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The Care and Maintenance of a Modern Diesel-engined Tanker Fleet.

READ
By H. S. HUMPHREYS (Vice-Chairman of Council).

On Tuesday, January 14th, 1936, at 6 p.m.

CHAIRMAN: Mr. J. J. McKENZIE (Vice-President).

Synopsis

THE paper has been divided into two parts, one outlining tanker construction and dealing with those special features arising from the carriage of a liquid, often a dangerous liquid, and the other dealing with the main features of Diesel engine maintenance.

The special features include safety precautions, the methods of dealing with unstable liquids of varying volume, the loading and discharging system, the care of the tanks, the various methods of cleaning and the particular problems of corrosion. The second part, dealing with Diesel engines, is a little more detailed. The important features have been described mainly from two broad aspects, that of trend of design and that of practice in care and upkeep.

A modern vessel is one that is best adapted for carrying its intended cargoes and it is proposed, therefore, to outline what is considered to be the best construction for the carriage of liquid cargoes before dealing with motive power.

Liquid cargoes carried include various products of petroleum, creosote, whale oil, turpentine, vegetable oils and molasses, in fact any liquid that is suitable for carriage in bulk, varying from the lightest fractions of petroleum, which are called casinghead products, having a specific gravity as low as ·65, to creosote of specific gravity 1·10 to 1·20, and in special cases to beet molasses and sugar molasses with a specific gravity of about 1·20 and 1·45 respectively. There are several characteristics inherent in these liquids which must be studied in order to obtain satisfactory carriage.

Liquids vary greatly in volume with change of temperature, particularly in the case of petroleum products and they are also unstable. The first vital factor, therefore, is a design which will allow safely for changes in volume and will permit of loading (including in particular part loading) without leaving large masses of liquid to surge and cause damage.

As other than liquid cargoes cannot be carried commercially, there must normally be many ballast passages and construction must, therefore, allow for safe ballasting, particularly with such cargoes as molasses, which is heavier than water and must on no account have had contact with water in the cargo tanks.

With liquid cargoes the particular difficulties are associated with the prevention of leaks and mixtures during storage and handling. This involves consideration of piping systems, valves, gas lines and cofferdam separation. The dangerous nature of some of the liquids necessitates special attention to fire prevention and fire fighting apparatus. Crude oil must always be treated as a dangerous cargo owing to its gassy nature and its low flash point; it will, when mixed with air, explode at any normal temperature in contact with a spark. Crude oil may vary in specific gravity from ·735 (Sumatra) to ·950 (heavy Mexican). For vessels carrying creosote and viscous oils, cargo heating coils are fitted in the tanks for warming the cargo, thereby reducing the viscosity when discharging.

In a modern tanker careful consideration is usually given to the crew's accommodation, which is well planned with ample space, privacy and comfort. All the essential aids to navigation are installed, including wireless direction finder, auto alarm, and in many cases an echo depth sounding device.

General Construction.

Tankers are no longer built with circular tanks or with machinery amidships. A modern tanker is divided longitudinally into compartments, the sequence being as shown in Fig. 1, viz.:—

- Fore peak tank.
- Fore hold with deep tank below.
- Fore cofferdam.
- Cargo tanks, with one or more pumprooms between tanks.
- Aft cofferdam.
- Cross bunkers.
- Boiler and machinery space.
- Aft peak tank.

Many modern tankers have a cruiser stern which gives greater buoyancy, less tendency to "poop", better water line and run, large engine room space and roomy accommodation. Some form of streamlined rudder is also frequently fitted. The majority of tankers are built on the longitudinal

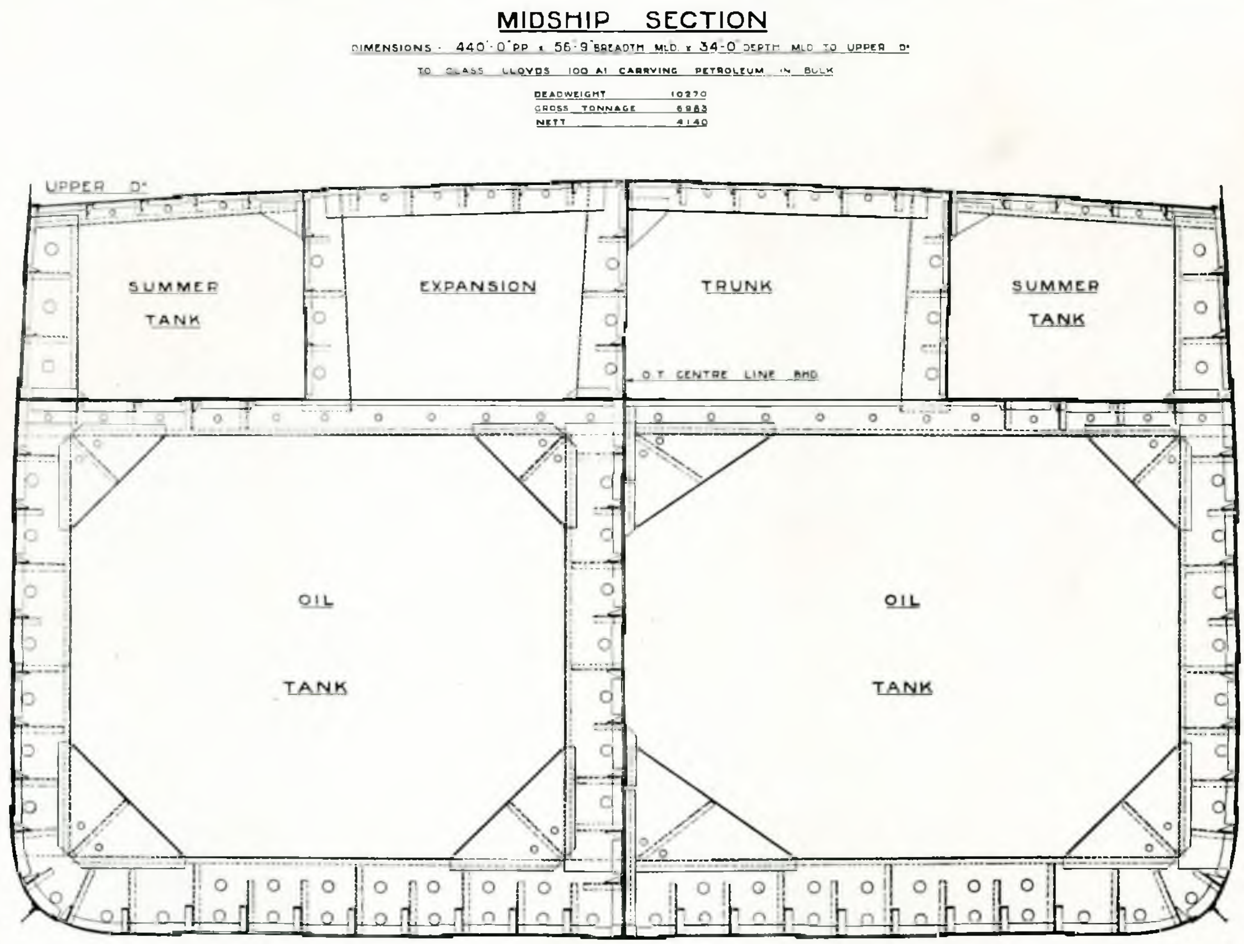
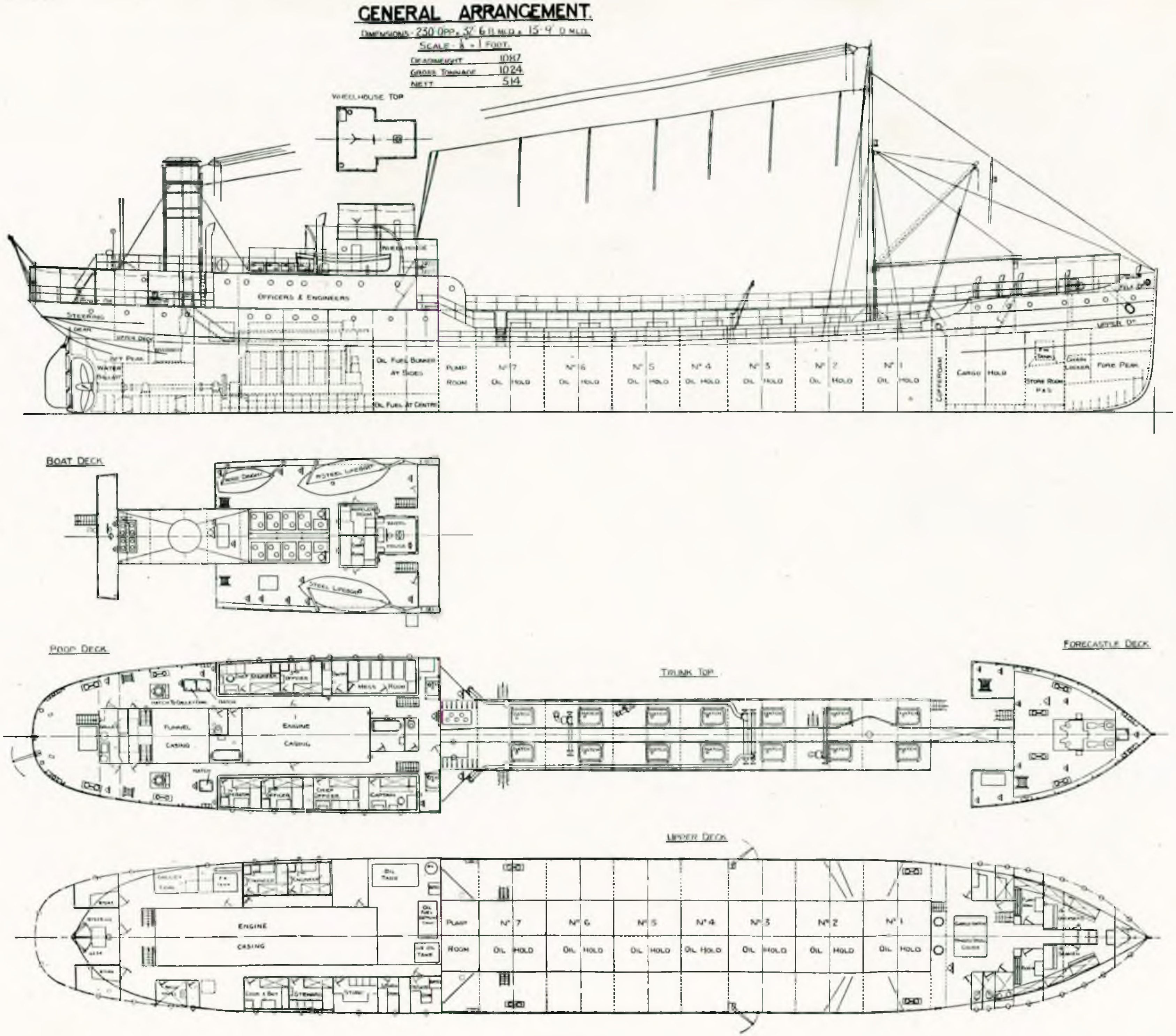


Fig. 2.



GENERAL ARRANGEMENT.
Dimensions 250' 0" BP × 32' 6" B MLD × 15' 9" D MLD
Scale 1/8" = 1 Foot.
Deadweight 1087
Gross Tonnage 1024
Nett 514
Wheelhouse Top
Wheelhouse
Officers & Engineers
Steering Gear
Upper Deck
Aft Peak
Water Ballast
Oil Fuel Bunker at Sides
Oil Fuel at Centre
Pump Room
No 7 Oil Hold
No 6 Oil Hold
No 5 Oil Hold
No 4 Oil Hold
No 3 Oil Hold
No 2 Oil Hold
No 1 Oil Hold
Cofferdam
Cargo Hold
Store Room P&S
Chain Locker
Fore Peak
Boat Deck
Steel Lifeboat
Poop Deck
Funnel Casing
Engine Casing
Mess Room
Chief Officer
Captain
Trunk Top
Forecastle Deck
Upper Deck
Engine Casing
Oil Tank
Galley

Fig. 3.

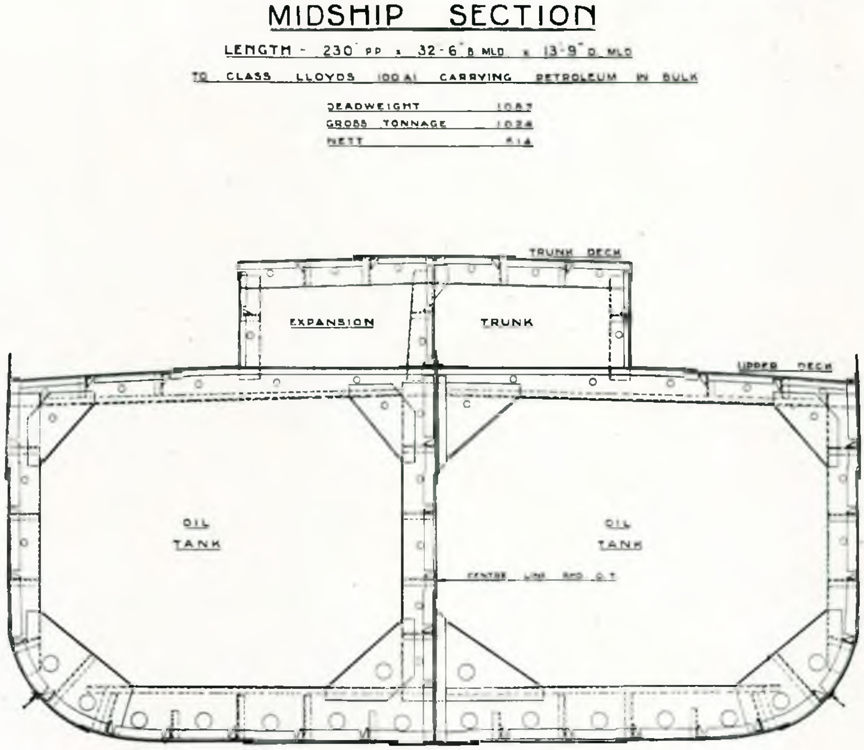


Fig. 4.

