suspended from the head (5) with suitable connexions at the respective upper and lower ends. In use, the steam entering the enclosure housing by the port (6) fills the container (4) and passes through the strainer (16) and up the riser pipe (15) by the openings (17) and eventually through the nozzle of the needle valve (8). From this valve the steam flows down the central pipe (10) to the first of the packing tubes (9) packed with a ball packing. From there it is led by a passage (23) into the successive pipes (10) and packing tubes (9) and finally to the outlet port (7). The steam passing through the successive packings will encounter a reduction in pressure to such an extent that the speed of the flow will be reduced to a minimum, in fact nearly to zero. This will cause condensate accumulated in the steam conduit to fill the container (4) until the water level reaches the top opening (17) of the riser pipe (15), when the water will rise in that pipe because of the steam pressing on the top surface of the water, and will eventually reach the nozzle of the valve and the several ball packings. Condensate will drain through an opening (17a), through which steam will also pass when this opening is uncovered. The water, not being an elastic medium, will pass through the ball packings without experiencing an appreciable reduction in pressure and will drain away quickly at the outlet (7). As soon as the water level in the container (4) sinks below the orifices (17) provided in the riser pipe (15), steam enters at those orifices and the rate of flow through the ball packings slows down, more time being thus available for the accumulation of condensate. If air accumulates with the condensate, this will be discharged through the orifices (17) in the pipe (15), but being a gaseous substance will not itself cause a reduction in pressure or of

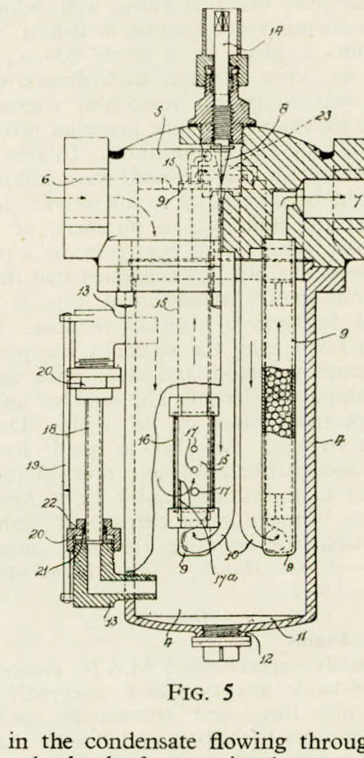


FIG. 5

rate of flow in the condensate flowing through the packings. In order that the level of water in the container (4) may be seen, there is provided a gauge glass (18) protected by rods (19) and mounted in right-angled tubes (13).—*British Patent No. 695,864, issued to E. Seaton. Complete Specification published 19th August 1953. Engineering and Boiler House Review, October 1953, Vol. 68; p. 320.*

ENGINEERING ABSTRACTS

Section 3. Shipbuilding and Marine Engineering

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Gravity Fenders

The horizontal forces of impact on marine structures are often the most costly to resist, unless effective shock absorption is provided by some kind of resilient fendering. Various kinds of springs and rubber blocks have been used, but it is difficult to obtain more movement than 12 to 15 inches with such devices, which is often not enough. Moreover, substantial vertical structural members are necessary to receive the pressure from the ships and to span it back to the springs. It is, therefore, logical to make use of such heavy members for shock absorption and to suspend them in a suitable manner for this purpose. Fig. 1 shows such a fender, which might be a concrete filled steel tube or reinforced concrete, or pre-stressed concrete, member suspended by a pair of links at the top and the bottom so that the load is fairly equally divided between them. Such fenders may be designed to recede when pressed two to five feet according to requirements, and to rise about two-thirds of the horizontal travel. They may weigh from 5 to 50 tons according to requirements, so that when fully pressed a 5-ton fender rising 20 inches would absorb 100 inch tons of kinetic energy and a 50-ton fender rising 40 inches, 2,000 inch tons. It is generally assumed that the kinetic of a glancing blow is $0{\cdot}4\left(\frac{MV^2}{2g}\right)$ where M is the gross tonnage of the vessel, and V the velocity in the direction of the blow. On exposed sites, or where navigational approach is difficult, vessels of 20,000 to 30,000 gross tonnage may collide against fendering and before collision have a kinetic energy of 500 to 2,000 tons inches. It is not easy to predict the magnitude of a blow on which a design should be based, and there is not sufficient data for statistical treatment, but at exposed berths, such as at Kuwait and Heysham where suspended fenders have been used, it has been possible to observe the movement of the fenders and thus to measure the kinetic energy of collision, and to compare with a value calculated from an assessment of the velocity of the ship at the time of impact. A velocity of one foot per second is a reasonable assumption to make on an exposed site for a large vessel and higher speeds for small vessels. If rubber and spring shock absorption devices are used where heavy collisions may occur, since their movement is limited generally to about one-third or one-fifth of the movement of gravity fenders of similar shock absorption capacity, the impact of collision is three to five times as great for collisions from vessels approaching with the same kinetic energy. Gravity fenders can also have the advantage of being able to rotate and move longitudinally slightly in order to evade projecting plates on ships, and so prevent heavy longitudinal rubbing. In cases where this is likely to occur, it can be seen From Fig. 1 that if the suspension points of the links are spaced farther apart, so that in elevation they are splayed upwards, the distance between the tops of the links being nearly twice as much as at the bottom, considerable longitudinal movement and rotation will be available. In addition to the lifting of the weight of the fender, the friction between the rubbing strips and the ship as the fender rises also absorbs energy. The determination of the size of the fender and length of links and position of their suspension points is important.

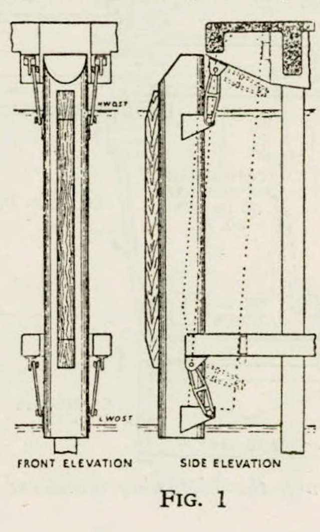
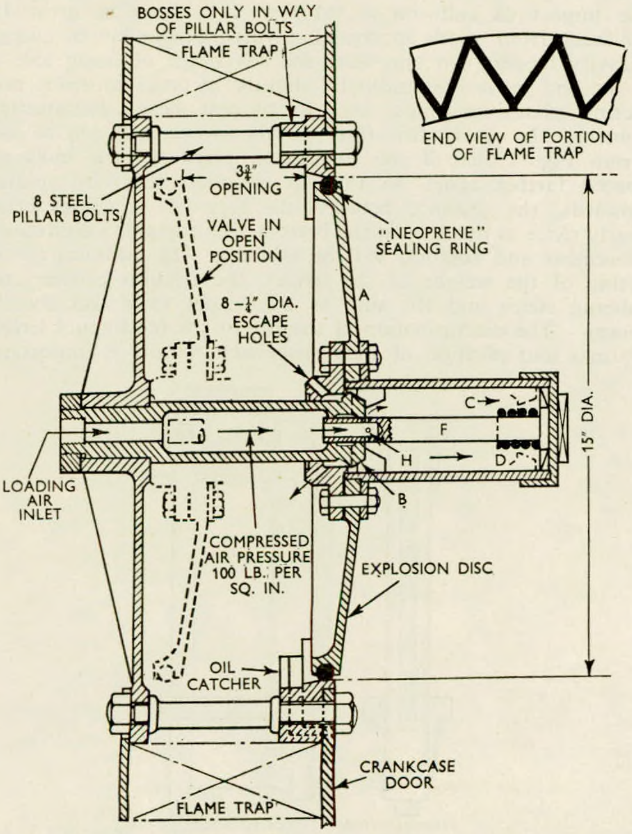


Fig. 1

The links must be set out so that extended lines through the centres of suspension intersect a vertical line through the centre of gravity of the fender. The face of the fender then remains vertical in the normal position. The length of the links must be sufficient to allow the fender to recede and to rise as much as possible and at least the required amount for energy absorption without the links being excessively stressed. A stop should be provided as a safeguard. The fenders must recede when pressed anywhere on the rubbing strip. Without care cases can occur in which a low level blow causes outward movement at the top. The fender must, of course, when fully receded, prevent ships bearing against all parts of the main structure. The horizontal force required fully to recede the fender must not be excessive in relation to the horizontal strength of the jetty or the resistance of ships to side pressure. The work required to lift the fender to its receded position must not be greater than the kinetic energy of the most severe collision which may occur. The required dimensions and weight of the fender or group of fenders therefore can be calculated from assumed weight of ships colliding at assumed velocities related to navigational conditions. It is found that such fenders can considerably reduce the forces of impact transmitted to the main structure, thereby effecting great economy and also preventing damage to shipping which would otherwise occur. Horizontal resistance is small for the initial movement so that the fenders work easily for normal conditions. Gravity fenders have been used successfully at Heysham, Kuwait and Thameshaven oil jetties and are being installed at Hamble.—*Professor A. L. L. Baker, Hansa, Vol. 90, 1953; pp. 1498-1499.*