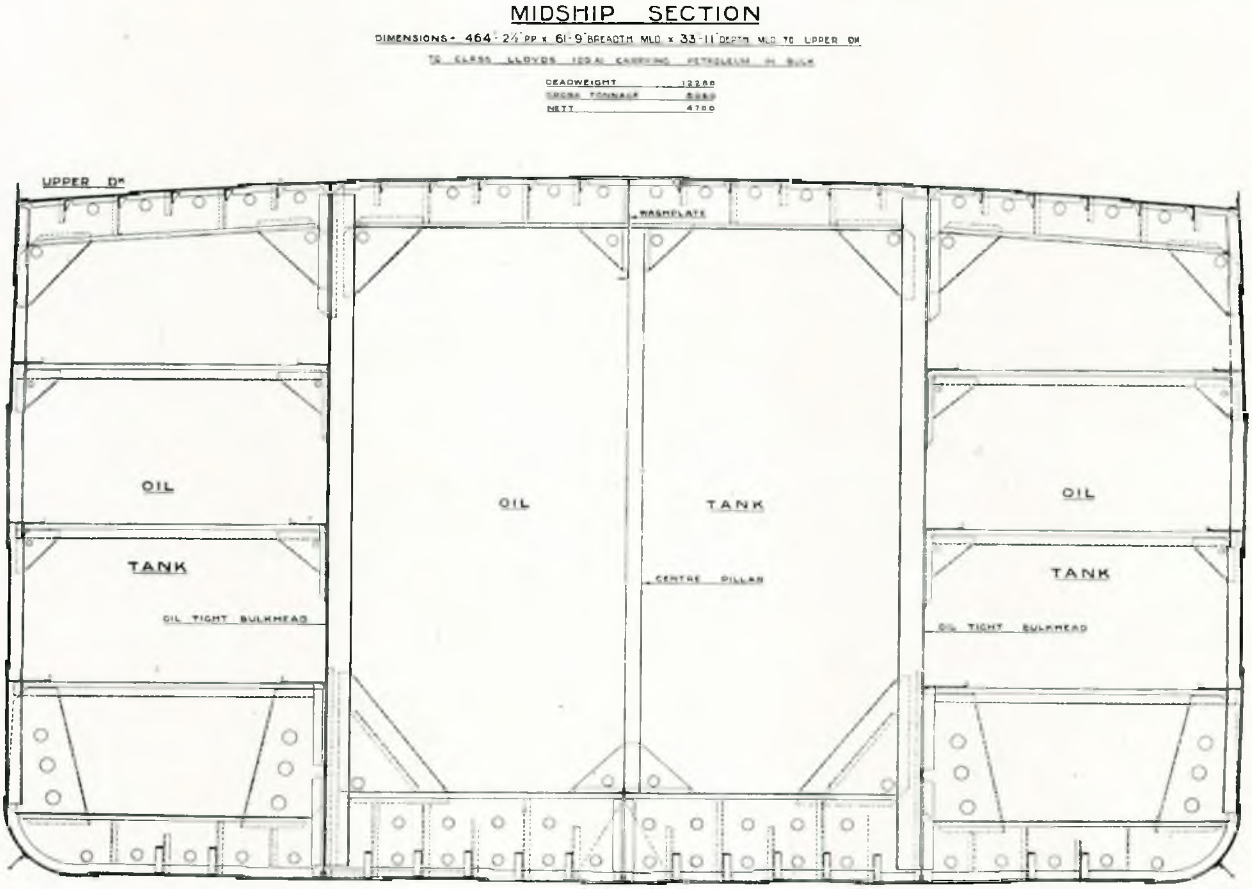


Fig. 5.

system of framing, except at the ends where transverse framing is used.

Until recent years the centre line longitudinal bulkhead tanker, with summer tanks, was the standard type. Fig. 2 (midship section). The midship section shows the summer tanks, port and starboard, which run longitudinally between the main and upper decks and extend from the ship's side to the side of the expansion trunk of the main tanks. It is necessary that sufficient cargo be loaded into the main tanks in order to prevent the surface of the oil from falling below the bottom of the trunk when the vessel is rolling. In arriving at this quantity of cargo due allowance must be made for temperature, since the change in volume, due to change in temperature, is considerable. When loading cargo at 60° F.

a space equal to about 2 per cent. of the capacity of the tank is generally left empty to allow for the expansion of the oil due to temperature rise.

A typical general arrangement of a coasting tanker is shown in Fig. 3 fitted with a trunk but no summer tanks. The deep trunk provides greater cargo capacity, which is very useful for the carriage of light cargoes, in addition to its duty of reducing the free surface of the liquid. The midship section of this vessel is shown in Fig. 4.

During recent years many tankers have been built with two longitudinal bulkheads and no summer tanks, the bulkheads being spaced as shown in Fig. 5 (midship section) each wing tank having about half the capacity of the centre tank, and this arrangement in some cases is modified by the two longitudinal bulkheads being so fitted that the wing tanks are extremely narrow, with an expansion trunk over the centre tanks. In this type the framing is longitudinal on the bottom and under deck, but a system of vertical framing is frequently adopted on the longitudinal bulkheads and ship's sides.

In the case of the centre line longitudinal bulkhead vessel, the summer tanks are mostly used when light cargoes are carried or to avoid slack main tanks; more flexibility, however, can be obtained with the two longitudinal bulkhead type, as the centre tanks can be filled and the wing tanks loaded to any desired ullage. Fig. 6 represents outline midship sections showing the liquid in the tanks when the different type vessels are listed.

There are, of course, tankers built for special purposes. For example, molasses carriers are constructed with tanks alternately long and short, the long tanks for the carriage of molasses only and never for water ballast, and the short tanks for the carriage of water only. The heavy weights are thus distributed throughout the vessel. Salt water is never allowed in the molasses tanks since this would cause fermentation in the molasses and spoil the cargo. Tankers for the carriage of lubricating oil may be constructed with a deep double-bottom tank throughout the length of the tanks for the carriage of water ballast to avoid the use of sea water in the cargo tanks when in ballast condition.

As, however, the object of this paper is to deal with care and maintenance and not design, it is felt that sufficient has been said with regard to modern construction of the hull.

Fire Extinguishing Arrangements.

A well organised modern tanker fleet will have certain regulations, which the crews will be trained to observe strictly on the basis that "prevention is better than cure". The Author believes that discipline applied with reason and consideration is a greater safeguard against fire than elaborate systems of chemical fire extinguishers, CO_2 gas or foam. Nevertheless, the modern tanker is fitted with efficient fire extinguishers in requisite places, and these are systematically used and refilled. All

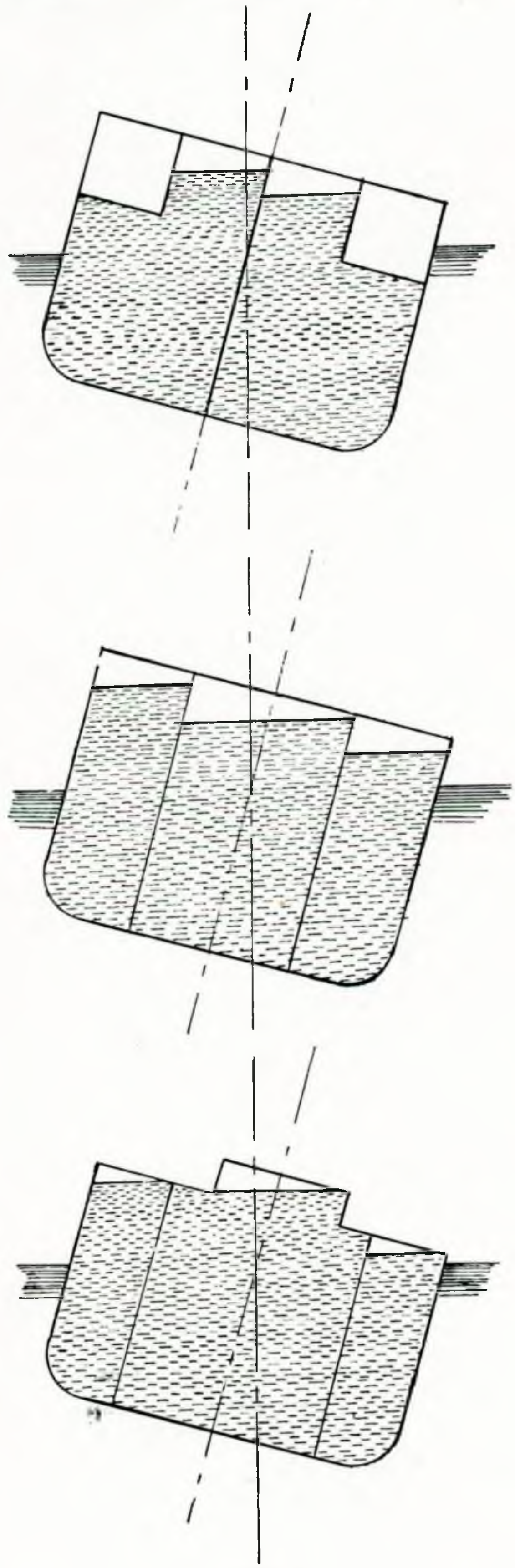


Fig. 6.

cargo tanks have steaming out connections which can be quickly brought into use in case of emergency. The boiler room is fitted with steam and water extinguishers which can be controlled from deck.

Ventilation.

Gas ejectors are fitted in each tank, cofferdam and pumproom. At the hatch of each tank a vapour valve is fitted, connected to one or more vapour pipe lines, which are led up the masts to not less than 12ft. above the masthead lights, each line being provided with a gooseneck bend with monel metal gauze flame arrester at the outlet. "U" water gauges fitted in the wheelhouse are connected to the vapour lines and the whole system is hermetically sealed. Any pressure or vacuum in the tanks can be observed at once from the "U" gauges and readily rectified. The vapour lines must be kept clear, particularly in vessels carrying gaseous crude oil, otherwise they will become choked due to the moisture from the crude oil and air causing iron rust. Hydrogen sulphide from the oil passing over the iron rust forms sulphur and iron sulphide; the latter is again oxidised by air forming more iron rust. These lines are periodically cleaned according to the nature of the cargo. Through this system, losses by evaporation are reduced to a minimum. When different grades of clean cargo are carried on the same voyage the vapour lines from the tanks containing the different grades are isolated from each other to ensure that no condensation from the gases of one grade may mix with another grade and affect the flash point.

Pipe Lines and Valves.

The tanker may be required to carry various grades of oil in the one cargo; therefore great care must be taken to avoid admixture. The cargo pumps, pipe lines, valves, bulkheads, etc. must be kept in efficient condition. Gate valves are fitted which have small access doors so that the lines can be flushed through and valves examined after discharge to see that they are all tight and shut properly. Stuffing box expansion glands, which are generally packed with lead shavings, must be kept tight, otherwise difficulty will be experienced in draining tanks through the pumps drawing air. The pipe lines are found to corrode and erode along the bottom internally, and it is good practice to turn these, say, one-half turn after three or four years—according to the cargo carried—thereby lengthening the life of the pipe line. Fig. 7 shows an arrangement of pipe lines in the cargo tanks of a modern tanker fitted with four cargo pumps required for carrying several grades of refined products. Each section of tanks can be completely isolated, each section having its own pump and pipe line. The emergency pipe line is also shown.

Cleaning Cargo Tanks.

Different standards of cleanliness are required according to the nature of the cargo to be carried. A "clean" tanker is one engaged in the carriage of clean oils such as benzine, kerosene or lubricating oils and the tanks must be absolutely clean and dry before commencing to load. A "dirty" tanker is one engaged in the carriage of black oils such as crude, fuel or creosote.

During the ballast passage, the empty tanks are steamed and washed down with the hose, all sediment and scale being removed. Various processes are carried out to clean tanks, according to the nature of the future cargoes to be carried. If steaming the tanks for six hours and washing is not sufficient to remove the dirt, then an open-ended perforated iron drum filled with caustic soda is suspended in the tank, and a steam hose is put through the ullage plughole and the end inserted in the drum and steam turned on. The caustic soda evaporates and penetrates into all corners and pockets. After, say, six hours' steaming, the tanks are washed down with hot water from the hose.

Another method employed is the Butterworth system* of tank cleaning which, it is claimed, eliminates steaming. With this system the tanks are washed by water heated to about 175° F. and directed against all internal surfaces of the tanks in a solid stream at 160 to 180lb. pressure per sq. inch from revolving nozzles, the nozzles turning slowly in horizontal and perpendicular planes, so that their positions change continually with each revolution. The nozzles are part of a machine that is lowered down through an opening in the deck over the tank and reaches down to about one-half the depth of the tank.

To convert a "dirty" tanker to a "clean" tanker is a costly business. After cleaning by one of the above methods, it may be necessary to pump gas oil or kerosene from one tank to another through the various pipe lines before the tanks are certified fit for clean cargoes.

Loading and Discharging.

The tanks must first be examined and passed by the inspector to see that they are dry and clear of sediment. All fires must be extinguished before commencing to load benzine or crude, steam being supplied from shore to the ship's steam lines. Upon completion of loading, the ullages, temperature and samples must be taken and the tank plugs screwed down.

Vessels passing through the Suez Canal loaded with low flash point oil must be in such a trim that they can be lightened by at least 20 inches mean draft in the event of the vessel grounding. The cofferdams, peaks, etc. are therefore filled with water before entering the Canal. Anchors, cables and fuel oil are allowed by the Authorities to be calculated as movable weights in emergency. Before discharging, the installation representative will

*See paper by Robert F. Hand read before N. E. C. Institution of Engineers & Shipbuilders, 5.4.35, entitled "Oil Transportation of Petroleum in Bulk".

come on board and take ullages, temperatures and samples and search for water. It should be noted that the water will lie at the top of the tank in creosote cargoes and not at the bottom as in petroleum cargoes. The flexible hoses will be connected and the cargo pumps started slowly at first to see that all joints, valves, etc. are in order and no undue pressure is on the pipe lines.

It is a convenience in modern tankers to fit floodlights on the masts, thus avoiding portable lights near the tanks.

While loading, as many valves as practicable should be opened to wash the dirt and scale clear of valve faces and then shut, but while discharging the minimum number of valves should be opened.

Periodical Hull Examination.

Although more frequent attention is required in tankers to avoid leakages, it is not proposed to deal with drydockings and usual examinations, which are common practice with any other type of vessel. The rules regarding tank testing to classification societies' requirements are too well-known to repeat in this paper, but in a tanker great care must be taken to see that the vessel is free from gas and in a fit condition for men to work in the tanks, cofferdams, pumprooms or any space where repairs are necessary. According to the nature of the cargo, the tanks are washed down, steamed and drained. Some repair yards are equipped with separator barges or sludge tanks for taking the drainings from the tanks; the Wheeler system may be used or, in certain cases, one of the ship's tanks or cofferdams may be used as a dump tank during repairs. When steam is used for gas-freeing purposes, each tank is steamed for about six hours, after which the tank lids are opened, the tanks are drained, and a windsail is placed in each tank. After the tanks are properly ventilated, an analytical chemist tests the atmosphere in the tanks, and when satisfied he will issue a gas-free certificate. When it is necessary for emergency work to be done in a tank or pumproom which is not free from gas, no man is allowed to enter without a gas mask and life-line. The gas masks, of course, must always be kept in perfect condition on board. No chipping must be done or naked lights used before the vessel is free from gas; special copper hammers and non-ferrous tools are used in emergency to prevent sparks.

On ballast passages, the crew is employed washing and cleaning tanks, and the engine-room staff overhaul the cargo pumps, and examine bulkheads and tank valves to ensure that no leakage will take place in the event of different grades of cargo being carried.

Corrosion in Cargo Tanks.

Corrosion, or the wasting away of the internal structure of cargo tanks, is an important factor in the maintenance and operation of tankers.

It has been found that tankers carrying fuel oil cargoes have practically no trouble in the cargo tanks due to corrosion.

Of the various grades of petroleum, there are some classes of crude which also do not cause any trouble. Where corrosion is caused, it may be due to a combination of factors—the nature of the cargo, its combination with sea-water ballast and salt-laden atmosphere, particularly river and harbour water containing injurious chemicals, or to temperature and corrosion-fatigue. It may be increased in those parts where the resisting qualities of the steel have been reduced by punching, shearing and riveting, e.g. in way of brackets, framing and stiffeners where the nominal stress is considerably increased. The nature of the cargo chiefly affects the location of corrosion. With some crude oils pitting may take place at the bottom of the tanks and on horizontal surfaces, but with very little corrosion on bulkheads and vertical surfaces; with benzine types practically no pitting may occur on the bottom yet corrosion may attack the bulkheads, particularly the upper surfaces.

With regard to the pitting on the bottom and horizontal surfaces in some crude carriers, this is not so serious as the corrosion sometimes found in benzine carriers. After discharge the crude residue, which may contain free sulphur, settles on these surfaces and mixes with the sea-water ballast or salt-laden sweat from the tanks, the uneven pitting being caused by sulphuric acid and hydrochloric acid. Added life may be given to the bottom shell plating where pitted by thoroughly cleaning with caustic soda, and cement washing and cementing the bottom internally where affected.

In vessels employed for the carriage of benzine, extensive repairs to the tanks may be necessary after say eight to ten years' service. The most severe corrosion is found at the upper portion of the bulkheads and at the expansion trunks. The upper structure wastes over its entire surface due to the presence of petroleum gases in the upper portion of the tank mixed with air and fumes from the stagnant sea-water ballast forming a corrosive gas, the cleansing action of the benzine cargoes on the structure assisting corrosion. The rate of corrosion increases as the structure gets thinner, this being due, no doubt, to fatigue of the metal; the plates become embrittled and cracks appear. This fatigue is due to whipping or reversal of stress and the stresses imposed upon the vessel by the different sea conditions and by the conditions of loading and ballasting. The rate of corrosion will depend on the number of cargoes carried alternating with the steaming out of tanks—with consequent stresses set up—and ballasting. For this reason, coastal benzine carriers are found to corrode more quickly than vessels on longer voyages.

The rate of corrosion can be greatly reduced by the periodical removal of scale from the tanks behind which corrosion is accentuated. The surfaces, when thoroughly cleaned, should be sprayed under pressure with a light lubricating oil, impervious to water, which will not discolour or affect the benzine cargo. After scaling the tanks of a 10,000-ton benzine carrier as much as 100 tons of scale has been removed; this, incidentally, is important from the point of view of carrying capacity.

Corrosion being cumulative as the structure weakens, it is considered good practice to increase the thickness of the plating and structure most commonly affected, above normal stress requirements. The use of special steels in construction, special paints and preservatives do not appear to have solved or even partly solved this problem. Wastage can be reduced by keeping all stresses down, by the addition of a chemical neutraliser or inhibitor to the cargo or water ballast, by the minimum use practicable of steaming-out tanks and by avoiding, as far as possible, the use of ballast water taken from rivers and waterways which may contain injurious acids and chemicals.

Sea and Repair Days.

An important factor, which must always be borne in mind by those responsible for the care and maintenance of a tanker fleet, is the number of days per year at sea which is expected of this type of vessel. It is quite usual for a tanker to go alongside a wharf and discharge, say, 10,000 tons of cargo, bunker, ballast, take in stores and be away to sea again in about 48 hours. In consequence of this a tanker averages about 300 steaming days per year against some 220 for ordinary cargo liners. It is expected that all repairs, including drydockings and periodical surveys, will be done on an average not exceeding 30 days per year and to achieve this it is of the utmost importance that systematic routine examinations and orderliness should be adhered to and accurate records kept of all parts liable to wear or corrosion. Steady upkeep is better than a lower irregular standard which, while it may result in less immediate expense, is liable to cause voyage delays and occasional big repair delays. It must be remembered that a tanker has very little loading and discharging time in which to do repairs and that this time is further restricted when dangerous gases are present —a cargo liner's berth periods are not only much longer but they do not interfere with repair possibilities in the same way.

Machinery.

A wide choice of machinery is available; the types most favoured are the opposed-piston (Doxford type), the two-stroke cycle single-acting, the two-stroke cycle double-acting and the four-stroke cycle single-acting, the latter type with or without supercharge. Under piston supercharge is often used in the 4 S.C.S.A. type, and supercharging by exhaust gas turbine or by rotary blowers may also be used.

In each type airless injection is now becoming accepted practice, thus dispensing with the main air compressor and thereby increasing the mechanical efficiency of the engine, and in consequence the fuel consumption is reduced.

The four-stroke cycle double-acting type is not included on account of inaccessibility for quick overhaul, and increased maintenance and the large choice available from lighter, less costly and simpler machinery.

It would be impossible to deal adequately with each type in the course of one paper, but many features concerning care and upkeep are applicable to all types. It is, therefore, proposed to deal with the common features and to illustrate some of the latest ones in a few well-known types developed largely as the result of experience gained during maintenance.

For specially high-powered tankers, twin-screw machinery may be necessary High speeds are expensive; the higher the speed, for a given size of vessel, the smaller is the carrying capacity by reason of the finer lines of the hull, the increased bunker capacity and the increased steaming weight of machinery.

Some average sizes and powers of modern Diesel tankers are as follows:—

TABLE I.

	Deadweight.	Average service speed (loaded).	Average service power.	Average daily consumption.
1.	10,000 tons.	10 knots.	1,850 b.h.p.	8 tons.
2.	11,000 „	11 „	2,350 „	10 „
3.	12,000 „	12 „	3,000 „	12·5 „

In ballast condition the speed will be about 1 to 1¼ knots faster for the same power output.

The consumption quoted does not include fuel used for special purposes, such as the supply of steam for cargo heating purposes where necessary, but is for all steaming purposes at sea and assuming exhaust gases are utilised in an auxiliary boiler for providing steam to the steering gear, dynamo, accommodation heating, etc. The consumption figures will vary slightly according to the type of machinery fitted and the speed may vary due to hull form, propeller and rudder design.

Reliability with simplicity, which is generally accompanied by accessibility and easy overhaul, is of primary importance. As the reliability of Diesel machinery is well established, vessels shown in Table I are generally fitted with single screws, there being less upkeep, cost and weight. On occasions, single-screw Diesel engines have been known to run passages with one of the complement of cylinders cut out owing to a cracked piston or cover or liner. In cases of such breakdown it is better, where practicable, not to remove the piston so that the full complement of cylinders can be used on air for manœuvring purposes. In one instance, a four-cylinder Doxford opposed-piston engine had one of the cylinders completely disabled due to the breakage of an upper piston rod stud; in consequence of this breakage the other stud broke, the centre bearing was carried away, the upper piston and skirt were fractured, the transverse was distorted and the side rods were badly bent. The upper and lower pistons were removed, the cylinder scavenge and exhaust ports were blanked, the side rods were removed, the fuel and starting air to the cylinder were blanked and the water service to the cylinder was shut off. The main and side connecting rods, crossheads and guide shoes were left in place, keeps being fitted to the side top ends to take the place of the side rod feet. The vessel then proceeded, on three cylinders, to the United Kingdom, ran into most adverse weather conditions through the Bay of Biscay, and manœuvred into port without a hitch.

Fuel Consumption.

All main engine types are subjected to test bed trials during which the fuel consumption is measured at various loads which are accurately obtained from a water brake. It is found in service that the consumption practically corresponds with test bed figures and sometimes even slightly better results are obtained owing to the improved mechanical efficiency when the engine has been "run in".

The main engine consumption may vary from about ·34 to ·39lb./b.h.p./hr. according to the type of engine fitted, and should vary little throughout the life of the engine; any increase in consumption will usually be due to faulty combustion, which can easily be observed by a smoky exhaust and higher exhaust gas temperatures. Such faulty combustion may be caused by choked spray nozzles or leaky fuel valves, due to water or abrasive materials in the fuel or to grit entering the fuel passage during overhaul. The spray nozzles should be cleaned out, fuel valve ground in (it is generally sufficient to use metal polish for this purpose) and the valves and spray plug joints tested on the test pump for tightness to about 4,000lb./sq. inch. Excessive liner wear will cause loss of compression and high fuel consumption, and faulty scavenge air valves or partially choked scavenge ports in 2 S.C. engines will reduce the quantity of air and cause bad combustion, as also will partially choked strainers to air inlet valves in 4 S.C. engines. If these faults are neglected, stuck and broken piston rings and other troubles associated with overheating may arise. In any case, bad combustion will always tend to increase the expenses of overhaul.

The mean indicated pressure in the cylinder under service conditions depends upon various factors in the particular design of engine, and average figures recommended for different types of engine are as shown in the following table:—

TABLE II.

Machinery.	M.I.P.
2 S.C. double-acting type	80lb./sq. inch.
2 S.C. single-acting type	85 " " "
Opposed-piston type	85 " " "
4 S.C. single-acting type	90 " " "
4 S.C.S.A. supercharged type	115 " " "

These figures allow a margin for emergency or for a cylinder temporarily receiving too much fuel, and the fuel consumption and engine speed should be based on the above figures. Should the fuel consumption exceed these figures, cracked pistons and other troubles due to overheating may occur.

A useful check on the performance of a vessel may be made by applying the two formulæ:—

(*a*) $$\frac{(\text{Displacement tons})^{\frac{2}{3}}\ (\text{Speed knots})^3}{\text{Shaft horse power.}} = \text{Constant.}$$

(*b*) $$\frac{(\text{Displacement tons})^{\frac{2}{3}}\ (\text{Speed knots})^3}{\text{Daily fuel consumption for steaming purposes, tons.}} = \text{Fuel coefficient.}$$

The results will vary according to hull form and propeller and rudder design, but for any given vessel, fair comparative voyage results can be obtained after due allowance has been made for weather conditions and the length of time since the previous drydocking and painting. In a modern tanker, fitted with a stream-lined rudder, a constant of about 350 may be expected for mean service conditions; the fuel coefficient may be 85,000 to 90,000.

It will be observed that in formula (*a*) s.h.p. is taken instead of i.h.p., which latter is used in arriving at the Admiralty constant. This is to make the formula applicable to all types of machinery, including turbines, by comparing power output at the shaft, thereby making it unnecessary to allow for the different mechanical efficiencies of the machinery types in arriving at a fair comparative constant. In any case, where constants are recorded it should be clearly stated whether i.h.p. or s.h.p. has been used in the formula for arriving at the results.

Diesel Fuel.

Some of the fuels which have been successfully used in Diesel engines that have come within the Author's experience are as follows:—

TABLE III.

	A.	B.	C.	D.	E.	F.
Specific gravity at 60° F. ...	·873	·873	·910	·888	·901	·885
Flash point° F....	186	178	190	204	206	186
Viscosity, Redwood 1 at 100° F. ...	secs. 36	secs. 44	secs. 107	secs. 61	secs. 77	secs. 65
Hard asphalt %	·002	1·07	Nil	1·98	Trace	·47
Conradson carbon residue % ...	·06	1·61	5·45	1·05	4·27	2·36
Sulphur % ...	·18	·99	1·47	·74	1·13	1·3
Calorific value B.Th.U.'s (gross) ...	19,150	19,450	19,300	19,400	19,100	19,580

Close observation has shown very little difference in cylinder liner wear with any of these fuels, although it will be noted there is a wide difference in the fuel characteristics.

The greatest care should be taken to avoid trouble due to contamination from water or dirt. Water will gradually accumulate in tanks due to condensation or sweating on the inside surfaces, and may be removed either by heating and settling or by separating in a centrifuge. Periodical use of drains in all settling tanks is advisable. In the case of cross bunkers built above the tank tops, syphon drain cocks are fitted to the bulkhead with internal pipes led to the bottom of the tank. These cocks should be periodically opened to draw off any water that may be in the tank. They are self-closing, so that they cannot be inadvertently left open.

The process of separation is, of course, speeded up by the use of a centrifugal separator. A small gravity disc, or adjusting ring, is fitted in the separator to vary the line of separation between oil and water. Various sizes of adjusting rings are provided to suit the different specific gravities of the fuels which may be used and for most efficient separation the lowest gravity disc should be used consistent with no oil coming out from the water outlet. The efficiency of separation is also improved by :—

(*a*) Heating the fuel, thereby reducing its viscosity; with the usual grades of Diesel fuel the temperature should be about 110° F. Gas oils and light non-viscous fuels, however, require no heating.

(*b*) Continually centrifuging at a slow throughput.

(*c*) Cleaning the separator each time the machine is stopped.

(*d*) Checking the speed of the machine as efficiency decreases with loss of speed.

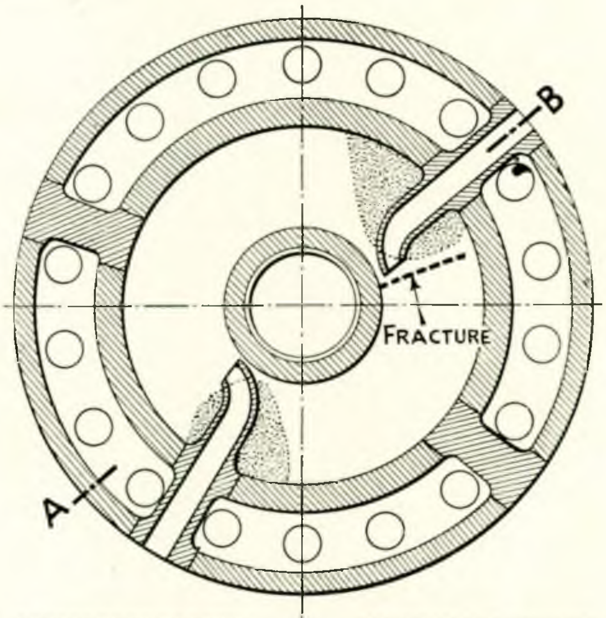
Water and impurities such as tank scale and rust are removed in this manner. It is emphasized that thoroughly clean fuel will rarely be a source of trouble, but even the best Diesel fuel or gas oil can cause much inconvenience and expense in maintenance if contaminated with dirt and impurities.

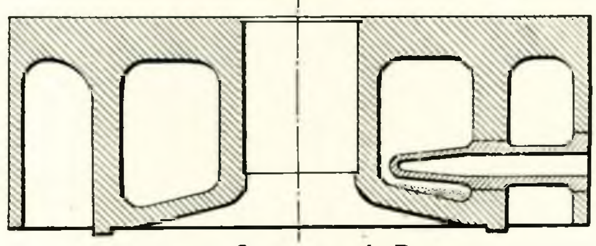
Lubricating Oil.

The consumption of lubricating oil may vary, according to the type of engine, from about two to four gallons per day per 1,000 b.h.p. for cylinder lubrication and from about one-and-a-half to three gallons per day per 1,000 b.h.p. for bearing lubrication. This is a costly item and care is necessary to see that the oil is kept in good condition and expenditure kept to a minimum compatible with efficiency.

Cylinder lubricating oil should be a purely mineral oil with a high flash point, say, 400° F. or above, and have a viscosity suitable for the engine. No definite viscosity is laid down in general practice; good results have been obtained with various grades from 200″ to 550″ Redwood I at 140° F. This oil is not recoverable after use, a considerable portion is burnt in the cylinder and the remainder drains off the liner walls, forming a black sludge of partly oxidised oil which is unusable. From 1 per cent. to 2 per cent. of colloidal graphite is sometimes added to the cylinder oil, it being claimed that an oil film will form over a graphoid surface more quickly than in the case of an untreated metallic surface, and in consequence that cylinder liner wear will be reduced.

Bearing oil, which may also be used for piston cooling, should be a pure mineral oil. It is continuously circulated throughout the system and re-used. Although free from acid when new, it will become contaminated with water, acid and other impurities after use, and it is of the greatest importance to use the lubricating oil purifier and streamline filter (if fitted) systematically. The bearing oil should be heated to about 180° F. before centrifuging and a constant trickle of hot distilled water, which can be obtained from the lubricating oil heater drain, should be mixed with the oil before it reaches the centrifuge since this washes the oil during separation and removes some of the acidity with the foreign matter. An independent hand pump should be installed with a suction pipe led to a hat box in the sump or drain tank; the water and sediment which collects at the bottom of the tank is thereby pumped out periodically, thus assisting in keeping the lubricating oil clean. The previous remarks as to centrifugal separators under the heading of "Diesel Fuel" apply generally to lubricating oil. Streamline filters are effective and they can be used in conjunction with centrifuges. Neglect of systematic and efficient use of the separators may sooner or later lead to corrosion of the shafting and other bearing surfaces, making necessary the more frequent adjustment of bearings.





SECTION A.B.

FIG. 8.

Piston and Jacket Cooling.