Safety Valve for Crankcases

Although crankcase explosions continue to be rare, Lloyd's Register of Shipping now recommends that devices to prevent damage to an engine, or injury to personnel, be fitted to all crankcases in which explosions may occur. Several devices are now being fitted to prevent a dangerous build-up of pressure, and one of these, the Lightning safety valve, is of interest in that it also affords protection to the personnel and engine room against the effects from flame being emitted from the crankcase as the result of an explosion. It is marketed by Trewent and Proctor, Ltd., London, and can be made in any size, the smallest yet having a diameter of 6 inches and the largest 30 inches. When assembled, the valves are bolted to those parts of the crankcase where the effect of an explosion would be most violent. They are completely independent of the installation, except for a small-bore pipe connecting the valve to a source of compressed air, the pressure of which may range from 50-500lb. per sq. in. The valve is usually designed to open when the pressure in the crankcase exceeds 2lb. per sq. in. and to close when the pressure falls to atmospheric. The operations are instantaneous, so that not only will a serious build-up in pressure be avoided, but entry of air into the crankcase, which is generally responsible for a second and more violent explosion, is prevented. With this safety valve, it is claimed that it is possible for an engine to continue running without any alteration in speed or output during and after an explosion. Moreover, the valve remains intact and is ready to cope with any subsequent explosion. Method of operation: Referring to the accompanying illustration, compressed air enters the central hole, as indicated, and passes through radial holes (H) into the loading chamber (C). The air pressure in this chamber produces an outward force, indicated by the arrows, and holds the valve (B) firmly on its seat. The loading of this valve is such that the explosion disc (A) is held in position until a predetermined pressure in the crankcase results. When this pressure occurs and acts on the explosion disc, the disc, and with it the chamber (C) and plunger (F), move to the left. This has the effect of "cracking" the valve (B) and closing the radial holes (H); thus the compressed air in the chamber (C) immediately spills and further replenishment is prevented by the closing of the radial holes (H). With the valve now unloaded, the disc (A) opens instantly to its fullest extent, the fully open position being indicated by the dotted line. When the crankcase excess pressure has been released, the compressed air acts on the inner end of the plunger (F) and forces the loading cylinder (C), and with it the explosion disc (A), to the right. This has the effect of reseating the valve (B) and uncovering the radial holes (H), when the loading pressure in the chamber (C) is immediately built up.—*The Motor Ship, Vol. 34, October 1953; p. 285.*



Section through the Lightning crankcase relief valve.

Boiler with Pressure Firing

Pressure-fired boilers are known as those in which the variable amount of fuel supplied to the generator is burnt at a constant pressure exceeding that of the atmosphere. This pressure is produced by a compressor which supplies the combustion air; this compressor being driven by a gas turbine which is operated by the combustion gases from the boiler. In this boiler the combustion gases are conveyed at a high velocity (for example about 200 metres per second) through the evaporator and superheater tubes which form the heating surfaces of the generator. A feedwater heater is located in the system in the usual manner after the gas turbine of the steam generator, so that the heat contained in the exhaust gases can also be utilized. For a steam generator to have as high an efficiency as possible, it is necessary to cool the combustion gases to a temperature as low as possible. This cooling process is restricted in two respects. The feedwater which is supplied to the generator often has a relatively high temperature, for example over 100 deg. C., for some reason or other. In such a case the combustion gases can only be cooled down in the feedwater heater to a temperature which is determined by that of the feedwater, and which is somewhat lower than that. Even if cold feedwater were available, this cannot be supplied directly to the feedwater heater without being previously heated. Moisture contained in the combustion gases would condense on the outside of the preheater tubes through which the cold water flows; this condensate dissolves harmful components which are present in the gases, for example, sulphuric acid or sulphurous acid, and this

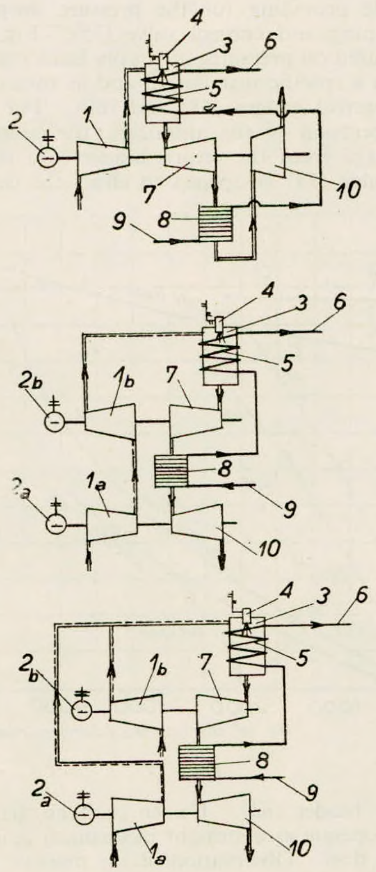


FIG. 4 *(upper)*, FIG. 5 *(middle)*, FIG. 6 *(lower)*

can result in the destruction of the preheater tubes. This condensation of the moisture in the gas is not determined by the temperature of the gas, but by that of the tube walls, the latter temperature being considerably below that of the gas. In view of these conditions it is an advantage if the last stage of the cooling of the combustion gases is not achieved by means of a transfer of heat in a feedwater heater but by allowing the gases to expand in a turbine. The walls of the casing and the rotor of a turbine attain practically the same temperature as the combustion gases; that is, they are not colder than the gas. The gases can, therefore, be cooled to a lower temperature in a gas turbine than in a feedwater heater, and this improves the efficiency. Furthermore, the amount of material in a turbine is much smaller than in a preheater, so that from the point of view of cost there is a greater possibility of using a special material which is more durable. In Fig. 4 the numeral (1) indicates an air compressor which raises the pressure of the air drawn in from the atmosphere to the pressure required in the combustion chamber. There is a starting motor (2), a combustion chamber (3) with the fuel nozzle (4), an evaporator (5) and a superheating surface. The steam which is generated is supplied through pipe (6) to the consuming components. The turbine (7) is the first turbine arranged in the path of the combustion gases. The feedwater heater (8) is supplied through pipe (9). The gas turbine at the end of the gas path is indicated by reference number (10). Compressor (1) and the two gas turbines (7) and (10) are mounted on the same shaft. In Figs. 5 and 6, the machines, apparatus and pipes, which are the same as those in Fig. 4, are designated by the same reference numerals. Since it may often be expedient to divide the compressor into the separate units like the gas turbines, these compressors (1a) and (1b) together with their driving turbines (10) and (7) respectively are located one behind the other. Fig. 6 shows the two compressors (1a) and (1b) which are arranged in parallel and are each driven by a turbine (10) and (7) respectively.—*British Patent No. 697,074, issued to Aktiengesellschaft Brown, Boveri et Cie. Engineering and Boiler House Review, November 1953, Vol. 68; p. 354.*

Radial Inward-flow Turbines

Recent developments in the field of small gas turbines have focused new attention on the radial inward-flow principle of turbine design. Though the interest has been great, it is said that the most important merit of the inward-flow design has been overlooked. In a recent A.S.M.E. paper (No. 53-S-16) it is stated that the inward-flow turbine can be designed for specific outputs far exceeding those of the axial-flow wheel. This characteristic is attributed to the ability to handle larger flows and higher enthalpy drops at higher r.p.m. with better efficiency and lower stresses. In the hydraulic turbine field, the inward-flow (centripetal) turbine—known as the Francis turbine—has been used widely for the past 100 years. No counterpart of the Francis turbine was designed for elastic fluids, however, until studies by De Laval, begun in 1928, led to the construction of two different experimental units. Experiments with these units proved the low-specific-speed type to be inferior to the axial-flow in many cases and led to the abandonment of the low-specific-speed version, and concentration on the high-specific-speed version. The high-specific-speed unit frequently can be used to better advantage than the axial-flow type. Three limitations are characteristic of the inward-flow turbine, though they are not always considered disadvantages. These are: It usually occupies a greater axial space than an equivalent axial-flow type—The stationary flow passages must be arranged outside the turbine wheel to allow radial inward feed into the wheel—Multi-staging is not as simple as it is with the axial-flow type, but it can be achieved—see Fig. 3. This factor may limit the design to single stage turbines or to the last stage or stages of multi-stage turbines. Contrasted with the preceding limitations, the following distinct advantages over the axial turbine are offered by the inward-flow type: It is capable of handling larger flows and/or of operating at higher rotational speeds under conditions of equal stress—It is capable of handling larger stage enthalpy drops efficiently—It gives higher turbine efficiency because the following factors are taken into account: (*a*) The effect of expansion takes place *against* the centrifugal force. (*b*) It operates at a higher Reynolds' number. (*c*) It operates at a lower Mach number. (*d*) Blades can be designed for balanced flow, while all-radial blade sections are maintained. (*e*) Excellent recovery is possible in a simple diffuser of the

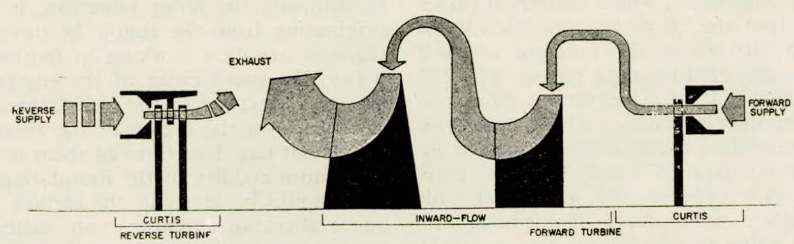


FIG. 3—*Diagrammatic representation of a multi-stage inward radial flow turbine of the marine type*

kinetic energy leaving the turbine blades. It is simpler and cheaper to manufacture and is more rugged because of the low number of blades and because of their configuration. The blades and hub can be made as a single piece, eliminating the age-old problem of blade fastenings. It has a lower moment of inertia permitting more rapid acceleration. Blades and hubs can be air-cooled effectively in a simple manner, permitting operation with very high gas temperatures for which the strength of the uncooled metal would be insufficient.—*Power Engineering, Vol. 57, No. 9; p. 100. Engineering and Boiler House Review, November 1953, Vol. 68; p. 351.*

Return Flow Fuel Burner

Mechanical atomizing liquid fuel burners adapted to operate with the return flow of fuel from a whirl chamber have the advantage that the efficiency can be maintained over a wide output range. The present invention includes the method of regulating the firing rate and with it flue gas production by a mechanical atomizing liquid fuel burner adapted to operate with the return flow of fuel from a whirl chamber which comprises concurrently, varying the pressure of fuel supplied to the burner, the pressure at the burner of fuel in return flow from the whirl chamber and the pressure difference between the supply and return pressure, so that all three quantities increase or decrease as the burner output increases or decreases.