Pistons may be either cooled by oil or by distilled, fresh or sea water; jackets may be cooled by distilled, fresh or sea water. When water cooling is employed it should preferably be either distilled water or sea water. Fresh water has the disadvantage that it may contain solid impurities in suspension, and as the water is in a closed circuit a gradual silting up occurs due to continual additions of solid impurities each time the system is replenished. There is no escape from these impurities, whereas in the case of salt-water cooling they are being continually discharged overboard.

Fracture of cylinder covers and liners may occasionally follow upon local overheating caused by deposits, usually when sea-water cooling is employed. This is particularly noticeable in vessels employed in confined waterways where the cooling system is contaminated with finely divided mineral matter. Deposits, with consequent overheating and fractures, are particularly likely to occur in parts of the system where the circulation is poor. Fig. 8 shows a typical example of a fractured 2 S.C. engine cylinder cover due to scale formation. The covers and jacket spaces should be periodically scaled and cleaned with scale-removing solution—hydrochloric acid with an inhibitor to protect the iron against the acid—in the proportion of two parts of fresh water to one part of solution, which is allowed to stand for not less than 24 hours, after which it is thoroughly washed out to remove the scale.

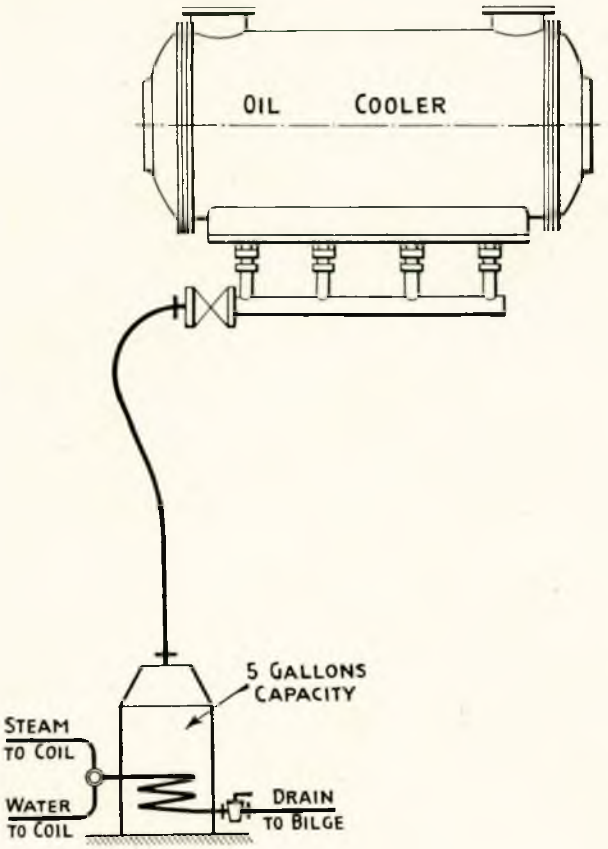


Fig. 9.

As the highest practicable jacket temperatures should be used to avoid, as far as possible, cylinder liner distortion and condensation of combustion products due to over-cooling, and taking into consideration the troubles associated with sea-water cooling, it is advisable to use a closed system with a rapid rate of circulation and to employ distilled water, which should be kept scrupulously clean. To some extent the saving in maintenance costs by the adoption of distilled-water cooling in preference to sea-water cooling is offset by the additional water cooler and pump required, but the increased reliability with less delay due to replacing defective covers and such parts outweighs this.

Where airless injection is fitted, the fuel valves should be cooled preferably by an independent cooling system with a visible return flow. This is to avoid oil contamination of the piston and jacket cooling system in the event of a leaky joint between the fuel and cooling water. Instances of cracked cylinder liners have occurred through oil deposits on the cooling surfaces caused by fuel leaking into the water system where a common cooling system has been employed. Distilled water in the system should be treated with potassium bichromate, which has an anti-corrosive effect on the water spaces, giving the jackets, etc. a rust-resisting skin. About 1lb. of commercial bichromate of potash (crystals) per ton of water in the system is dissolved in two buckets of hot distilled water and added to the jacket cooling system. An inspection of the water in the hoppers will show that this addition has given the water a yellow tinge and this must not be allowed to weaken. As the colour fades, more crystals must be added at the rate of 2lb. per 1,000 gallons of water in the system. After several applications the yellow tinge does not fade, whereupon the treatment can be stopped. Periodical tests are made by drawing off in a glass tube and, after allowing for settling, the colour of the water should be examined by daylight. A polished steel dip rod is left in the jacket water tank and daily inspected; should corrosion be observed further treatment is applied. Care must be taken not to use potassium bichromate if there is any salt water leakage in the system, since it would have the opposite effect owing to the formation of hydrochloric acid. The internal surfaces of pistons of Doxford opposed-piston engines using this system have been examined after six or seven years' service and the condition has been found as clean as new.

When the cooling medium for pistons passes through either telescopic or walking pipes which are partly arranged in the crankcases, there are advantages in using oil for piston cooling since this keeps the moving parts in better order and removes any risk of contamination of the bearing oil.

The piston and jacket coolers are cleaned

MAIN ENGINE CYLINDER LINER WEAR.

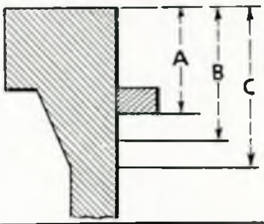
M.V. " "A" " *Port* London *Voy. No.* 27

Date of previous gauging of any main liner 14. 11. 1934.

No. of hours run since previous gauging of any main liner 7150

Type of Main Engine 4 Stroke Single Acting Air Injection

	FUEL.	CYLINDER LUB. OIL.
Quality (Brand)	Standard Marine Diesel.	Grade 205.
Average daily consumption ... ... Main Engines only.	10·25 tons.	5·25 gallons.



A—At top piston ring at edge farthest removed from Combustion Chamber.
B—3in. below position A.
C.—6in. " " "

TOTAL LINER WEAR MEASURED IN 1/1,000in. ABOVE ORIGINAL DIAMETER.
IN F. & A. AND ATHW. DIRECTIONS.

Liner No.		1.	2.	3.	4.	5.	6.	7.	8.
Date Fitted		11·34	11·34	11·34	11·34	7·33	7·33	7·33	7·33
Orig. Diam. Inches ...		29·134	29·134	29·135	29·135	29·134	29·135	29·135	29·135
A	F. & A.	45	47	43	42	98	94	101	98
	Athw.	39	38	39	37	92	89	94	88
B	F. & A.	44	42	47	46	102	104	102	97
	Athw.	41	38	41	40	91	90	95	92
C	F. & A.	40	36	42	43	96	94	96	93
	Athw.	31	28	39	37	92	87	92	85
Total		240	229	251	245	571	558	580	553
Mean Wear		40	38	42	41	95	93	97	92

REMARKS.

Piston heads clean and rings free. Several butt overlaps broken. Grooves machined to 16mm. in Nos. 5 & 7 pistons and oversize rings fitted. Triple seal rings fitted in 3rd and 4th grooves of all pistons. Several plain rings renewed. Maximum vertical clearance between rings and grooves 8/1,000in.
Nos. 5 & 7 pistons removed from rods. Small amount carbon removed.

Signature of Chief Engineer.

Signature of Superintendent (Engineering).

Date 15. 11. 35.

FIG. 10.

periodically internally and externally; in the case of oil coolers it is preferable to clean the oil side by a solvent, e.g. trichlorethylene. To do this a vaporiser is coupled to the cooler, as shown in Fig. 9 and the solvent is heated in the vaporiser; the vapours will then rise and condense on the surfaces of cooler and the condensate loaded with oil and grease will run back into the vaporiser. The cooler is then washed out with hot soda water, since hydrochloric acid may be formed if C_2HCl_3 remains in the cooler.

The engines should be warmed before starting, by means of a steam coil in the cooling system. When this is done the viscosity of the oil film on the cylinder walls is reduced, thereby making it easier to turn the engines. Less compressed air is required for starting and, owing to the higher temperature of air at the end of compression, the fuel is more readily ignited and combustion shock is reduced.

Cylinders, Pistons and Rings.

Cylinder liners always show most wear at the end nearest the combustion chamber, from the position of the top piston ring at the end of the stroke to, say, 3in. to 6in. below this. It is, therefore, recommended that liners should be gauged, both fore and aft and athwartships, at these positions only and the mean of these readings taken as the wear. A typical form used for recording liner wear is shown in Fig. 10 and this is card-indexed as shown in Fig. 11. These particulars should be accurately recorded so that a complete history of the liner can be kept for easy reference. The liner wear is generally found to be greater in the fore and aft direction than athwartships. This is chiefly due to the better guiding of the piston in the athwartships direction, as in the fore and aft direction the guide surfaces are much less and the clearances are generally considerably greater than in the athwartships direction. The lateral piston movements are thus more in the fore and aft direction than in the athwartships direction. While this applies to pistons rigidly attached to the crosshead and which, therefore, follow all movements of the crosshead, in the latest Doxford engine the piston attachment is "floating" with the object of allowing the piston to take up its own position in the cylinder irrespective of lateral movements of the crosshead (see Figs. 17 and 18).

In dealing with fuel and with cooling earlier in this paper, emphasis was laid upon the importance of carefully centrifuging the fuel to remove water and impurities and of keeping the cooling temperatures as high as practicable and maintaining the least possible difference between the inlet and outlet temperatures. These two measures were dealt with largely from the point of view of their bearing on liner wear and it is necessary here, therefore, only to repeat the emphasis laid upon them. A high rate of circulation, the use of distilled water cooling, heating of the liners, and pistons before starting, and the avoidance of local

MAIN CYLINDER LINER WEAR CARD.

M.V. "A".

TYPE :— 4 SCSA. Air Injection. 8 cyl. 740mm. × 1,500mm. Cyl. No. 1.

Date.	Total Hours Run.	Total Wear in 1/1,000in.	1/1,000in. Wear per 1,000 hrs.	Cylinder Lub. Oil.	Remarks, Characteristics, etc.
11.3.29 ...	10,650	64	6·0	Grade 190	Cast iron "X".
15.5.30 ...	19,075	109	5·7	,,	
6.7.31 ...	27,550	146	5·3	,,	
27.8.32 ...	33,350	173	5·2	Grade 205	Piston grooves machined, triple seal rings fitted to 3rd and 4th grooves only. Plain rings in remaining grooves.
25.7.33 ...	40,500	206	5·1	,,	
14.11.34 ...	50,100	255	5·1	,,	
	Liner renewed 11/34				
15.11.35 ...	7,150	40	5·6	Grade 205	Cast Iron "Y".

FIG. 11.

hot spots which may limit the average cooling water temperature, will reduce liner wear.

The mechanical cylinder lubricators should be worked freely before the completion of a voyage and again before starting up; in the latter case they should be worked by hand while the engines are given a complete revolution with the turning gear to ensure a film of oil on the liner walls. This practice will tend to prevent rust formation on the walls while in port and ensure that the lubricating oil is admitted to the cylinder immediately the engines are started.

Cylinder liners of 24in. to 30in. diameter will require renewal when the wear is between $^{200}/_{1000}$ths in. and $^{250}/_{1000}$ths in. according to the type. The liner material which so far has given the best results in the Author's experience is vanadium-titanium mixture.

In machinery types where it is necessary to remove the covers to withdraw pistons, the liner should be clamped down from the cover studs before removing the piston to ensure that the ground faced joint between the liner and jacket is not disturbed, otherwise scale may lodge there and cause leakage and corrosion at the joint face. Liner ridges should always be ground off when the cylinders are opened for examination.

In some types of 4 S.C. engines, pistons sometimes develop cracks in the crown and have been known to crack in way of the piston grooves. A method employed to overcome this is to cast the piston with a core in the crown and to machine and screw a nickel cast iron plug in the hole. Care must be taken to ensure a good joint at the shoulder. Screwed dowels locate the plug as shown in Fig. 12.

The internal surfaces of pistons should be examined periodically and any scale or carbon deposits removed. Pistons generally require examination every 12 to 15 months.

Piston rings may be fitted with lap joints or scarf joints. Lap joints form the best seal, but are more liable to break at the joint, particularly if the liner is worn. Scarf joints are quite satisfactory

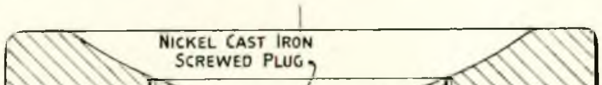


Fig. 12.

and less breakages have been found in 2 S.C. engines when the corners near the butts are well chamfered. Scarf joints should be cut and fitted alternately right and left hand. Joint clearances should not be less than ·003in. per inch diameter since if the clearance is too fine they may butt at the ends under working conditions, thus causing breakage of the rings.

The radial thickness of the ring is generally equal to the $\frac{\text{piston diameter}}{30}$ and the free gap is two to three times the radial thickness, the wall pressure being about 7lb. per sq. inch. This thickness is governed by the spring necessary to insert the ring over the piston without permanent distortion. When the ring is touching the bottom of the

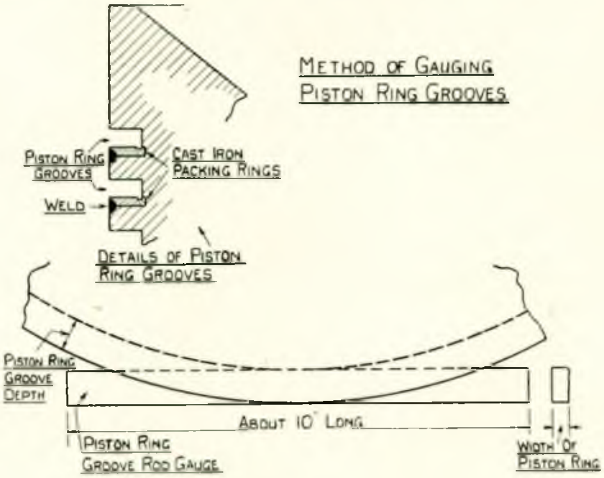


Fig. 13.

groove it must not be proud of the piston otherwise breakage will occur, so that carbon behind the ring should be cleaned out before new rings are fitted.

The up and down clearance in the piston grooves increases by wear. The wear is far greater when salt-water cooling is employed than when distilled water is used, owing to the higher cooling water temperatures which can be employed with the latter, thereby reducing effects due to the condensation of combustion products. The necessary vertical clearance should be made when the rings are supplied and this is generally about $^{4}/_{1000}$ths in. When the vertical wear in the groove exceeds about $\frac{1}{2}$ mm. the grooves should be machined to the next mm. size and oversize width rings fitted. In the case of small pistons the grooves should be reconditioned in $\frac{1}{2}$-mm., sizes; by this means three or four standard sizes can be carried instead of numerous rings having different widths. Fig. 13 shows the piston grooves of a Doxford steel piston, fitted with renewable cast-iron wearing ring. A radius should always be left at the bottom corner of the groove to prevent ridges being formed. This figure also shows a gauge for ensuring the working surface of the grooves being perfectly flat.

Piston rings bed themselves on the piston collars where gas tight joints are required if the rings are to be effective, and they should consequently not be removed unnecessarily. Rings can easily be overstrained or distorted, causing them to be less liable to hold tight and more liable to break. In consequence care in handling, stowing and fitting them is most important.

Compound rings of the double-seal and triple-

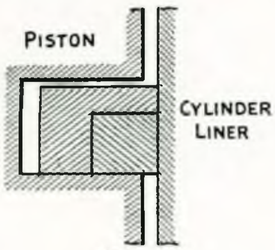


FIG. 14.

seal types similar to that shown in Fig. 14 have been successfully employed, particularly in worn liners since they form a better seal than a plain ring and maintain better compression in the cylinder.

Care should always be taken in 2 S.C. engines when cylinders are opened for examination to see that all inward springing scraper rings acting on the piston skirt have ample butt clearance, without which the scraper rings become ineffective and over lubrication results.

Valves.

Great care should be taken in overhauling fuel pumps and valves to see that they are not damaged in handling. They must be kept scrupulously clean and dirt excluded. In 4 S.C. engines, the exhaust valves should give at least 60 days running service before examination and should then only require slight regrinding. Systematic inspection is preferable to waiting for the valve to leak—small leaks spread rapidly due to erosion caused by the velocity of the escaping gases. Where exhaust valves burn out rapidly, the trouble is caused either by overloading and excessive exhaust temperatures or by dirt or solid impurities in the fuel which should have been removed by efficient centrifuging. Inlet valves should be examined at intervals of 150 to 200 days running—apart from anything else the valve cages may be found difficult to remove if left too long in place. Air inlet valve strainers should be kept clean since loss of power and high exhaust temperatures will be the result if they are allowed to become partially choked. Air starting valves sometimes leak due to carbonised oil caused by over lubrication, but generally these valves give no trouble. It is good practice to keep a complete spare set of exhaust valves and boxes ready for use so that the used valves can be periodically replaced and reconditioned at convenient opportunities.

Indicator Cards.

Diagrams should be taken as soon as practicable after leaving port and periodically during the voyage. Phase or "draw" cards should always be taken. At the top of Fig. 15 a typical card from a 2 S.C. air-injection engine is shown; the second figure shows a typical diagram for a Doxford opposed-piston engine with phase card; the third figure shows a firing diagram taken with the indicator mounted on the fuel valve; the fourth figure shows a compression card taken from a 4 S.C.S.A. (unsupercharged) air-injection engine. All Doxford opposed-piston engines are fitted with a cam at right angles to the indicator cam so that the compression pressure, the timing, the point at which combustion commences and the nature of the pressure rise above compression pressure can all be checked on the phase card. It is thus possible to detect faults arising from too early or too late injection and to control the rate of fuel injection

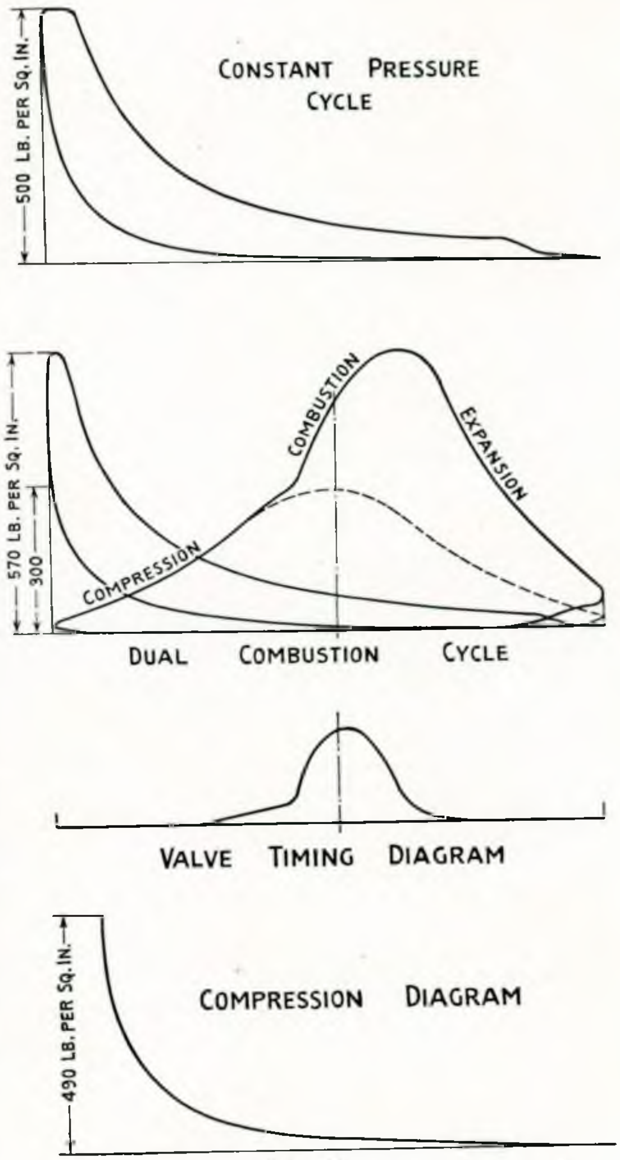


FIG. 15.

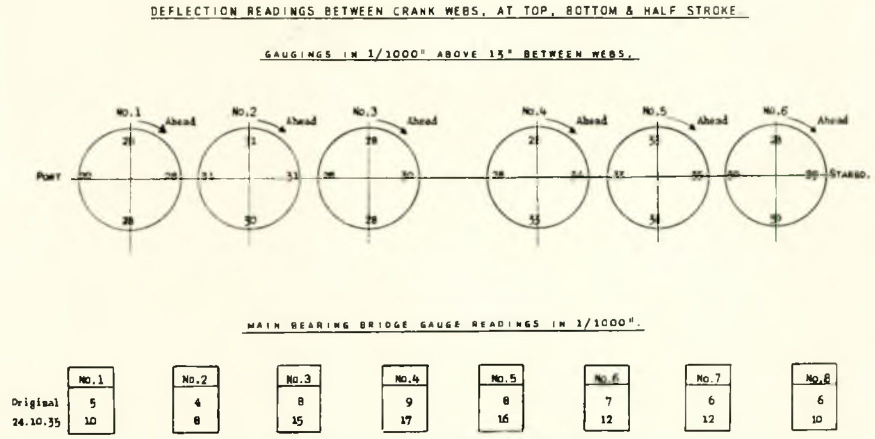


FIG. 16.

to secure a correct diagram. It is strange that some engines are still not provided with the 90° mechanism, this being much more useful than the ordinary type. When taking the ordinary type of diagram the compression diagram is taken with the fuel shut off the cylinder to check the indicator mechanism. Should the compression line not coincide with the expansion line an incorrect diagram will result until the mechanism is rectified. An error of 15 per cent. was found in the diagrams of one engine due to the wear on the gearing of the camshaft drive, the indicator being driven from the camshaft. Too much reliance, therefore, should not be placed on the indicator cards for determining the mean indicated pressure. They should be read in conjunction with the exhaust gas temperatures and the fuel consumption. The exhaust gas thermometers should be checked periodically for accuracy.

Vibration.

Hull pulsations will occur when the natural period of vibration of the vessel coincides with the disturbing forces from the machinery or propeller. For instance, several vessels of the same form and dimensions having a deadweight of about 11,000 tons are known to be subjected to vibration when the engines are running at about 85 r.p.m. The machinery of these vessels, however, is not identical, some being of the six-cylinder 2 S.C.S.A. type, some eight-cylinder 2 S.C.S.A. type, and others 4 S.C.S.A. type. These engines are in every case fitted with 4-bladed propellers and it has been observed that by increasing or decreasing the engine revolutions from 85 r.p.m. the pulsating ceases. The indications are that the natural period of vibration of the hulls is an integral multiple of 85 coinciding with a harmonic of the forces transmitted from the propeller to the hull.

Shaft vibration, as distinct from hull vibration, occurs when the natural period of vibration for systems having one or more nodal points coincides with some order of the harmonic forces from the engine cranks. The importance of the various orders may be obtained having regard to firing order, crank sequence, and the normal elastic curve. It is found that the effects of some orders almost cancel out, whilst others from the various cylinders build up. The latter orders, if coinciding with the natural frequency, would build up a critical speed and it would be necessary to avoid this speed in service, since it is not possible for an engine to be run long under a condition of critical speed which imposes severe stresses on the shaft without defects and perhaps failure occurring in the shafting.

Torsiograph readings are now often taken from a shaft system and the problem is carefully looked into by engine designers so that dangerous criticals are avoided.

It is practically impossible under the best weather conditions to keep a constant engine speed

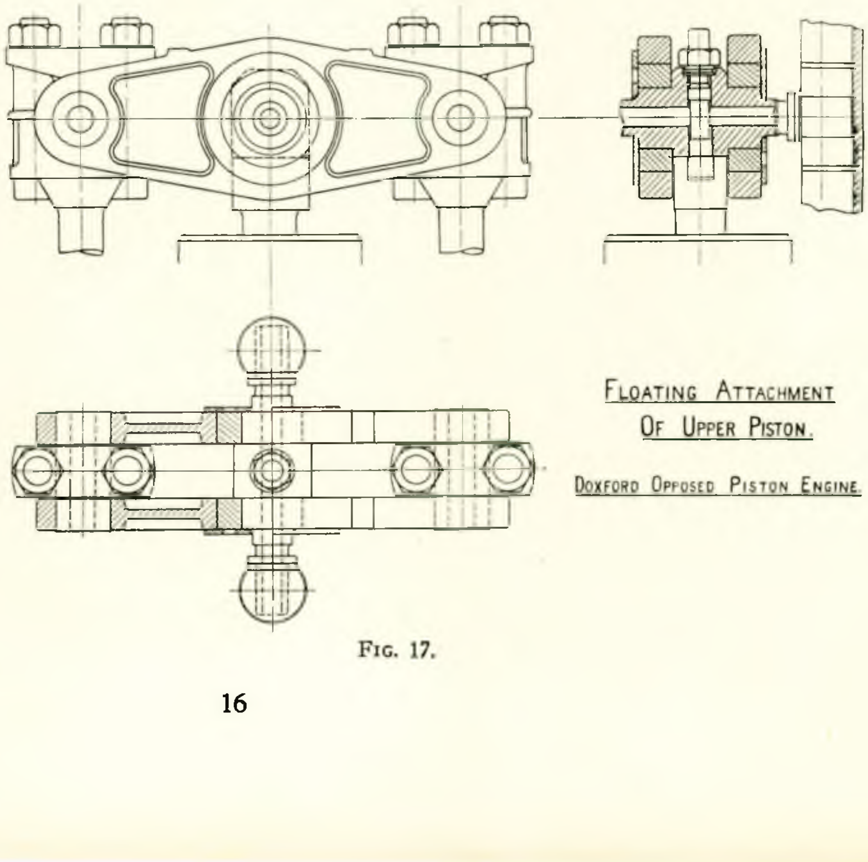


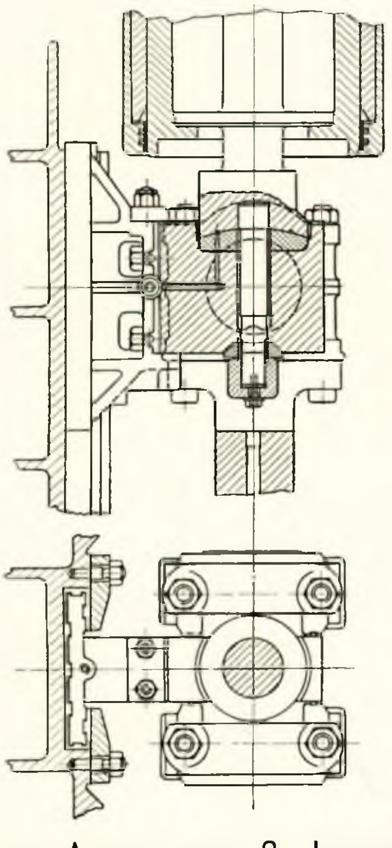
FIG. 17.

within 2 or 3 r.p.m. and within a far greater range in bad weather. It may, therefore, be considered advisable in some instances to fit a damper or anti-phase desynchroniser at the forward end of the shaft, when it is known that an important critical is within the range of normal service revolution speed, in order continuously to change or damp out the crankshaft's own vibration.

The necessity of good engine seatings need not be stressed; further, good workmanship is of paramount importance to obtain a lasting job. All intercostals in double bottom tanks should be planed and be a tight fit between floor plates. Care should be taken in the construction of the seatings to see that the vertical plates are bedded on their top and bottom edges, so that they transmit the load direct and not through the rivets holding the angle bars. All rivet holes should be punched ⅛in. less in diameter and reamered in place.

Routine.

It is not intended to lay down any hard and fast rule for periodic examinations, etc. since these may vary with the type of machinery, trade and service conditions, but it should be emphasised that a systematic and orderly inspection should be carried out at specified periods and accurate records kept of all examinations, adjustments, wear, etc. No part, however insignificant it may appear, should be neglected. Cleanliness is essential to efficiency.



FLOATING ATTACHMENT OF LOWER PISTON.

DOXFORD OPPOSED PISTON ENGINE.

FIG. 18.

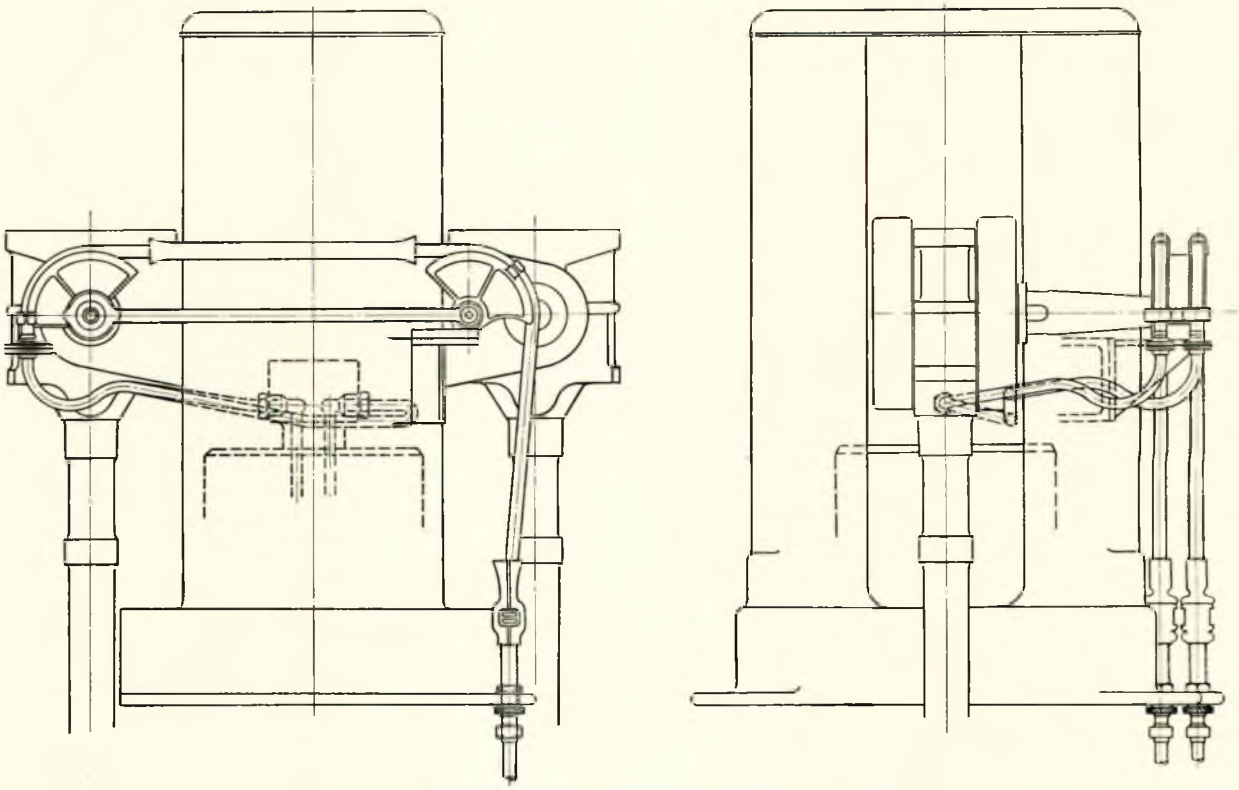
Whenever the crankshaft is opened, careful scrutiny should be made for cracks appearing on pins, journals and webs and also for any signs of incipient corrosion on shaft surfaces. Means should also be adopted for ascertaining with accuracy the wear-down at the main bearings, both by clock gauge readings between the crankwebs to measure the deflection and by bridge gauge readings taken from the journals, and these should be compared with the original readings. Fig. 16 shows typical readings taken from a six-cylinder 2 S.C. engine. Failures may occur through wear-down of bearings and misalignment. This requires more frequent checking in a Diesel engine than in a steam reciprocating engine, since in the latter case misalignment will show up in the form of a hot bearing, whereas in a Diesel engine with forced lubrication the shaft may be apparently running well for long periods with considerable flexing taking place.