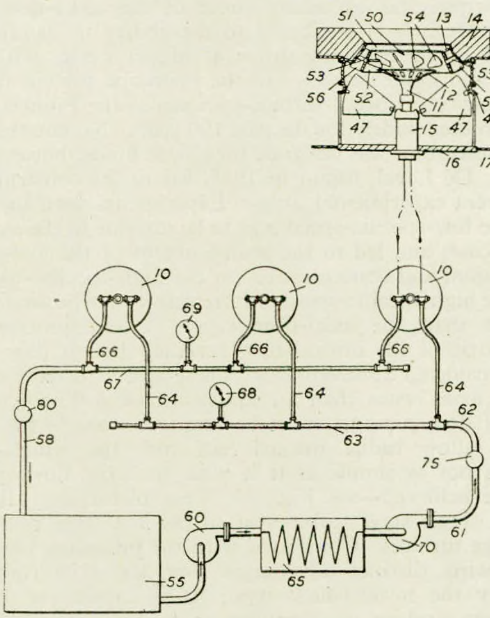


FIG. 6

Fig. 6 shows in outline a combination of oil piping, pumps and control apparatus to serve a number of return flow atomizers so that burner operation may be efficiently performed through a wide range of burner capacities without carbon deposition and with a relatively low rate of return oil flow, and with a fixed axial position of the atomizer. The oil is taken from a storage tank (55) by a pump (60), which delivers it under pressure to a heater (65) to heat the oil so that its viscosity is sufficiently reduced. Pump (70) raises the pressure to such an extent that the oil may be delivered through piping (61) (62) (63) and (64) and control valve (75) to a number of atomizers (10) at the desired maximum inlet pressure. The return flow passage (20) of the several individual atomizers is connected by branch lines (66) to a return oil main (67) and this main connects through control valve (80) and pipe (58) to the oil tank (55). For operation through a wide range of loads an oil pressure of a predetermined value is required in the manifold (63) for maximum output operation, and the two pumps (60) and (70) operating in series are selected to provide such a pressure, while providing for the pressure drop through the connecting piping and control valve (75). Fig. 7 shows the relation of desired oil pressures in supply header (63) and return header (67) in a specific installation and as measured and indicated by respective gauges (68) and (69). For maximum or full rating operation of the atomizers, the control valve (80) on the discharge from the return header (67) is closed, while the control valve (75) is opened to effect the desired pressure

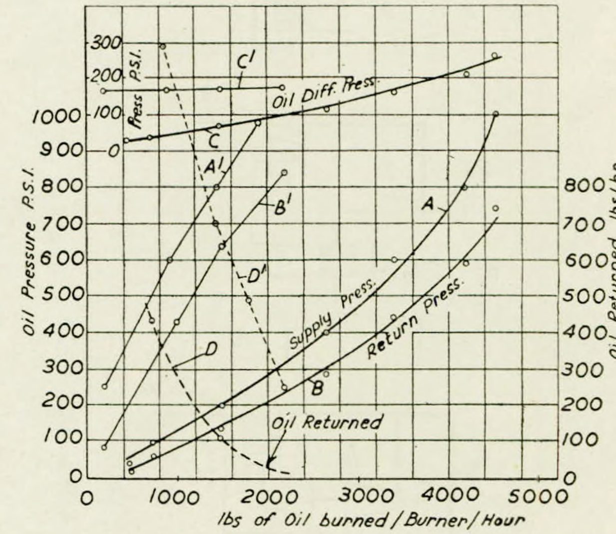


FIG. 7

in the supply header (63). Under such an arrangement the atomizer will operate as a straight mechanical atomizer without any return oil flow. Observation of the pressure gauge (69) of an operating installation of the described capacity has shown that the whirl chamber static pressure is of the order of 740lb. per sq. in. if the supply pressure is, for example, of the order of 1,000lb. per sq. in., and the atomizers operate with a pressure difference of 260lb. per sq. in. between the supply header and the whirl chamber at such supply pressure. For effective wide range operation down to a minimum output, a predetermined reduction of pressure difference between the supply header and the return header is effected as shown for the specific installation by curve C of Fig. 7 through manipulation of both of the control valves (75) and (80).—*British Patent No. 696,071 issued to Babcock and Wilcox Limited. Application in U.S.A. made on 8th September 1949. Complete specification published 26th August 1953. Engineering and Boiler House Review, October 1953, Vol. 68; pp. 320-321.*

Marine Diesel Engine Installation

A marine Diesel engine cannot be arbitrarily installed in the ship, and the foundation must be correlated with the ship's structure. If the frames and plates below the engine foundation are designed solely in consideration of hull construction, there is a strong possibility of the engine inducing resonance vibration in the hull, due to flexibility of the foundation and resultant low frequency range of transverse vibration. In order to eliminate the latter vibration, it is essential that the forces originating from the engine be directly transmitted to the hull through members working in tension so as to avoid resonance within the speed range of the engine. The foundation should be rigidly secured to the hull frames. It is advisable to design the frames in the region of the engine as closed frames and to install not less than three of them over the length of the engine. Maximum rigidity of the foundation is a point to which attention should be given in the earliest possible stage, since subsequent alterations require more material, time and expense and the results are often questionable. As a rule, it is very difficult, if not impossible, to eliminate vibration in a finished installation. If vibrations of large amplitude persist, they will adversely affect the engine, foundation and hull over the long run, and damage is liable to ensue after an extended operation period. It is possible to secure the engine to the hull ribs with tie-rods at the level of the cylinder heads, with the result that the inertia moment is considerably increased and the transverse vibration resonance is shifted above the operational speed range. Such expedients, however, are a mixed blessing since they curtail the limited available space, render the engine more difficult of access and inconvenience the personnel. Such measures can only serve a useful purpose in cases where the hull as such has adequate rigidity and the engine foundation or its connexion with the hull is too weak. Recent measurements have shown that stiffening of the aforementioned type can be employed to advantage with high-powered engines if the stiffening members are integrated into the hull structure. Efficient cross-staying of the engine at the level of the cylinder block, of course, presupposes stiffness of the elements to which the cross-stays are attached. The best way of ensuring rigidity is to install cross-members of the hull frames at the ends of the engine, thus transforming the frames into ring-shaped structures. The cross-members, of course, must be rigidly secured to the frame to ensure stiffness of the ring frame. This arrangement results in large moment of inertia and provides points of attachment for reamed bolts or cross-stays with which the engine can be braced at the level of the cylinder block. The natural vibration frequency of the system is thereby pushed up to such a degree that transverse vibration becomes practically impossible. This expedient has been successfully tried in long high powered engines (ten cylinders) which were made to run so quietly throughout their speed range that it was no longer possible to notice any vibrations in the whole ship's hull. Since this eliminates resonant conditions, the stresses imparted to the bolts or stays are so unimportant as to be of no consequence. In order to protect the engine against the action of external forces, the fastening bolts can be designed as shear bolts calculated to withstand a predetermined fracturing load.—*W. Biber, M.A.N. Diesel Engine News, No. 27, 1953; pp. 7-15.*

Turbo-electric Tanker

The *Helix* is the first of four turbo-electric vessels forming part of the Anglo-Saxon Petroleum Company's order for fifty-one general purpose tankers, the remainder of which (apart from the *Hemisinus* which will be propelled by gas turbines) are to be steam turbine driven. The turbo-electric form of propulsion enables a main turbine to be removed from the ship for survey and replaced by another unit in a very short time, since the prime mover is independent mechanically of the propeller drive. The *Helix* embodies a number of interesting new features developed by the Shell Research and Development Department under the direction of Mr. John Lamb. The *Helix* is a single-screw turbo-electric vessel, of which the following are the principal particulars:—

Length b.p.	530ft. 0in.
Breadth moulded	69ft. 3in.
Depth moulded	39ft. 0in.
Load draught	29ft. 8⅝in.
Load deadweight	17,780 tons
Gross tonnage	12,089 tons
Capacity of oil cargo tanks ...	17,467 tons at 50 cu. ft. per ton 98 per cent full
Service speed	14½ knots

The cargo spaces are fitted with what is called the "double ring main pipeline". The purpose of this is to obtain with one pump room the same cargo flexibility as is obtained in a vessel with two pump rooms. Four main turbo-driven cargo pumps of the vertical centrifugal 12in. by 12in. two-stage type, each having a discharge rate of 400 tons of oil per hour, were supplied by Drysdale and Co., Ltd., and are fitted in the main pump room for dealing with the main oil cargo, with the necessary pipe lines for loading and discharging. The steam turbines driving the pumps are placed in the engine room on a

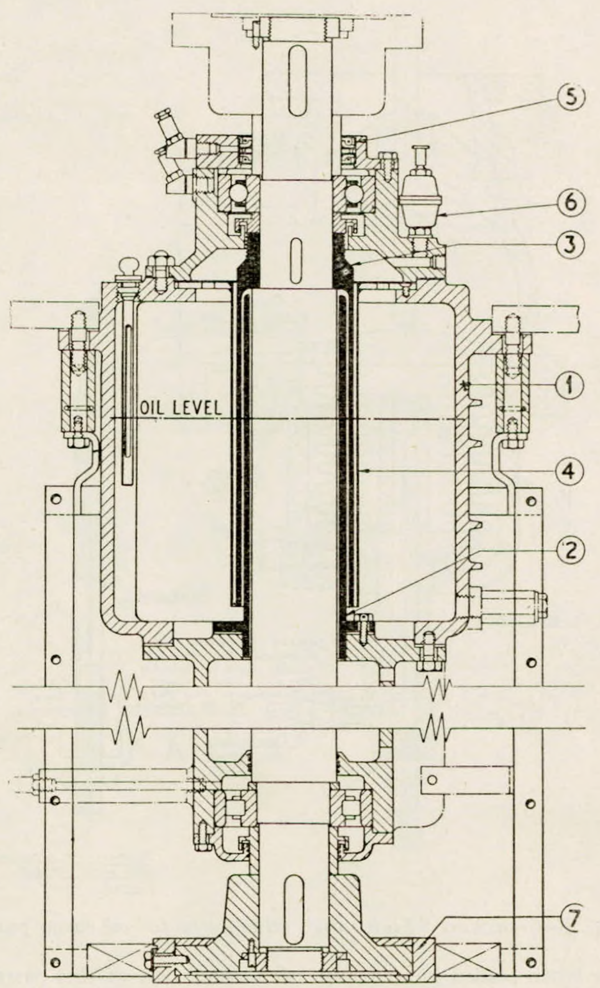


FIG. 1—*The pump drive gas seal*

The gas seal casing (1) is partly filled with light turbine oil, which is the sealing medium. The stationary sleeve (2) and rotating air bell (3) form the gas-tight labyrinth sealed by the oil. When sleeve (3) is rotating, the oil tends to be thrown off in a forced vortex which would seriously reduce the effectiveness of the seal. This is prevented by the closely fitting gauze screen (4) which stops the formation of the vortex and also prevents undue aeration of the oil. The engine room is protected against ingress of oil should the pump room become flooded by the shaft seal (5) and the float operated air vent valve (6). The seal casing is air cooled by the fan (7). The joint between the turbine stool and the deck plating is made by a gland and packing ring. The seal is effective against a gas pressure of 3in. water gauge when the pump is running and 6in. water gauge when the pump is stopped.

flat situated 18 feet above the keel, thus reducing the risk of petroleum entering the engine room in the event of partial flooding of the pump room. Such an arrangement saves floor space and at the same time enables a frictionless gas sealing gland (see Fig. 1) to separate the pump and engine rooms and, therefore, eliminates the explosion hazard of an overheated packed gland. Special cargo pump relief valves are fitted, since with centrifugal type pumps it is particularly important that excessive build-up of pressure should be prevented. It is also an advantage if the load on such pumps can be thrown off in a much shorter time than would be required to stop the pump or to open a screw-down valve. To meet these conditions the "Lightning" relief valve is loaded by compressed air instead of a spring, the compressed air being applied in such a way that the load on the valve remains constant throughout the whole of its travel instead of being progressively increased as is the

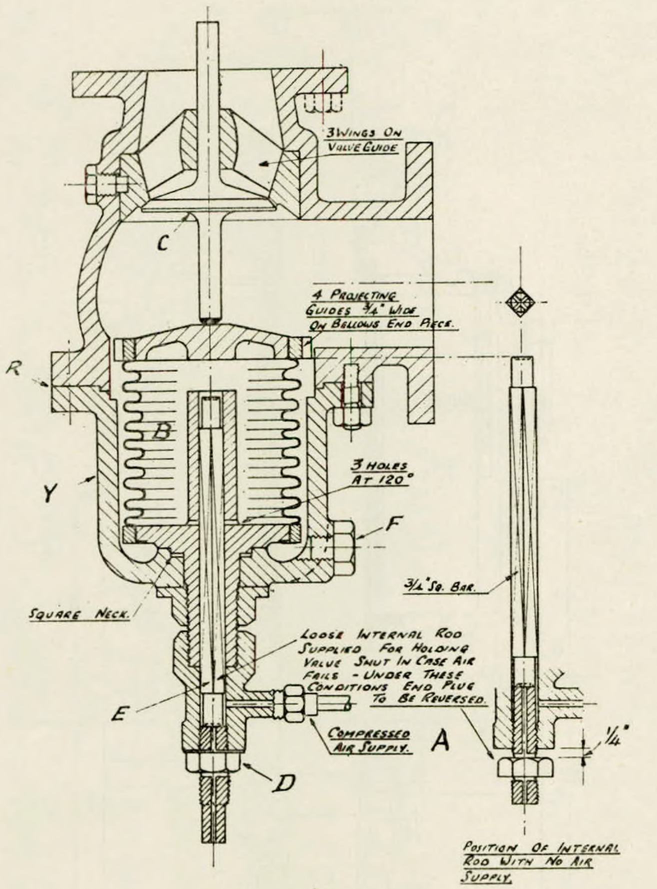


FIG. 2—*5-in. bore "Lightning" relief valve for oil cargo pumps*

case when loaded by a spring. A section view of this patented relief valve is shown in Fig. 2, from which it will be seen that compressed air at 100lb. per sq. in. enters at A and passes into the 3-ply corrugated bellows B, the upward pressure exerted on the upper end of the bellows holding the mitre valve C firmly on its seat. Should the pump discharge pressure exceed the maximum permissible pressure the relief valve will open. As the loading pressure on the relief valve remains uniform the predetermined relief pressure will not be exceeded even should the discharge line be abruptly blocked. Two stripping pumps, one 15in. by 10in. by 18in., vertical duplex steam-driven, supplied by Hayward Tyler, Ltd., and one rotary pump supplied by Drysdale and Co., Ltd., driven by an electric motor, are fitted, the motor being situated in the machinery room. Instead of the usual stuffing box, which may leak and cause contamination or admixture of cargoes, the cargo line expansion fitting is in one piece and permanently bolted at both ends to the cargo pipeline, thus eliminating any possibility of leakage. The fitting is in the form of a corrugated tube and is made of a material which is resistant to corrosion. The propelling machinery in the *Helix* is similar to that supplied for the *Helicina* and the *Hyalina,* completed shortly after the war, in that two propulsion turbo-alternator sets are mechanically paralleled to power the single screw. In this system the propeller motor frame contains two electrically independent units, the rotors of which are mounted on the same shaft, but each stator is supplied independently by one of the turbo-alternators. The equipment was supplied by the British Thomson-Houston Co., Ltd., and includes the 7,500 s.h.p. normal rated turbo-electric propelling machinery, two 700 kW. normal rated geared turbo-alternator/exciter sets, auxiliary motors, control gear and switchgear. Two ratings are allowed in the design of the propelling machinery, the normal rating of 7,500 s.h.p. at a propeller speed of 100 r.p.m.; and the trials rating, which is also the maximum, of 8,300 s.h.p. at 103 r.p.m. Steam conditions at the turbine stop valve are 425lb. per sq. in. gauge, 750 deg. F. temperature, and the turbines exhaust to a vacuum of 28½in. Each turbo-alternator set is rated 2,900 kVA. normal, 3,000 r.p.m., 3,000 volts, three-phase, at unity power factor, which condition corresponds to a propeller speed of 100 r.p.m. For the maximum rating each set is to be capable of 3,200 kW. at 3,090 r.p.m., with sea water not exceeding 75 deg. F.—*The Shipping World, 4th November 1953, Vol. 129; pp. 377-379; 383.*

Oil Burning Method

It is well known that as the operating temperatures of oil fired equipment such as boilers have been gradually increased during the past decade, the general problem of slag formation and decomposition of the metal heating surfaces and supports of such equipment has become more acute. In more recent years, this problem has been further complicated by the fact that the present fuel oils, which are mostly Venezuelan and Arabian oils, result in the formation of combustion products which, at the higher operating temperatures are extremely corrosive as regards the metal tubes, alloy supports, partitions, plates and so forth, which are located in the high temperature zone of the boiler. This corrosion problem appears to result from the fact that the present fuel oils have a high vanadium content as compared with American oils. It has been noted that with these fuel oils, the actual type of attack on the heating surfaces is of two types. One is a washing fluxing or local rapid oxidation of the various metal tubes, supports, partitions, plates, etc., due either to a direct attack by the vanadium oxides, such as Va_2O, acting as a catalyst, or reaction of the vanadium with the sulphur compounds present in the ash, such as sodium sulphate, Na_2SO_4, to form a lower melting eutectic and, in addition, the formation of substantial quantities of the more corrosive sulphur trioxide. In any case, the vanadium apparently exists in some phase in the oil ash and this phase is molten and runs like molasses at temperatures approaching 1,200 deg. F. A eutectic or melting point of 950 deg. F. has been noted with 75 per cent V_2O_5, and 25 per cent Na_2SO_4. The other type of attack is a corrosive or oxidation attack which is general on the heating surface but exceeds the normal oxidation rates of the materials. Since many of the component parts of present oil fired equipment operate at surface temperatures of at least 1,200 deg. F., the pitting, worm tracking, and washing away of the surfaces is particularly evident. For example, the life of the boiler tubes and supporting elements under these conditions may be only a matter of months. Various methods have been proposed for remedying this situation. For example, ceramic and metallic protective coating on the surfaces which are subject to attack have been tried with some degree of success. However, such coatings do not always adhere well, and the ceramic coatings are inherently poor heat transfer materials. At the best, such coatings offer only temporary protection, particularly in the case of vanadium bearing oils. Even in those cases where good adherence of the coatings is obtained, their effective life as neutralizing agents is limited. The aim of the present invention is to prevent the formation of a vitreous coating which adheres to the furnace wall, and to ensure that the ash occurs in a loose state. This is achieved by raising the melting or fusing point of the ashes above the usual furnace temperature (i.e. 1,200 deg. F.) so that the ashes can exist only in the form of loose or pulverizable material, and are not liable to deposit as a hard continuous coating on the furnace wall. This is achieved by introducing into the combustion zone a mixture which comprises, or upon combustion results in titania, alumina and silica. It has been found that normally coal fly-ash obtained from burning pulverized coal contains such a mixture which can be readily used for the purpose of the invention. The coal fly-ash in itself has a fusing temperature of from 1,900 to 2,300 deg. F., and may be employed in amounts sufficient to introduce into the combustion zone from 0·5 to about 1·5 parts coal fly-ash for each part oil ash normally obtained, so that the mixture of the oil ash, including the vanadium oxide, and the fly-ash will produce a material having a melting point well above the operating temperatures of surfaces comprising the oil fired equipment. About a 1 : 1 ratio of coal fly-ash containing titania and oil ash is preferred.—*British Patent No. 697,101, issued to The British Thomson-Houston Co., Ltd. Engineering and Boiler House Review, November 1953, Vol. 68; p. 355.*