Main bearings and top and bottom end brasses should be examined every 18 months. The bearings should not be allowed too much clearance as this may cause cracking of the white metal and will cause a loss in lubricating oil pressure.

Systematic examination of all nuts and attachments should be carried out at regular periods to avoid gradual slacking off in engine framing and running parts. Examples of such parts to be borne in mind are: holding down bolts securing the bedplate and thrust block to seatings, the nuts securing columns or engine frames to bedplate, through tie bolts, attachment of columns to cylinder jackets, main engine piston rod attachments to pistons and crossheads, piston rod attachments in air compressors and scavenge pumps (where fitted) and crank shaft coupling bolts.

Horseshoe brass liners should not be fitted under piston rod feet. Where liners are necessary for adjusting the compression, one solid steel plate, machined top and bottom, should be fitted.

Breakage of bottom-end bolts in Diesel generator engines may be caused by a sudden seizure of the piston, but it may also be caused by the location of the brasses on the foot of the rod, or their location relative to one another being insufficient, thereby allowing the brasses to move relatively to the rod. This causes alternate bending stresses in the bolts, in addition to the normal tensile stresses, and ultimately causes fracture. When new bolts are fitted in old brasses it is, therefore, essential that the bolts are a good fit in the brasses and also that the top brass is well spigoted to the bottom of the rod so that no side movement



UPPER PISTON COOLING WATER SERVICE.

DOXFORD OPPOSED PISTON ENGINE.

FIG. 19.

is possible between the brasses and rod or between each brass.

All bolts of special material should have identification marks since cases have occurred of bolts being annealed, with consequent failure, through no record being available that the material was of high tensile steel.

All overhauling tackle, spare gear, tools and equipment should be stowed in an orderly manner and kept efficient and ready for immediate use in case of emergency.

Improvements in Design.

Many improvements in design have been made in the past few years which have had a marked effect in reduced maintenance. A few examples may be touched upon.

In the Doxford opposed-piston type, the trans-

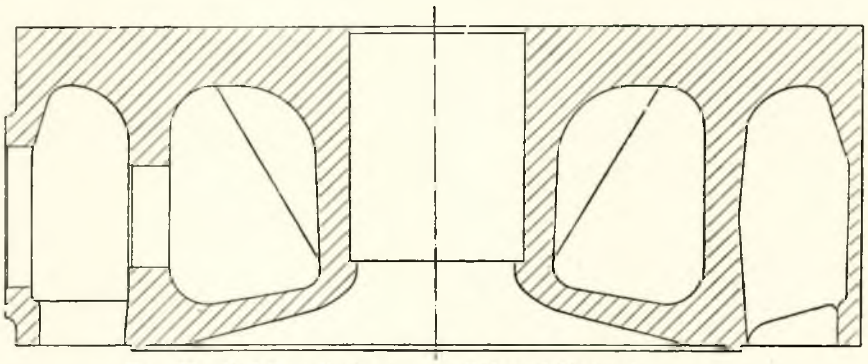


FIG. 20.—2 S.C. cylinder cover—Sulzer (old type).

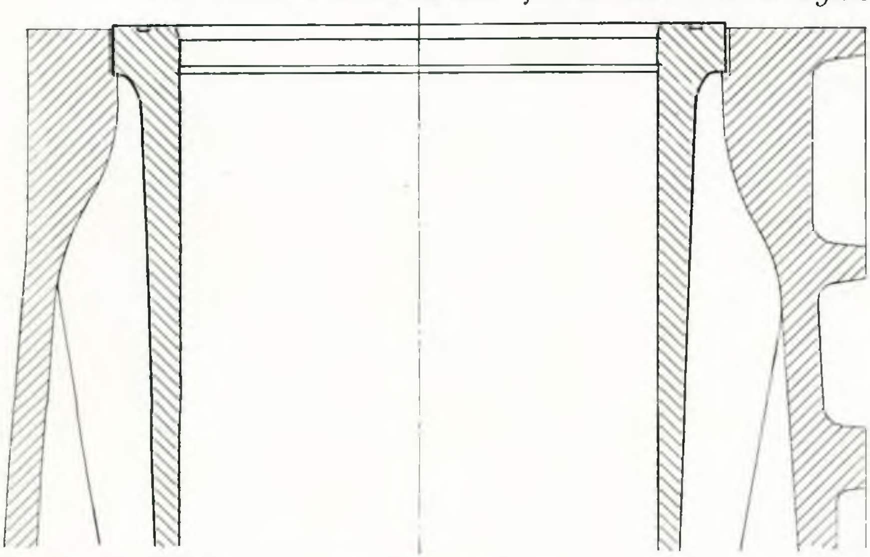


Fig. 21.—2 S.C. cylinder liner—Sulzer (old type).

verse connection to the upper piston rod and side rods has been modified as shown in Fig. 17. The centre bearing has been dispensed with and two large bearings, packed with lubricant, are embodied in the side arms, thus allowing the upper piston to float more freely. Fig. 18 shows the method employed for allowing the lower piston to float. The spherical seats on the lower floating pistons are of cast iron, and the cap nut is screwed hard against the end of the stud leaving $^{2}/_{1000}$ths in. clearance on the spherical seatings, which are also lubricated. The liner wear is reduced with these arrangements for reasons already given (see page 13). Fig. 19 shows the latest method of upper piston cooling. It will be seen that a flexible hose supplies cooling water to the piston and another flexible hose takes the water from the piston outlet. Travel is allowed for by a slipper guide. Adjustment for the hose is provided but this requires practically no attention. This method dispenses with the former spring glands and elbow pieces which require frequent overhaul, and also with the outlet jet pipe which was found to inject air into the piston cooling system. Deaerators are not required with this system.

Until recently the type of cylinder cover fitted in the Sulzer 2 S.C. engine was as shown in Fig. 20, and the upper portion of the cylinder liner as shown in Fig. 21; the liner was modified as shown in Fig. 22 as cases have occurred of fractured liners and jackets due to insufficient allowance for expansion owing to scale formation in the recess between the liner flange and cylinder jacket. Fig. 23 shows the latest type Sulzer cylinder cover; the cast-iron inner cover takes the heat stresses and the steel outer cover forms a bridge to transfer the tensile stresses to the bolts without imposing bending stresses on the part designed to take the heat stresses. Rapid circulation is provided by the spiral flow to prevent deposit formation. Fig. 24 shows the latest type Sulzer cylinder liner. The thinner metal will be noted at

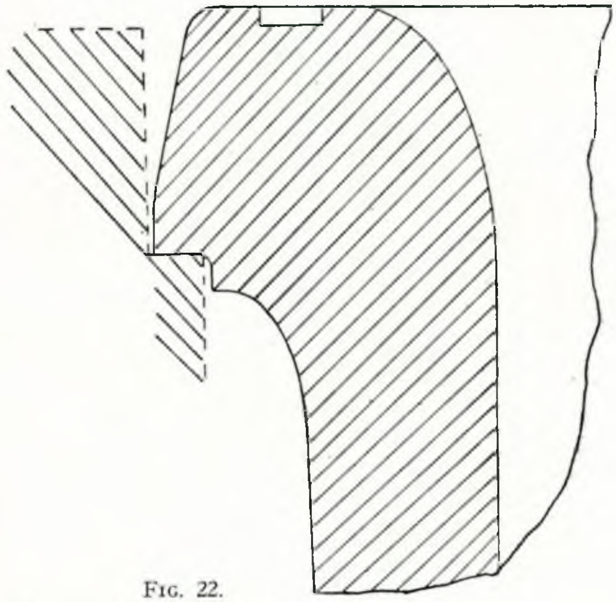


Fig. 22.

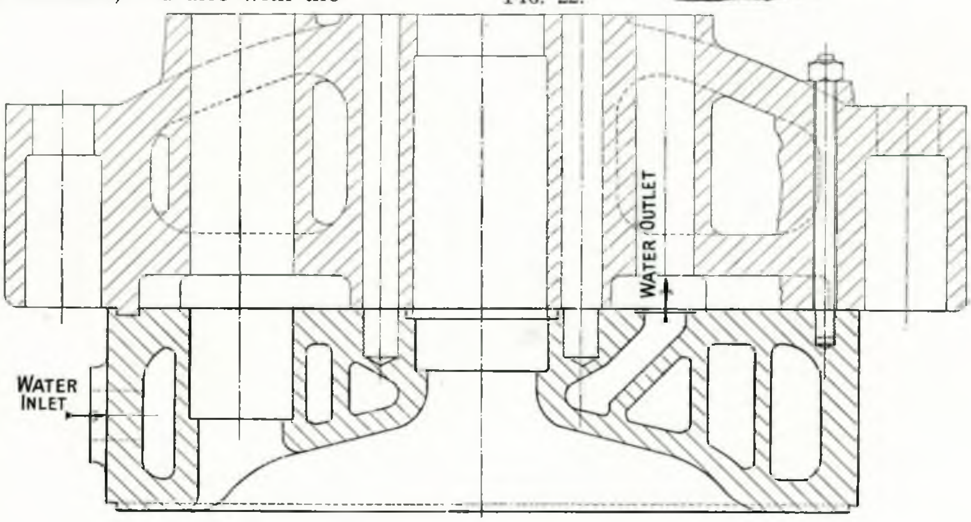


Fig. 23.—2 S.C. cylinder cover—Sulzer (latest type).

the top, this being made possible by dispensing with water-cooled exhaust and scavenge bars, thereby permitting the liner to pass through a smaller jacket aperture. A more even temperature is maintained—with consequent less liner wear—since the bottom of the liner is not cooled, and improved and rapid circulation is provided for at the top. The internal firing ring will be noted at the top of the liner. There are practically no bending stresses from the joint and less tensile stress on the cover due to the smaller area inside the joint ring.

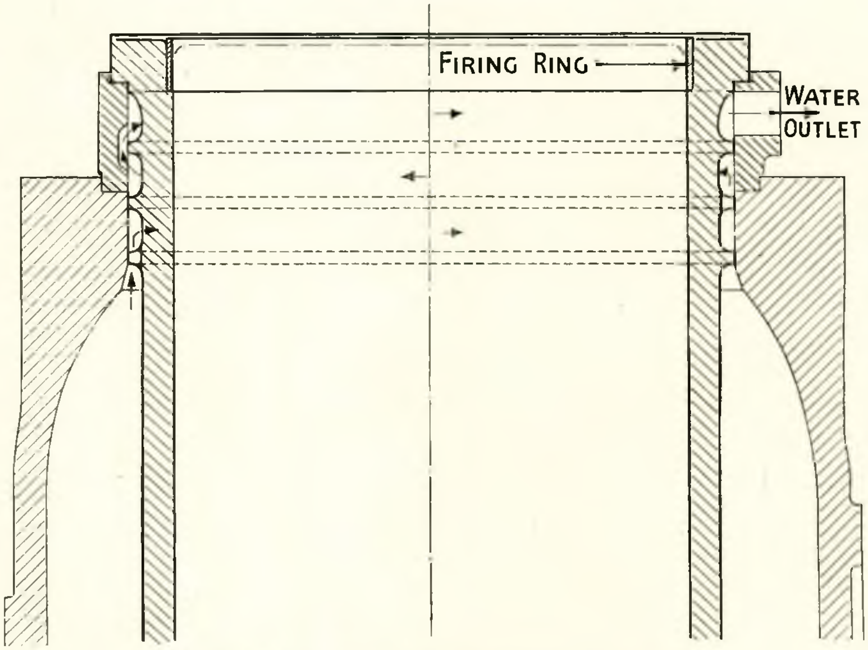
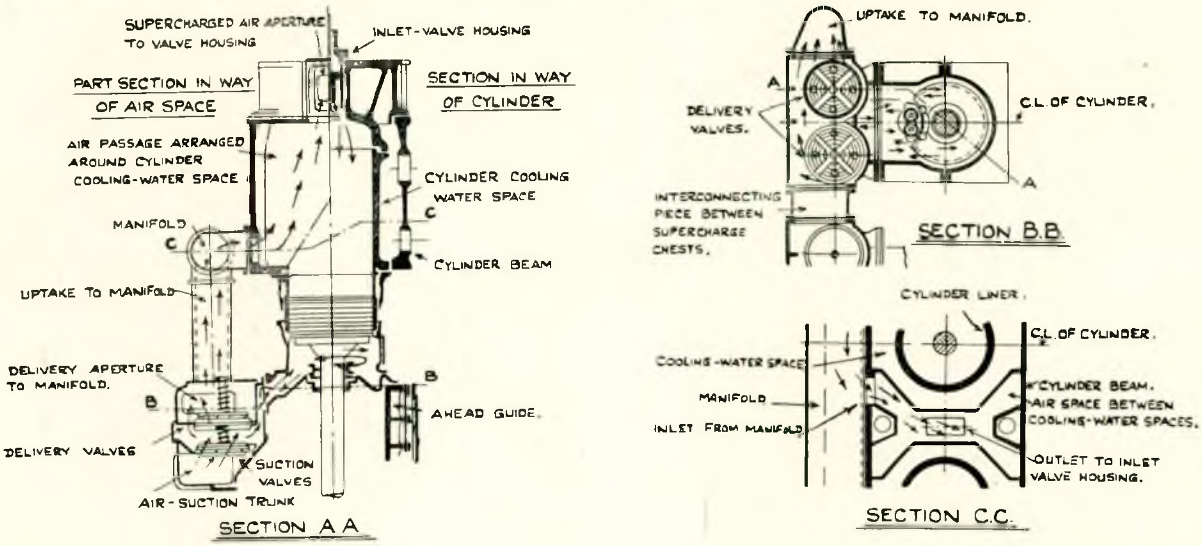


Fig. 24.—2 S.C. cylinder liner—Sulzer (latest type).

To increase the power of the 4 S.C.S.A. engine supercharging is now employed, by which means an increase in power of from 25 per cent. to 40 per cent. can be obtained. Supercharging does not mean overloading. In an overloaded cylinder the mean pressure is increased by injecting more fuel without at the same time introducing the larger quantity of air which is required for perfect combustion. This gives rise to high temperatures and high stresses in the material of covers, cylinders and pistons. In a supercharged engine, however, fuel and air are introduced in the same proportion as in a normally rated engine. The mean temperature of the cycle is not higher than a normal load and the strain on the material is the same. Fig. 25 shows the method employed in the Werkspoor system with under piston supercharge. The only moving parts added to the engine are the automatic lift valves.

Auxiliaries and Boilers.

Although in many cases nowadays tankers discharge by their own steam, tankers carrying benzine and crude cargoes are still frequently discharged by



WERKSPOOR SUPERCHARGING ARRANGEMENT.

Fig. 25.