Marine Applications of Polyvinyl Materials

The special properties of polyvinyl chloride have been known for many years, but the emergence of this material as a major plastic is a comparatively recent development, largely a consequence of the rapid increase in the production of home-produced acetylene to meet wartime demands. P.V.C., as it is generally known, is derived from vinyl chloride, a gas which is produced commercially by the catalytic hydrochlorination of acetylene. The single molecules of vinyl chloride monomer are caused to link up under carefully controlled conditions to form the longer chain molecules which constitute polyvinyl chloride. The polymerization products can be made with considerably varying properties by modifying the conditions under which the reaction takes place. The resins so produced are compounded with various other materials in order to develop the particular properties which their ultimate purpose demands. Although P.V.C. will soften under heat it will not propagate flame. The basic resins are by their chemical nature non-inflammable and their inherent fire resistance can be increased by a suitable choice of added ingredients. Compounds which for all practical purposes are completely non-inflammable can be produced with selected plasticizers, such as the phosphates, which are themselves fire-repellant. P.V.C. products are extremely hard wearing, are easily cleaned, and apart from a slight and temporary stiffening in extreme cold, are unaffected by low temperatures. Due to its fire-resistant properties, P.V.C. is replacing rubber to an increasing extent as a wire and cable insulation. Though its resilience is not so great as that of rubber, P.V.C. sheathing is tough and flexible and will stand up to most applications. It is more resistant to abrasion than all but the toughest grades of rubber, and for this reason wires insulated with this material need no outer covering. The electrical properties of P.V.C. are influenced by the nature of the compounding materials used, but in general the dielectric strength is comparable with that of vulcanized rubber. Though its temperature range is more limited than that of rubber, P.V.C. is readily compounded to comply with the requirements for tropical and arctic service. A considerable amount of work has been carried out in Britain on P.V.C. linoleum-type floor covering. This product is made in two-yard widths, has a hessian back and all the appearance of conventional linoleum, but the linseed oil binder has been replaced by plasticized P.V.C. The Admiralty has been greatly interested in this flooring and has produced a specification (DNC/M/92) controlling its performance. Some years ago several small ships were equipped with this material as a deck covering and observations show that its wearing qualities are equal to that of linoleum.—*The Marine Engineer and Naval Architect, October 1953, Vol. 76; pp. 408-414.*

Sweat Damage on Board Ship

The new Cargocaire system prevents sweat damage by removing the normally present excess moisture and maintaining a balance in the hold so that the moisture will not condensate in the air. The three main components of this system are as follows: the Cargocaire unit, the hold ventilation system, and the Brown recorder and Foxboro Dewcels. A unit is installed in the engine room of a *Mariner*-class vessel and, through a dry air duct, supplies dry air to the hold ventilation system. Once started the unit is completely automatic, thereby assuring a continuous supply of dry air when required. The hold ventilation system is, in effect, comprised of two separate duct systems, supply and exhaust. The flow of air through each of these systems is controlled pneumatically by dampers fitted in the cargo hold damper houses. There is one damper house for each hold. A supply fan located in each produces the necessary air flow through the supply duct system in each hold. The exhaust ducts provide a path for the air to return to the damper house. It is then exhausted to the atmosphere or returned to the supply duct, depending on the position of the recirculation damper. The dry air duct from the Cargocaire unit is also connected to each cargo hold damper house. Selection of the correct method of operating the system is determined by reference to the Brown recorder. This recorder indicates the condition of the air in each hold, outside air, and the conditions of the dry air produced at the Cargocaire unit; as well as keeping a permanent record of the factors involved. These three components, the Cargocaire unit, the hold ventilating system, and the recorder and Dewcels make up the system—a system for maintaining the dewpoint of the air in the holds at safe level. The unit is fundamentally an air drying machine. It takes air from the atmosphere (machinery space) and extracts moisture from it, thereby providing a supply of air which has a low dewpoint. This air, when mixed with the hold air, lowers the dewpoint of the hold air so as to prevent sweating on either cargo or structure. The air drying agent used in the new Cargocaire unit is triethlyene glycol, known as "Caire-col". It is a green liquid that is soluble in water, odourless, with an initial boiling point of 540 deg. F. It has a specific gravity of 1·125 and weighs 9·3lb. per gallon. In its pure state, it has the ability to absorb large quantities of moisture. In the Cargocaire system, the air stream is brought into intimate contact with the Caire-col solution by passing it through a chamber which is continually sprayed with finely divided particles of Caire-col. The difference in the partial pressure (vapour) of the water in the concentrated Caire-col and the partial pressure of the water vapour in the air causes the water vapour to be given up by the air to the Caire-col. The water vapour is condensed during this operation and its addition to the Caire-col solution results in a decrease in its concentration. To keep the solution at a high concentration, a portion of the absorbent must be continuously reconcentrated. The weight of the initial charge of Caire-col is 1,400lb. Both main and stripping sumps are provided with a system of baffles. In this way operation of the unit is not affected by pitching and rolling of the vessel.—*The Shipping World, 30th September 1953, Vol. 129; pp. 268-269.*

Oil Pollution Research

As a contribution to the campaign against the pollution of the beaches by oil residue jettisoned by ships, the National Institute of Oceanography will shortly initiate an intensive research into the surface currents in the North Atlantic to the west of the British Isles. It is planned to drop into the sea 10,000 plastic envelopes, many of which will eventually float ashore on the coasts of Britain and other N.W. European countries. Each envelope will contain a franked addressed postcard, on which will be printed a simple questionnaire, and a small sheet of instructions for completing it. To make the envelopes catch the eye more readily, the postcard will bear a wide red stripe and the instructions will be on yellow paper. Finders will be asked to write on the postcard their name and address, and the date and place of recovery of the envelope. The instructions will be printed in eight different languages. A reward of 2s. 6d., or its equivalent in foreign currency, will be paid to the sender of each postcard which is received by the National Institute of Oceanography at its headquarters at Wormley, near Godalming, Surrey. The cards will be numbered, and the date and position in which each is dropped will be recorded. It will thus be possible to work out the approximate speeds and directions of the currents from the information received on the postcards. If a large percentage of the envelopes dropped in any area is recovered, it follows that oil jettisoned in that area is very likely to drift ashore. Dropping will be carried out from Coastal Command aircraft on navigational training flights. In order to learn more about seasonal changes in the currents, about 2,000 envelopes will be dropped in early spring, autumn and winter of 1954; double that quantity will be released in early summer, when the chances of recovery by holidaymakers between one and three months later, will be greater than during the other seasons.—*The Shipping World, 28th October 1953, Vol. 129; p. 355.*

Largest German-built Motor Tanker

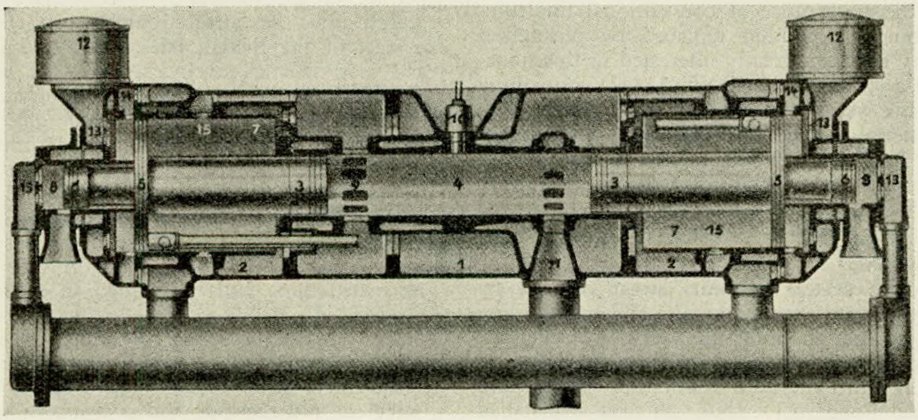
The largest Diesel-engined ship built in Germany since the end of the war is the 21,500-ton tanker *Jarmina,* recently completed by the Kieler Howaldtswerke A.G. for the S.A. Noravind, Sandefjord (mgrs. Anders Jahre). The hull is identical with that of a series of eleven turbine-driven tankers ordered from the same yard, the *Jarmina* being the only vessel of this type to be Diesel-engined. She has the following main particulars:—

Length overall	590ft. $0\frac{13}{16}$in.
Length b.p.	552ft. 0in.
Breadth moulded	73ft. 10in.
Depth to main deck ...	41ft. 0in.
Deadweight capacity ...	21,500 tons
Corresponding draught ...	31ft. 6in.
Gross register	13,996 tons
Net register	8,298 tons
Capacity of cargo tanks ...	1,020,508 cu. ft.
Capacity of dry hold ...	38,387 cu. ft.

The ship has been built to the requirements of Norske Veritas Class +1A1− carrying petroleum in bulk. The equipment is in accordance with the Board of Trade Rules and the accommodation complies with the regulations of the Norske Seekontrolle. The *Jarmina* is of all-welded construction except for two riveted bilge seams and main deck stringer angle bars. The forward and after ends of the hull and the wing tanks have been constructed on transverse frames, while the centre tanks, deck and bottom are built on longitudinal frames. The ship has ten centre tanks and twelve wing-tanks and there are two pump rooms. The whole crew has been accommodated in single-berth cabins, partly amidships, partly in the poop. A service speed of 15 knots is maintained by a seven-cylinder two-stroke double-acting Howaldtswerke-M.A.N. engine of 8,400 b.h.p. running at 116 r.p.m.; the cylinder bore is 700 mm. and the stroke 1,200 mm. The turbine-driven vessels of this type will have 10,000 s.h.p. and a speed of 16 knots.—*The Motor Ship, September 1953, Vol. 34; p. 245.*

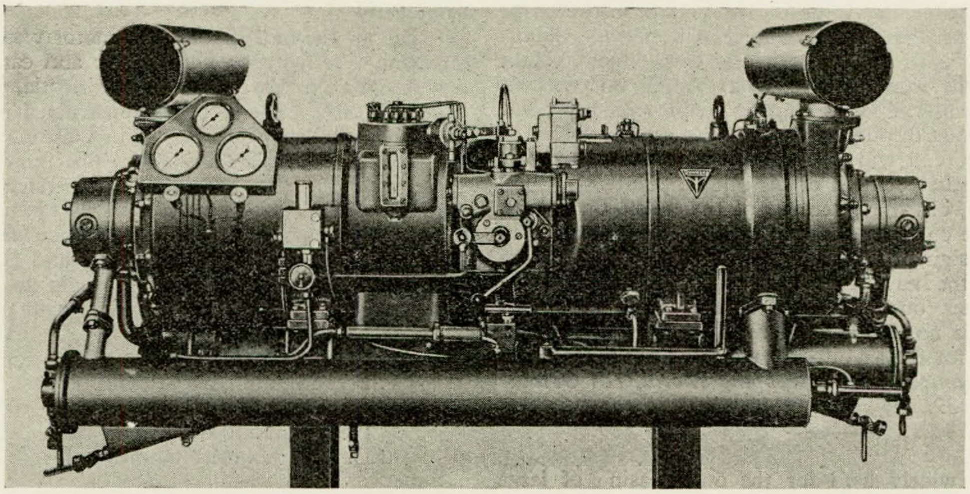
Junkers Free Piston Compressor

For several years Junkers compressors have been used in German submarines and merchant vessels, and reports have been received of Junkers compressors giving 25,000 hours' service without major overhaul. This type of unit employs the opposed piston principle of the Junkers engine and completely dispenses with the crankshaft, rotating parts, flywheel, couplings and other connecting parts, etc., which are to be found in conventional Diesel-driven air compressors. No separate prime mover, either electric, petrol or Diesel is required, the unit being completely self-contained. This being so, there is no clutch to operate, adjust or to get out of order. The Diesel section is a fully scavenged single-cylinder two-stroke built-in unit. In view of the fact that there is but one fuel injector and one fuel pump unit, there is a minimum of small high precision parts to give trouble which are costly to renew. Due to the simplicity of this design, such parts as may require attention are readily accessible and little delay is caused through repair work. A very high efficiency, coupled with low fuel consumption, is brought about by the fact that the stroke of the engine is not limited mechanically and that it is automatically varied according to the load put on the compressor. The amount of fuel injected is also automatically adjusted at the same time. As a result of this variability of stroke the compression ratio varies between 20:1 and 28:1. This ratio is much greater than the usual one of 15:1 found in normal Diesel engines. These points, coupled with the spherical form of the combustion chamber, gives high thermal efficiency, low heat losses and low fuel consumption. For example, the model 2FK115A gives 120 cu. ft. F.A.D. per minute at 100lb. per sq. in., using approximately one gallon less per day of fuel oil than a comparable size of a normal designed compressor. Air is cooled between each stage and an aftercooler is fitted. Because of the few moving parts in this type of compressor, there is less likelihood of mechanical breakdown. The total enclosure of all parts gives 100 per cent protection from dirt and damage by extraneous causes, yet complete accessibility is not sacrificed. Lubricating oil is metered to all parts requiring lubrication giving positive feed in correct proportions. Another factor leading to reliability is the slow speed of the unit, being approximately 800 strokes per minute. As no connecting rods are used, side thrust to pistons is eliminated and cylinder wear reduced. Due to the fact that lubricating oil is held in the metering oil pump reservoir and there is no sump for holding the bulk oil, there is no necessity to provide a filtering medium as no oxidization can take place. The method of operation is simple and is effected in the following manner: For starting, the pistons are brought to their outer positions and compressed air is admitted to the compressor cylinders, the pistons thus being forced



Sectional view of two-stage free piston compressor

1.—Engine housing. 2.—Compressor housing. 3.—Power piston. 4.—Power cylinder. 5.—Compressor piston, single-stage. 6.—Compressor piston, 2-stage. 7.—Compressor cylinder, single-stage. 8.—Compressor cylinder, 2-stage. 9.—Scavenging ports. 10.—Injection nozzle. 11.—Exhaust. 12.—Air intake filter. 13.—Suction valves. 14.—Pressure valves. 15.—Scavenging air valves.



Junkers free piston Diesel compressor

inwardly to compress the air for combustion in the power cylinder. Diesel oil is then injected into the air heated by the high compression and the resulting combustion in the cylinder causes a build-up of pressure that drives the pistons outward again. A part of the air confined in the compressor cylinder is used for scavenging the power cylinder and is passed on through the scavenging air valves to an air receiver. The remaining part of air is compressed in two stages up to the working pressure of the compressor.—*The Shipping World, 21st October 1953, Vol. 129; pp. 337-338.*

Cargo Liner Speed Ten Years Hence

The average service speed of new cargo liners is now considerably above that of corresponding vessels constructed fifteen years ago. It is uncertain whether this trend towards higher speed will continue, but the probabilities are that it will. In ten years' time, which is only half the lifetime of a cargo liner built today, it is not unlikely that an average speed of about 16½ knots will be considered desirable rather than one of 15 knots, which represents the speed of a good proportion of the world's cargo liner tonnage today. British cargo liner owners cannot ignore the higher speeds of some of the new vessels which have lately been ordered or completed by Scandinavian, German and Netherlands owners, all with substantially higher speeds than those now current. These include, for instance, the East Asiatic Company's 10,000-ton 17-knot ships with about 9,000 b.h.p. machinery, vessels for A. P. Möller, Copenhagen, of the same tonnage with 10,600 b.h.p. machinery and a speed of 18 knots, the Wilhelmsen 10,600-ton 17½-knot cargo liners with 9,100 b.h.p. machinery and the Hamburg American Line's 10,000-ton class with a speed of 17¾ knots and geared machinery of 10,600 b.h.p. All of these are single-screw ships, the power in question coming well within the scope of the Diesel engine. The higher speed involves a substantially bigger outlay and some owners may not consider the time is ripe for paying the large sums which are called for to give the necessary power. Nevertheless, with the demand for faster ships probably becoming insistent in the next few years, the expenditure will be deemed necessary, but in cases where it is thought that the lower speed is sufficient for normal circumstances under present conditions, it would seem to be a wise plan to design the ship so that the power of the machinery can be increased without difficulty. The opportunity is now offered for such an arrangement to be made, since if two-stroke single-acting non-supercharged engines of a power suitable for the lower speed be installed, the additional speed of, say, 1½ knots could be obtained by providing the necessary equipment for turbo-charging at a later date, since the extra power would not be less than 35 per cent. This, however, means making the necessary provision in the early design stage so that the modifications do not involve a larger engine room, or structural alterations that would be difficult or impossible to make. It is almost certain that the turbo-charged two-stroke engine will become universal within the next five to ten years, so that there is not much risk in making the necessary plans at the present time. It need hardly be added that in order that such a ship should be economic it should, in the first instance, be designed to operate on boiler oil, but here again, if owners do not wish to go so far at the time of building, the design should be such that the changeover to heavy oil can readily be made. With turbo-charged two-stroke engines becoming more common, the question will undoubtedly arise in the minds of shipowners as to the feasibility, in certain cases, of increasing the speed of existing ships by adding the necessary turbo-charging equipment to normally aspirated two-stroke engines. There are possibilities in this direction, no doubt at the expense of higher fuel consumption unless some modifications are made to the engine, but the disadvantages might be accepted provided the increased power could be obtained in the existing machinery space. The cost might be substantial, especially if new propeller shafting were required—which is probable—but would be very much lower than that involved by the installation of new machinery.—*The Motor Ship, September 1953, Vol. 34; p. 222.*

Lighting on Tankers

The classification societies limit the lighting system, whether D.C. or A.C., to 115 volts on tankers, which is readily accepted as a wise and right regulation. It presents an easy proposition through the use of transformers for any A.C. system, and also for those tankers where 115-volts generation is adopted throughout for the D.C. system on account of the restricted displacement of the tanker or the use of steam-driven auxiliaries making the lighting load a high proportion of the whole. For medium size and large tankers where 230-volts generation is installed, the simplest and most economical provision for lighting is to install the three-wire system with a static balancer or compensator. For this, the generator of standard marine construction has added to it slip rings mounted on its shaft, connected to a compensating or choking coil mounted in a protecting case. The coil resembles the primary winding of a small transformer having no secondary winding, and the middle point of the winding maintains a constant potential while enabling the out-of-balance current to flow freely from this point to the armature, and thence through the armature conductors to whichever brush is at the time connected to the more heavily loaded wire of the distribution system. The slip rings receive an alternating current from the armature, its frequency being proportional to the speed of the armature multiplied by the number of pairs of poles, with the maximum voltage value the value of the voltage between the main commutator brushes. This voltage will provide only a small magnetizing current in the choking coil, which will be chiefly idle since the iron losses are very small, but the self inductance of the coil will be so high as to prevent any but a very small A.C. passing through it. At any instant, the voltage at the terminals of the coil will be the voltage between the rings at this instant, and the potential of the middle point of the coil will be midway between the potentials of the ends. Of course, large out-of-balance currents cannot be satisfactorily dealt with by the standard design of generator, and 20 to 25 per cent of normal full load is the heaviest out-of-balance current effectively dealt with. Greater out-of-balance currents require special designs, but there is no difficulty in manufacturing generators to give satisfaction for higher out-of-balance loads up to 33 per cent.—*H. J. D. Thompson, The Shipping World, 7th October 1953, Vol. 129; pp. 297-299.*