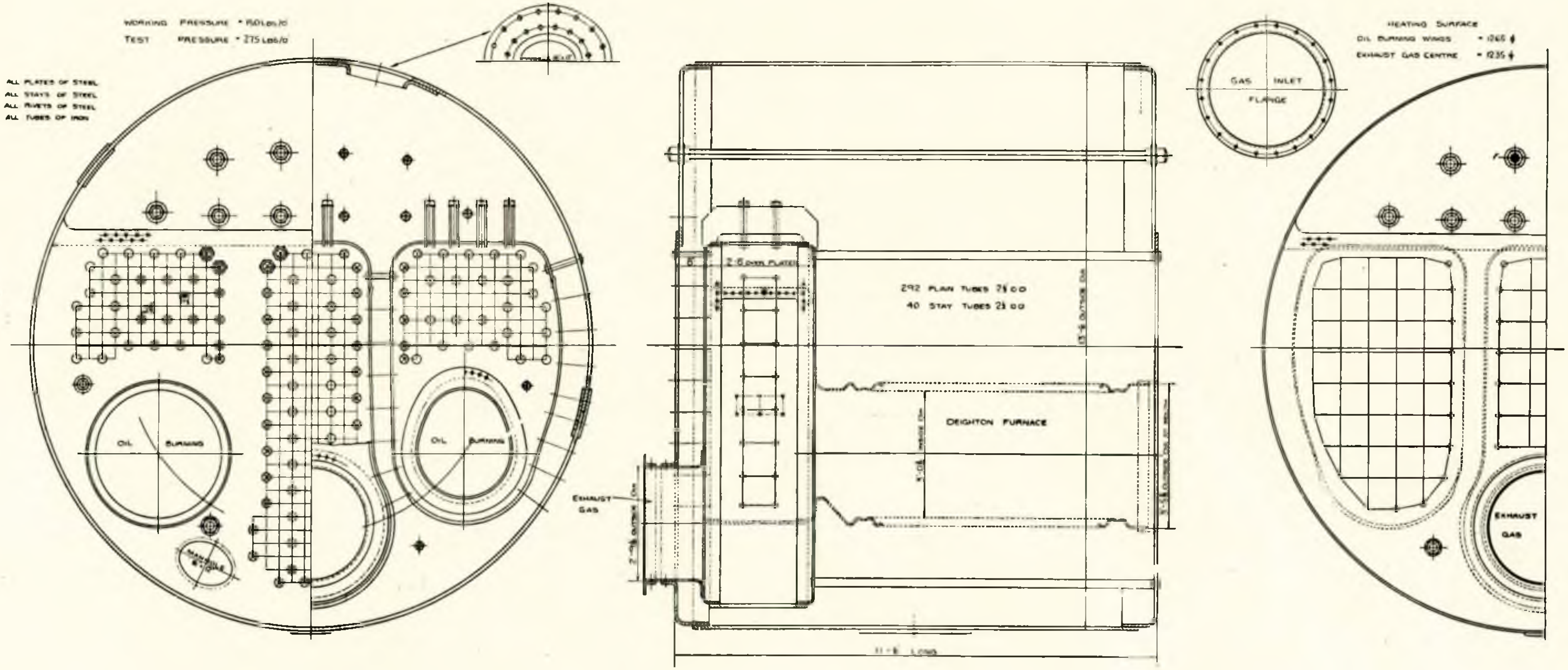


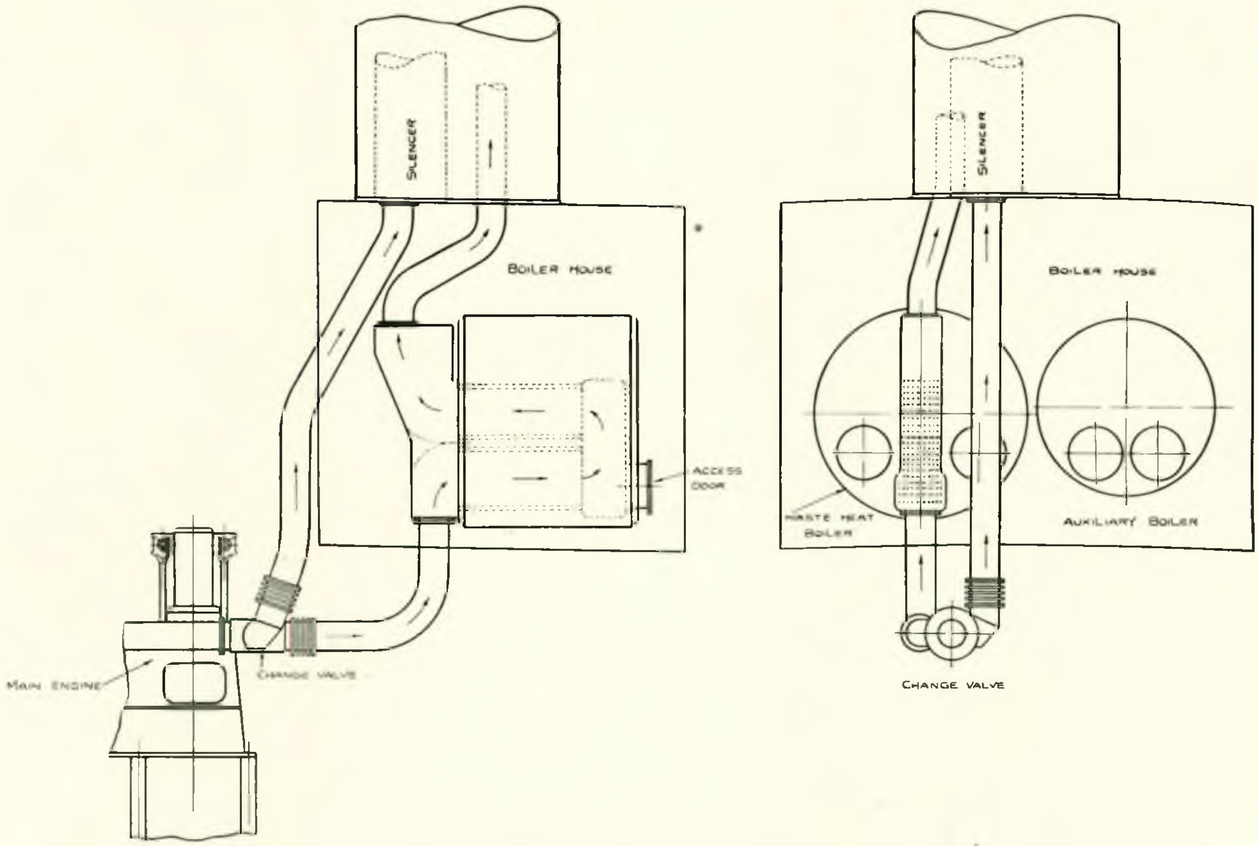
Fig. 26.—Waste heat boiler.

shore steam supplied to the ship's cargo pumps; the latter, therefore, are usually of the steam-driven type. To provide for this a comparatively large boiler installation is required on board.

Fig. 26 shows a method of utilising main engine exhaust gases at sea in an auxiliary boiler which is also used for the discharge of cargo. This boiler has been used in a Diesel tanker for nearly twelve years and has given excellent service. No major repairs have been necessary and, in fact, there has been less expense in upkeep with this than with another boiler, of similar construction but without the waste heat arrangement, which is fitted in the same vessel. With this arrangement, the waste heat passes through the centre combustion chamber and nest of tubes and then by an independent uptake to the funnel top. The wing furnaces can be utilised for oil burning at the same time as the waste heat gases are used, should it be necessary to augment the steam supply. The waste heat gases can be byepassed to the silencer if necessary. Fig. 27 shows the path of exhaust gases in a later arrangement. It will be observed that in this arrangement the gases go through the centre nest of tubes by means of a double pass.

In this type of boiler, when waste heat gases only are utilised, the evaporation is generally from 0·5 to 0·85lb./b.h.p./hr. Owing to higher exhaust gas temperatures in a 4 S.C. engine the evaporation is higher than in a 2 S.C. engine, and the steam available will depend on the type of machinery fitted and whether the exhaust steam is led to a feed heater or direct to the auxiliary condenser.

In conclusion, it cannot be stressed too strongly that to ensure a successful job the same care should be given to the auxiliaries and boilers as is required for the main engines. Electrical machinery, refrigerating plant, steering gears, pumps, etc., although having an important bearing on maintenance, cannot be dealt with in one paper; in fact, the Author realises that he has dealt very inadequately with many other important subjects.



ARRANGEMENT OF ENGINE EXHAUST & WASTE HEAT BOILER

Fig. 27.

Discussion.

Mr. T. R. Thomas, B.Sc. (Chairman of Council), opening the discussion, said that the Author had already given them such a great deal of interesting and useful information that it seemed rather unfair to ask him to add to it. The speaker did not consider that he was qualified to discuss a great part of the paper, although it had been no less interesting to him on that account, but he would ask the Author to place them further in his debt by giving a little more information regarding the statement that he had sprayed the surface of tanks carrying benzine with lubricating oil as a partial prevention against corrosion. The speaker's experience had not led him to think that lubricating oil would stand up satisfactorily in the presence of benzine.

The subject of corrosion was of great interest to those who were concerned with the maintenance of vessels carrying oil cargoes, and the speaker's experience had been that the corrosion below the surface of the liquid was very much less than that at and above the level of the oil. There was no doubt that the alternation of benzine and water ballast had a serious effect on the rate of corrosion, as in cases where the tanks were not used for ballast, that portion of them which was below the surface of the spirit remained in reasonably good condition. So far as he knew, there was no satisfactory means of preventing corrosion of a serious character in ships carrying spirit in bulk, and it was for this reason that he would ask the Author to be good enough to give them the benefit of his experience with lubricating oil spraying.

The Author showed a very useful method of repairing the piston head of a Diesel engine, and the illustration was more interesting because that day the speaker had seen a piston head being repaired by welding a star-shaped crack, which was in process of being welded by the oxy-acetylene blow pipe after suitable preparation. He would be glad of the Author's opinion on the efficiency of such a repair.

He felt sure that the Members of The Institute would greatly appreciate Mr. Humphreys' paper on account of the valuable practical experience which it contained, and he thought that it would not only be read with interest but kept on account of its value for purposes of reference.

Mr. F. W. Youldon (Member of Council) said that in regard to construction he was under the impression that most of the tankers built in recent years were of the two-longitudinal bulkhead type and he agreed with the Author that this type of construction offered more flexibility in regard to loading, in view of the widely varying specific gravities of the oil which the modern tanker had to carry. Further, it was a distinct advantage when a vessel was required to carry what was known as a "mixed cargo", i.e. several grades of oil on the same voyage, as the required quantities could be more closely gauged, and the smaller wing tanks permitted smaller parcels of one grade to be carried than was the case with the centreline bulkhead and summer tank type of vessel.

In regard to pipe lines, these might be either steel or cast-iron. The speaker strongly favoured the latter, as it had been his experience that steel lines corroded very quickly in spite of giving them a quarter or half turn periodically. Further, he preferred the hemp-packed expansion gland to the lead-packed sleeve, as it gave the staff less trouble in keeping it airtight and rendered the pipe line more accessible for turning.

He thought the Author had made an entirely unintentional error on page 2 in regard to the necessity for heating cargoes. The reason for maintaining the temperature of creosote at between 90° and 100° F. was not to reduce the viscosity but to prevent the precipitation of anthracine which, if the temperature fell below 90°, settled down on the stringers and tank bottoms and could not be reliquefied unless a temperature of at least 200° C. was applied, which was naturally beyond the temperature obtainable by means of heating coils as fitted in a ship's tanks.

Corrosion in cargo tanks was, of course, the main bugbear of the tanker owner. The speaker had found that heavy corrosion had started on the internal side of the bottom plating of a tanker even before she left the builders' hands, due no doubt to inefficient drainage, which allowed pools of rainwater and obnoxious liquids to remain on the plating and start corrosion. He had also found that with some oils corrosion was more active on new material which still retained its mill scale than on older material which had a coat of hard scale already on it, and he was of opinion that the factors which caused corrosion were four, i.e. mill scale, salt water, acidity (particularly napthenic acid) and lack of protection. It was very noticeable that corrosion was always heavier at such portions of the structure as were subjected to working strains, i.e. fatigue of the material at these points accentuated corrosion.

In regard to machinery upkeep, here again he found himself in sympathy with the Author as regards the time available for the engineers to carry out the necessary routine examinations and upkeep overhauls. Not only was the time in port very short, but an additional trouble was that there were a number of ports where the local regulations forbade any work being done in the engine room while low flash oils were either on board or being loaded or discharged. It was therefore essential that an adequate number of spare parts such as inlet and exhaust valves and their housings, be carried so that these parts could be changed in their entirety and what overhauling was necessary done at sea.

He was pleased to note that the Author laid stress on the necessity for periodical scaling of cooling-water jackets. Not only should this be given attention at drydocking, but in between dry-dockings periodical examination of these spaces should be made (particularly if salt-water cooling was used and if the vessel had made a passage up muddy rivers) and any mud found in them should be washed out.

The importance of examining all nuts and attachments, either of stationary or running parts, had not been overstressed, and efficient locking arrangements on all nuts inside the crank case should be provided where possible. In regard to top and bottom end bolts and also main bearing bolts, these should be marked to a trammel and tested for elongation on each occasion that they were removed when overhauling the bearings.

On the subject of boilers, there was no doubt whatever as to the economy to be obtained by utilising the otherwise waste heat of the exhaust gases for the purpose of raising steam and for this reason he favoured steam auxiliaries for such essential services as steering gear, refrigerator and dynamo, as ample steam for these purposes could be obtained from exhaust gases alone.

The boiler or boilers could be arranged with two or three furnaces each, of the ordinary corrugated type, the back ends, smokeboxes and funnels for each furnace being independent. By these means, one or more furnaces of each boiler could be used for exhaust gas heating, while the others were in use with the oil burners working, if more steam was required for such purposes as heating cargo, etc. The smokeboxes and uptakes required to be suitably stiffened to stop vibration due to pulsations of the exhaust gases.

Mr. Harry Hunter, O.B.E. (Member) said that the Author dismissed the twin-screw tanker rather abruptly. Had the Author's Company investigated from the propulsion point of view the advantage of better immersion and/or lighter draft in ballast which twin screws would permit? The speaker was well aware, of course, that from the upkeep point of view the twin-screw vessel might be ruled out, but he would be glad to know if the matter had been investigated from the propulsion point of view, particularly on ballast voyages.

Under the heading of "Corrosion in Cargo Tanks" the Author suggested that corrosion-fatigue had some connection with the corrosion in the tanks. If that were so, one would expect the corrosion to be a maximum where the alternating stresses were at a maximum both as to magnitude and frequency. This would probably be near the main engine. Did the Author's experience indicate that corrosion was greater in the tank immediately forward of the main engines or at any rate where alternating stresses were of greater frequency?

Could Mr. Humphreys give them more information in regard to the hard asphalt referred to in Table III? Was that hard asphalt combustible and, if so, what percentage of incombustible ash was present in the various fuels? They had here 1·98 per cent. hard asphalt and below the Table was the Author's comment that "close observation has shown very little difference in cylinder liner wear with any of these fuels". About the same percentage of ash was found in some coals.

With regard to indicators, the Author advocated that an indicator operated by a cam which reproduced the combustion motion should be fitted. After telling them about the troubles of that method the Author went on to emphasize that a compression card must be taken to correct the power card. A compression card which superimposed the lines was not strictly correct. It could be seen from cards taken between the top and second piston rings of an engine that the pressure rise between the rings was out of phase with the cylinder pressure and therefore gas escaped. The speaker submitted that the crosshead gear was the right type and an out-of-phase card should be taken from indicator gear of an adjoining cylinder.