Aluminium Funnels

Aluminium is widely used for the outer casing of large funnels and, in fact, is rapidly becoming standard practice. Weight ratios of 0·45 are obtainable, which permit the prefabrication of very large units. The design of an aluminium funnel is fundamentally the same as that of a steel one. It is, however, of advantage to use specially designed sections, but particular attention must be paid to the elastic stability of compression members used for internal staying. The inner casing will generally be of steel, as will the internal machinery and equipment. Differential expansion must, therefore, be allowed for by suitably designed connexions, which must avoid direct bimetallic contacts. Where small funnels, such as for trawlers, are to be considered, it is, in general, advantageous to manufacture them from flanging quality material which is flanged at one end and lap riveted. This results in integral stiffening and is especially suitable where the temperatures involved are fairly high. Single funnels without inner casing are, in general, recommended, and there are some very successful examples afloat. There is no evidence that the material suffers in any way from the temperatures involved. As replacements on the older trawlers, single aluminium funnels present a very attractive case economically, due to the greatly reduced expenditure on repairs and painting.—*The Shipping World, 7th October 1953, Vol. 129; pp. 292-294.*

Torsional Vibration Dampers and Electric Slip Couplings

It has been suggested that some misapprehension may arise over the dual fitting of torsional vibration dampers and electric slip couplings to geared Diesel installations. In certain instances one device may be complementary to the other, but an electromagnetic coupling should possess, of course, the required elastic and damping characteristics to give the gearwheel teeth all the protection needed. With the B.T.-H. type, to quote an example, the makers undertake that the torque at the gearing will not become negative even should the engine run at any torsional critical speeds in the operating range. On the other hand, there may occur within this range criticals which represent orders of a dangerously high magnitude, with the certainty of crankshaft breakage if means are not adopted to damp out the effects. The trouble can be minimized, if not entirely eliminated by a torsional vibration damper secured to the crankshaft. This damper is immediately associated with the electric slip coupling, insofar as it takes care of the shafting up to and including the fixed half of the coupling which, with its highly elastic properties does no permit vibrations, small or large, to pass through the teeth of the gearing. The question of fitting the two types of device arose in connexion with the geared Diesel machinery of the *Middlesex* class of motor cargo liner, in which Holset dampers are arranged on each crankshaft and B.T.-H. electric slip couplings transmit the power to the tailshaft through reduction gearing. The electric couplings also serve the important purpose of allowing one engine to be stopped while the other continues to propel the ship.—*The Motor Ship, October 1953, Vol. 34; p. 299.*

Large Ferry

One of the most remarkable motor vessels now under construction is a twin-screw passenger and car ferry intended for service between Yarmouth, Nova Scotia, and Bar Harbour, Maine, U.S.A. She is being constructed by the Davie Shipbuilding and Repairing Co., Ltd., Lauzon, Levis, Quebec, and the keel was laid on 30th July. The ferry is for the Department of Transport and will be operated by the Canadian National Railways. Delivery will be taken in the autumn of 1954. There are to be two automobile decks, each having a capacity of seventy-five motor cars, and the lower motor car space has increased 'tween deck height to give headroom for the carriage of heavy trailers. The main particulars are:—

Length o.a.	345ft. 10in.
Length b.p.	320ft. 0in.
Breadth, moulded at main deck	65ft. 0in.
Breadth, moulded at l.w.l. ...	57ft. 0in.
Depth, moulded to main deck	22ft. 0in.
Depth, moulded to mezzanine deck	30ft. 0in.
Depth, moulded to upper deck	37ft. 6in.
Depth, moulded to promenade deck	45ft. 6in.
Draught, loaded	16ft. 6in.
Number of automobiles ...	150
Number of passengers... ...	600
B.H.P. total	12,000
Service speed	$18\frac{1}{2}$ knots

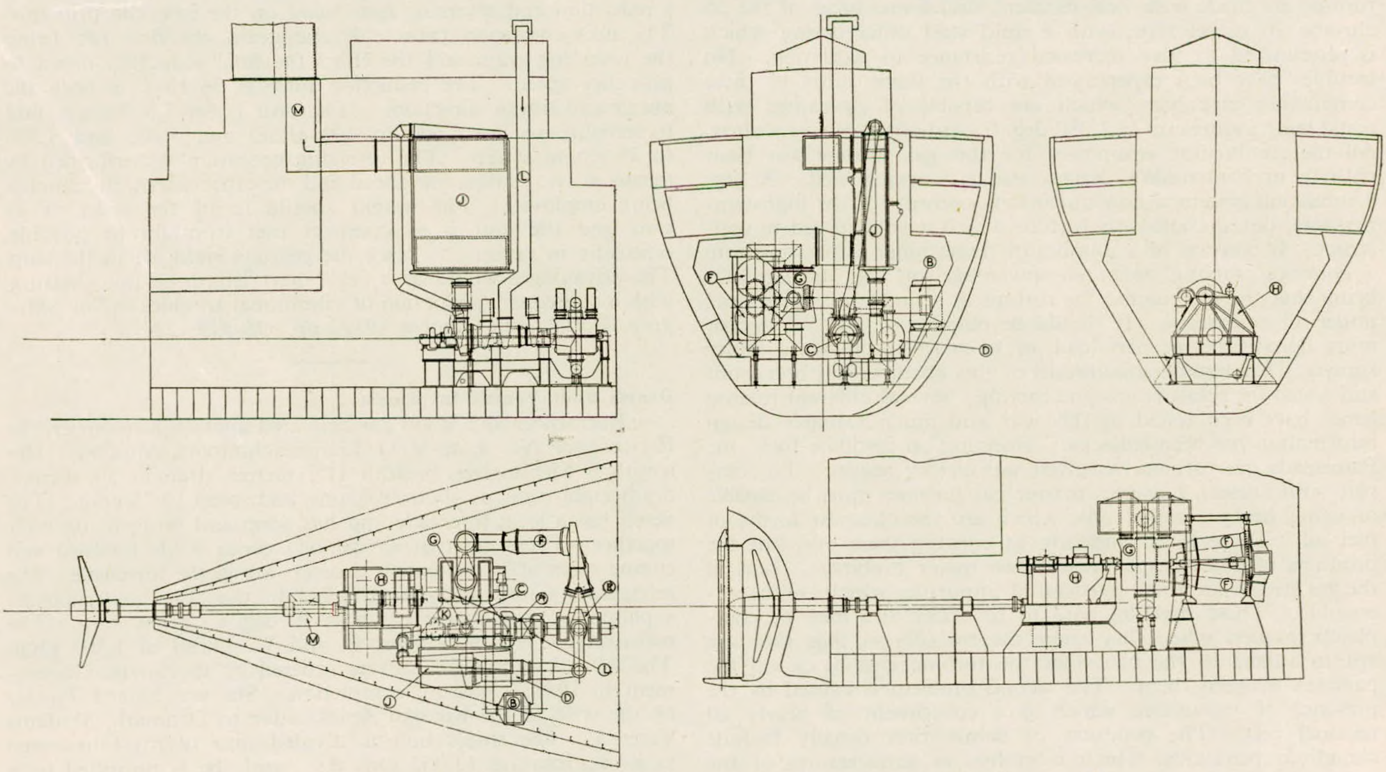
The machinery consists of two sets of opposed-piston engines with hydraulic couplings and reverse-reduction gear. Each group comprises three 12-cylinder Canadian Fairbanks, Morse units, with a continuous rated output of 2,000 b.h.p. at 750 r.p.m. Each set of three is connected through hydraulic couplings to a Modern Wheel Drive reverse-reduction gear, giving a total s.h.p. of about 5,580 at 200 r.p.m. The engines are of a type of which large numbers were built during the war for United States submarines. The diameter of the cylinders is $8\frac{1}{2}$ inches and the piston stroke 10 inches. There are two crankshafts driven respectively by the upper and lower pistons and these are connected by a vertical shaft and gearing. This vertical shaft is made up of three sections; the upper and lower pinion shaft and the flexible intermediate shaft. The two spiral impellers of the scavenging blower are driven by flexible gear from the upper crankshaft. The overall length of the engine is 27ft. 2in. and the width 6ft. 8in.—*The Motor Ship, October 1953, Vol. 34; p. 294.*

Viscous Drag of Bodies of Revolution

A completely immersed body moving rectilinearly with uniform velocity in an infinite fluid at rest experiences a resistting force which may be termed viscous drag, as it results primarily from the viscous properties of the fluid. In designing low-drag bodies there often arises the need for calculating the viscous drag of streamlined bodies of revolution in axial motion when considering various proposed shapes. For bodies moving at high Reynolds' numbers, which are of great technical importance, the viscous drag of streamlined bodies of revolution is readily amenable to analytical treatment on the basis of the boundary layer concept. Historically, the theoretical analysis of the drag of bodies in uniform motion by assuming an ideal (non-viscous) fluid gave the fruitless result of zero drag for all bodies, the classical D'Alembert paradox. At the other extreme, the theoretical analysis of drag by applying the complete set of Navier-Stokes equations of motion for the flow of a viscous fluid led, in general, to mathematical difficulties which were virtually unresolvable owing to the complicated non-linear nature of these equations. For bodies moving at high Reynolds' numbers, however, the flow is virtually that of an ideal fluid except in a thin boundary layer next to the body where substantial viscous forces are produced by the rapid drop in velocity to zero at the body surface. Accordingly, by considering the viscous flow confined to the boundary layer, Prandtl was able to derive the simpler boundary-layer equations of motion from the Navier-Stokes equations. The principal purpose of this report is to describe methods of solving the boundary-layer equations of motion to arrive at the viscous drag of bodies of revolution of arbitrary shape in uniform axial motion. The study is restricted to hydraulically or aerodynamically smooth streamlined bodies in incompressible flow. A streamlined body may be defined as one without appreciable separation of flow from its surface and consequently with small pressure drag resulting from the generation of separation eddies. It is to be noted, however, that some pressure drag is still present in the viscous drag of even perfectly streamlined shapes owing to the effect of the boundary layer in displacing the main flow outward, especially near the tail. The calculation of the viscous drag of a body of revolution requires a detailed analysis of the development of each phase of the boundary layer from its origin on the nose of the body to its final phase as the frictional wake far downstream. In the downstream direction the boundary layer may consist successively of a laminar boundary layer, a transition zone from laminar to turbulent flow, a turbulent boundary layer, and a frictional wake. In the laminar boundary layer an approximate method involving a simple quadrature is derived for the rapid calculation of the changes in momentum. The derivation consists of an extension to axisymmetric flows past bodies of revolution of a method of successive approximation introduced by Shvets for two-dimensional laminar boundary layers. New empirical criteria are presented for locating the position of transition for either low-turbulence or turbulent free-streams from the position of neutral stability. The position of so-called self-excited transition occurring in low-turbulence streams or under flight conditions is based on the average pressure gradient from the position of neutral stability to that of transition. Although the test data are for two-dimensional flows, the criterion is extended to axisymmetric flows past bodies of revolution by means of Mangler's transformations. An approximate criterion for estimating the position of transition on a body in a turbulent free-stream is established on the basis of measured positions of transition for flat plates in free streams with various degrees of turbulence. The analysis of the axisymmetric turbulent boundary layer on a body of revolution is divided into that for the main portion of the body, where the boundary layer is relatively thin compared to the radius of the body, and that for the tail portion, where the boundary layer is relatively thick. The momentum changes in the thin boundary layer may be calculated by a rapid method involving a simple quadrature wherein a power-law relation for the skin friction of flat plates is incorporated. Flat-plate values for skin friction are deemed reliable even where the local skin friction is diminishing in an adverse pressure gradient owing to the compensating effect of the Reynolds' normal-stress term. The turbulent boundary layer on the tail is analyzed by means of appropriate linear simplifications which give a rapid method for calculating the momentum changes. An expression is derived for the change in momentum produced by the pressure difference in the wake at the tail and in the wake far downstream in accordance with a method presented by Yound wherein, however, a more general relationship is employed for the variation of the shape parameter of the velocity profile. For quick reference the various steps involved in calculating the development of the boundary layer on a body of revolution are summarized at the end of the report.—*P. S. Granville, Navy Department (U.S.A.), The David W. Taylor Model Basin; Report 849; July 1953.*

Pametrada Steam and Gas Turbine Research

The 3,500 h.p. gas turbine which Pametrada has designed and which has run so satisfactorily could itself be installed in a ship; but it is considered that the knowledge and experience gained by the association should be applied to the construction of new units of improved design. At present the turbine is being developed to burn residual fuels. These are much cheaper, but their use presents considerable problems, which are being investigated. The following items are included in the



Layout of Pametrada 3,500 s.h.p. marine gas turbine installation

A.—Low pressure compressor. B.—Intercooler. C.—High pressure compressor. D.—High pressure combustion chamber. E.—High pressure turbine. F.—Low pressure combustion chamber. G.—Output turbine. H.—Double reduction gear. J.—Heat interchanger. K.—Starting turbine. L.—Air inlet to low pressure compressor. M.—Exhaust duct to funnel.

research programme. The Pametrada 3,500 s.h.p. marine gas turbine is among the first gas turbines designed for use in a ship and to drive the propeller directly. This turbine has been developed to a point where it can be regarded as a working proposition for a ship, but before gas turbines are accepted by the shipping industry it has to be shown that they offer economic advantages because they require fewer auxiliaries, occupy less space, and have a lower fuel consumption than comparable steam turbines. Last year the 3,500 s.h.p. gas turbine completed a continuous run of 100 hours at full power, when it was completely opened up and inspected by Lloyd's Register, who produced a very satisfactory report. The test was carried out on distillate fuel, a fuel consumption of 0·505lb. per b.h.p. per hr. being recorded. Moreover, it has operated at a specific consumption of 0·491lb. per s.h.p. per hour. To compete with Diesels, however, marine gas turbines must be capable of running on residual oils, and difficulties are presented because these oils contain impurities which cause corrosion and fouling of the turbine blades. For instance, trouble was experienced with distortion of the h.p. and l.p. turbines at high temperatures. This was largely overcome by modifying the flanges and altering the method of support of the cylinders. The behaviour of the turbine casings is still not satisfactory under conditions of rapidly changing temperatures, but a new method of supporting the casing has been developed, which should overcome the remaining distortion problems. Further modifications are being made to a reversing coupling which has been developed at Pametrada. When the set is running again, manœuvring trials will be carried out to determine the behaviour of the set and of the hydraulic transmission under rapidly changing conditions. When these trials are completed, further work will be carried out on the problems attending the burning of residual oils. The marine gas turbines at present being tested are constructed from austenitic stainless steels, and both open cycle and closed types are being investigated. The combustion chambers of the Pametrada 3,500 s.h.p. marine gas turbine are made with heat resistant steel flame tubes of the 25 chrome 20 nickel type, with a mild steel outer casing which is aluminized to give increased resistance to oxidation. No troubles have been experienced with the flame tubes of these combustion chambers, which are capable of operating with metal temperatures up to 1,050 deg. C. without excessive scaling. All the combustion equipment for this gas turbine was built entirely to Pametrada's design, and is working well. A new combustion system is now under development for the high-temperature, liquid-cooled gas turbine which it is proposed to construct. It consists of a number of flame tubes exhausting into a common mixing zone, an advantage of this arrangement being that the gas reaches the turbine at a uniform temperature under all conditions. It should be possible to shut off one or more flame tubes at part load, or in order to change a faulty sprayer. An aerodynamic model of this chamber has been built and tested for pressure loss and mixing. Several different mixing zones have been tested in this way and much valuable design information has been collected. Running on distillate fuels, the Pametrada gas turbine has given satisfactory results. To compete with Diesels, however, marine gas turbines must be capable of using heavy residual oils, which are the cheapest forms of fuel oil. There is no difficulty in burning these oils, but the products of combustion present two major problems. Left in the gas stream are small particles of impurities which are incombustible. These particles tend to be sticky and may be completely molten when they enter the turbine, so that they are apt to adhere to the blades of the turbine and block up the passages between them. The second problem is caused by the presence of vanadium, which is a constituent of nearly all residual oils. The products of combustion usually include vanadium pentoxide, which is molten at temperatures of the order of 650 deg. C. Vanadium pentoxide when molten is highly corrosive and will attack all commercially available heat resisting alloys; in fact, no metal appears to be completely immune. The rate of attack is very high and increases progressively with temperature. Pametrada and other research organizations are endeavouring to solve the corrosion and fouling problems, the main line of attack being the addition of compounds to the fuel, or to the gas stream at the combustion chamber, that would have the effect of rendering the ash non-sticking and non-corrosive. A certain amount of success has been obtained with additives of the basic metal oxide type; e.g. zinc oxide, calcium oxide, magnesium oxide, etc. These additives reduce corrosion to a considerable extent, but they have not provided a solution to the fouling problem, which is being attacked on other lines. There is every indication that before long these problems will be solved. To provide experience with high temperature steam in conjunction with high pressure, an experimental unit known as *Pamela* has been designed. Particular attention will be paid to the performance of the special materials used. The investigations will also include the effect of thermal stressing and distortion of the austenitic steel casing under changing temperature conditions. *Pamela* is a six-stage impulse turbine, which represents the h.p. cylinder of a 3-cylinder turbine developing about 3,000 h.p. The maximum conditions are 1,100 deg. F. and 1,100lb. per sq. in. The power is absorbed by a steam brake which uses steam exhausted from the driving turbine and consists of a two-row astern turbine. Power is measured by a torque meter situated between the two turbines. The steam temperature leaving the brake will be of the same order as that at the inlet to the driving turbine. In order to lower it sufficiently to allow it to be condensed in a surface condenser, a desuperheater, consisting of six water sprayers, is incorporated in the exhaust line. A boiler using normal residual oil delivers steam at about 1,200lb. per sq. in. and 950 deg. F. The steam is superheated to 1,100 deg. F. by a separately fired second-stage superheater burning gas oil. The use of a reversing gear offers important advantages in ships which are propelled by steam turbines, since it avoids the necessity for an astern turbine. Pametrada is, therefore, designing a reduction and reversing gear based on the epicyclic principle. The unit comprises three epicyclic gears, the first two being the reversing train and the third the final reduction down to propeller speed. The reduction ratio is 36 to 1 in both the ahead and astern directions. The unit is for 7,500 s.h.p. and its revolutions are 3,600 to 100 ahead and 5,500 and 3,200 to 89 r.p.m. astern. The reversing operation is performed by means of two brakes, one ahead and the other astern, no clutches being employed. The weight should be of the order of 25 tons and the unit is so compact that it might be possible, especially in tankers, to place the gearbox right aft in the ship. The advantages would be a very short length of line shafting, with a consequent reduction of vibrational troubles.—*The Shipping World, 14th October 1953; pp. 318-319; 321.*

Danish Built Vessels for Russia

Burmeister and Wain has delivered another fish carrier, the *Refrigerator No. 8*, to V/O Transmachimport, Moscow. Her length is 65·5 metres, breadth 11·5 metres, draught 5·8 metres, deadweight capacity about 900 tons, and speed 10·75 knots. The vessel has a long forecastle and has poop and bridgehouse built together. There are two refrigerated cargo holds forward and engine room aft. The crew's quarters are in the forecastle. The refrigerating machinery is installed in the poop and includes a plant for freezing the catch at 30 deg. C. below zero. The main engine is a 6-cylinder B. and W. Diesel of 1,300 i.h.p. The second of two oil tankers ordered by the Soviet Government in 1948 is nearing completion. She was named *Tuapse* by the wife of the Russian Ambassador to Denmark, Madame Vetrova. The ship's hull is divided into twenty-four cargo tanks aggregating 13,000 tons d.w., and she is propelled by a 6-cylinder two-stroke single-acting B. and W. Diesel of 6,900 i.h.p. calculated to give a speed of 14·5 knots.—*The Shipping World, 14th October 1953, Vol. 129; pp. 309-310.*