The Author referred to the annealing of bottom-end bolts. Did he adopt annealing as a regular practice? It seemed to the speaker doubtful whether annealing was of any practical use. Present-day knowledge of fatigue indicated that if a part was going to fail from fatigue a crack would start and extend, and therefore the correct procedure seemed to be to watch for the crack starting.

With regard to the cracks in pistons, cylinder liners and cylinder heads mentioned by the Author, one inferred from the paper that these occurred on the hot side of the member. Was that inference correct? It was of interest, because according to the generally accepted heat stress theory such cracks should start from the cold side of the wall. Dr. Dorey, in his recent paper on failure of piston rods of double-acting Diesel engines, applied the heat stress theory successfully, it being noted that the cracks in the cases Dr. Dorey cited all started, as per theory, from the cold side. Possibly the Author might have had reason to investigate his cylinder part failures in detail, and might be able to give an explanation of this apparent divergence.

In dealing with the vertical wear of piston ring grooves, the speaker's experience was the same as the Author's, namely, that this wear was greatest with the older engines having pistons cooled by sea water. In analysing the deposit in such grooves, both chlorine and sulphur had been found, indicating that either sea water or condensed combustion products might have caused the excessive corrosion wear. It was also quite certain that with oil-cooled pistons, as with hot fresh water, the wear was negligible. Had the Author definite evidence that it was condensed combustion products rather than splash from sea water which caused the corrosion wear?

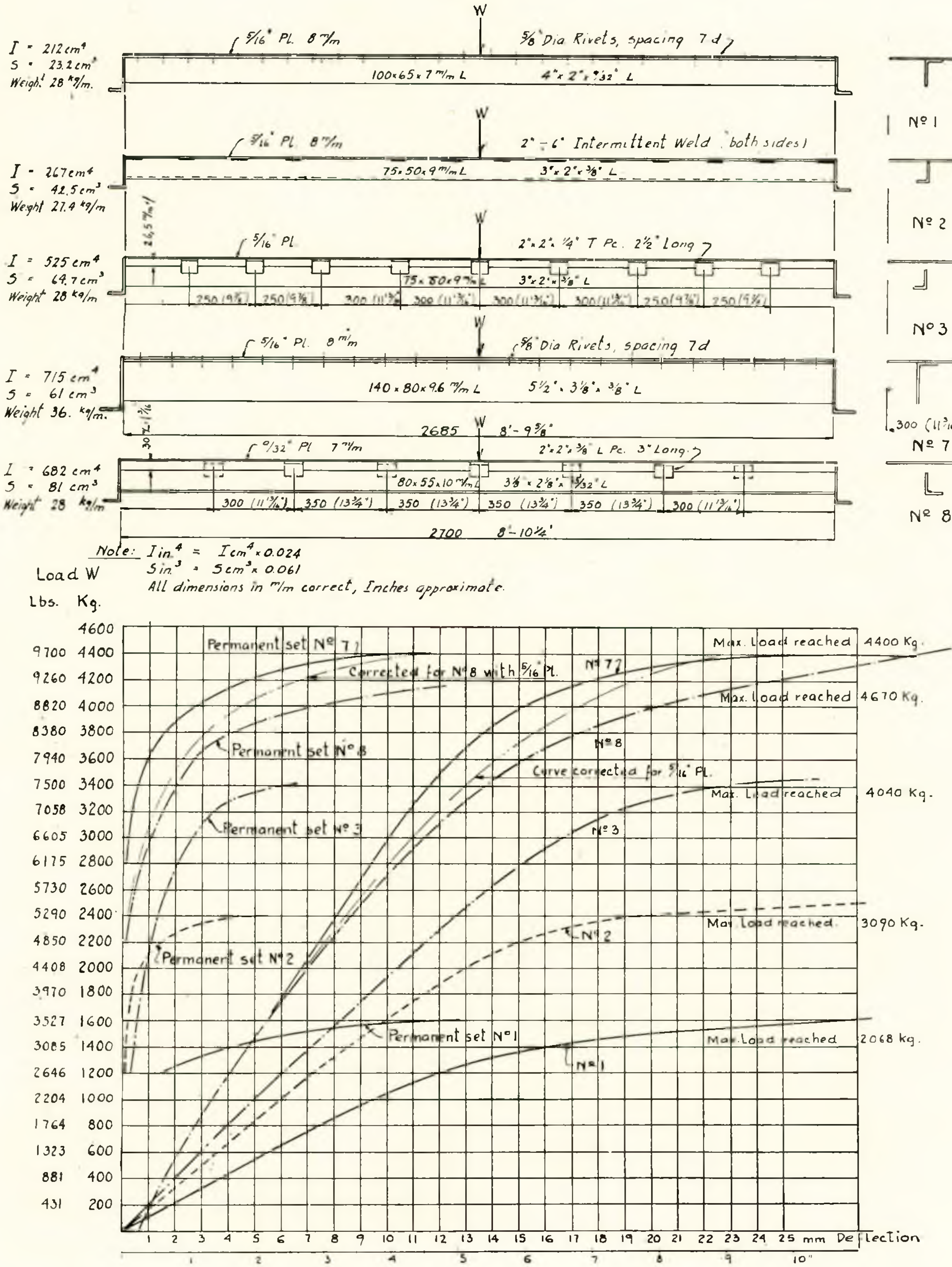

W
5/16" Pl. 8 m/m
5/8" Dia Rivets, spacing 7 d
I = 212 cm⁴
S = 23.2 cm³
Weight 28 kg/m.
100×65×7 m/m L
4"×2"×9/32" L
No 1
5/16" Pl. 8 m/m
2" - 6" Intermittent Weld (both sides)
I = 267 cm⁴
S = 42.5 cm³
Weight 27.4 kg/m
75×50×9 m/m L
3"×2"×3/8" L
No 2
5/16" Pl.
2"×2"×1/4" T Pc. 2½" Long
I = 525 cm⁴
S = 64.7 cm³
Weight 28 kg/m
75×50×9 m/m L
3"×2"×3/8" L
No 3
5/16" Pl. 8 m/m
5/8" Dia Rivets, spacing 7d
I = 715 cm⁴
S = 61 cm³
Weight 36. kg/m.
140×80×9.6 m/m L
5½"×3⅛"×3/8" L
No 7
2685 8'-9⅝"
9/32" Pl 7 m/m
2"×2"×3/8" L Pc. 3" Long
I = 682 cm⁴
S = 81 cm³
Weight 28 kg/m
80×55×10 m/m L
3⅛×2⅛"×9/32" L
No 8
2700 8'-10¼"
Note: I in⁴ = I cm⁴ × 0.024
S in³ = S cm³ × 0.061
All dimensions in m/m correct, Inches approximate.
Load W
Lbs. Kg.
Permanent set No 7
Corrected for No 8 with 5/16" Pl.
Permanent set No 8
Curve corrected for 5/16" Pl.
Permanent set No 3
Permanent set No 2
Permanent set No 1
Max. Load reached 4400 Kg.
Max. Load reached 4670 Kg.
Max. Load reached 4040 Kg.
Max. Load reached 3090 Kg.
Max. Load reached 2068 kg.
mm Deflection

FIG. 28.

Mr. L. J. Le Mesurier (Member) drew attention to the Table showing the various fuels used in the engines referred to in Table II. Although the characteristics of these fuels varied enormously they had proved satisfactory for use in the types of engines referred to by the Author. It might perhaps be remembered, however, that these fuels were used in types of engines unlikely to be adopted in modern tankers as they were practically all of the air-injection type with the exception of the Doxford engine, and the auxiliaries were also air-injection engines running at slow speeds. The modern tanker would always be fitted with airless injection and the auxiliaries would be steam driven, or if Diesel engines, probably of the fast-running type. It was a point that tanker owners should bear in mind when ordering machinery for their vessels, to ensure that the type of engine fitted would burn the grades of fuel obtainable at the bunkering ports which the vessels would use. The Doxford engine was particularly insensitive to fuel, but nevertheless there were types of airless-injection engines to which insufficient attention was given in regard to their ability to burn a wide variety of fuels. One of the chief requirements was a well-cooled fuel atomiser, such as that fitted in the Doxford engine. Engines for new vessels should invariably be tested on fuels which were at least no better than the fuels they might be called upon to use in service. If that practice were adopted the experience gained during shop tests would help in avoiding difficulties in service when using various fuels.

If he might reply to one of Mr. Hunter's questions, hard asphalt was completely combustible. This was shown by the test made for the ash content of fuel, the ash content being the only material which was incombustible. The quantity of incombustible material in any Diesel fuel was something very small, probably never more than ·05 per cent., whereas hard asphalt might be anything up to 4 per cent. Hard asphalt was combustible, but it was generally considered to be more difficult to burn than the other hydro-carbons.

The Author had a very interesting paragraph on hull and shaft vibration. Hull vibration, while not leading to any structural damage, was nevertheless very important, because if a vessel was run at a particular speed which caused acute discomfort to the ship's staff, it was obvious that this speed must be avoided. In considering shaft vibration, therefore, it was important to select speeds which at the same time would not lead to hull vibration and the two problems of hull and shaft vibration must really be considered together.

In his remarks on methods of reducing liner wear, the Author described the latest method of floating attachment of the pistons in the Doxford engine. It was interesting to note that the upper piston was lubricated by grease and the bottom piston attachment by forced lubrication. The lubrication of parts between which there was only a small relative movement was always difficult, as it was almost impossible to form a continuous oil film as in the case of a shaft bearing. On the upper piston attachment there was a large bearing area and grease lubrication might be satisfactory, but it seemed less certain in the case of the bottom piston attachment where the spherical bearing area was relatively small and reliance was placed on an oil film.

In conclusion, he would draw attention to the remarkable degree of reliability of engines which, after many years' service, only required the cylinders opened up for examination once every fifteen months. This corresponded to something like 9,000 hours running, during which the engine had made about 54,000,000 revolutions. Presented in another form, this performance was equivalent to 225,000 miles service in a motor car averaging 25 miles per hour. It was apparent that this was a very high standard of reliability which in all probability could only be obtained by the adoption of relatively slow-running machinery.

Mr. E. F. Spanner, R.C.N.C., ret. (Member) said that the paper provided a great stimulus to designers. Two points, in particular, were of exceptional interest to the speaker. One was connected with the hull structure, the other with the overall economy of the machinery as affected by the waste-heat boiler.

On pages 6 and 7 of the paper were sections dealing with cleaning cargo tanks, periodic hull examination and corrosion in cargo tanks, matters of outstanding importance to those responsible for running oil tanker tonnage. For his part the speaker was quite certain that the

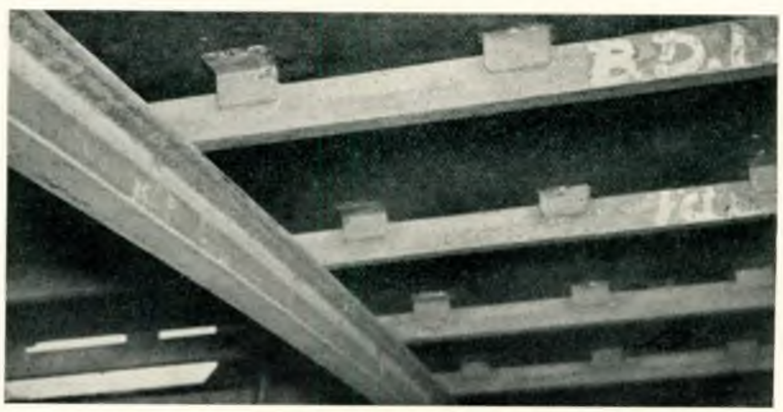


FIG 29.—Underside of deck with fore and aft beams and one transverse.

FIG. 30.—Fore and aft stern frames and part of stern post.

future would see great changes in the method of stiffening tanker shell and bulkhead plating, changes in which electric welding would play a very great part with benefit to the vessels and their owners. He wished to draw attention to certain recent advances in design which he felt sure would enable marked improvement to be achieved in the task of thoroughly cleaning tanks and maintaining the hull structure, in addition to offering a saving in initial cost and increased deadweight on given dimensions.

During the past two or three years a good deal of experimental work had been done by American and Norwegian interests in investigating the possibility of introducing an entirely new system of ship framing, aiming at increasing the effectiveness of diaphragm stiffening along lines explored theoretically by the speaker in 1932/33. Fig. 28 showed the results of authoritative loading tests recently carried out on diaphragms stiffened by angle members sustained in a position a little removed from the actual surface of the diaphragm, instead of being fitted close down to the surface.

The curves for series 1, 2 and 3 represented results for members of approximately the same weight, and showed that this new system of stiffening was a great improvement on stiffening riveted or welded to the plating in accordance with accepted practice. Riveted members gave way to a permanent set of about 2·7 to 3mm., intermittently welded members at a permanent set of about 4 to 4·5mm. and members of the new construction at about 6 to 6·5mm. Curves 7 and 8 represented results obtained for riveted members and members of the new construction of the same moment of inertia, with weights which were greatly in favour of the new system of framing.

These results were in accord with theoretical calculations placed before Lloyd's Register of Shipping and British ship designers generally by the speaker during 1932 and 1933, and there seemed good reason to hope that designs of large tankers incorporating these new proposals would eventually be produced. Figs. 29 and 30 were photographs of the actual application of the system in a small vessel now building.

Briefly, the advantages of the system for oil tanker tonnage were:—

(1) Reduced hull weight.
(2) Improved facilities for cleaning.
(3) Much better ventilation when freeing tanks from gas.
(4) Entirely sealed welded joints between stiffeners and shell.
(5) Elimination of practical difficulties in welding construction owing to short lengths of welded connections.

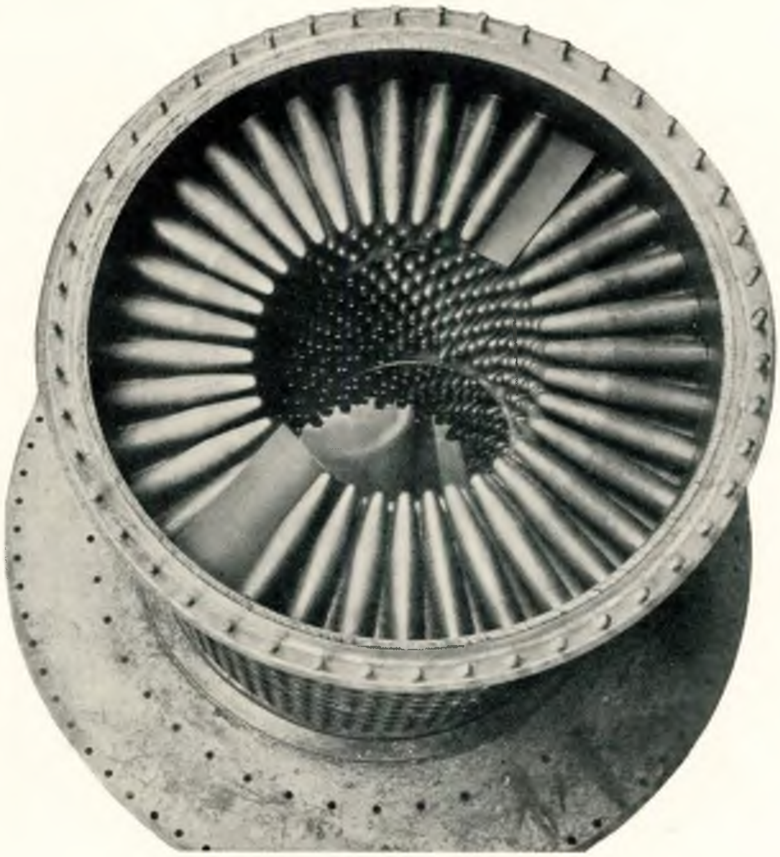
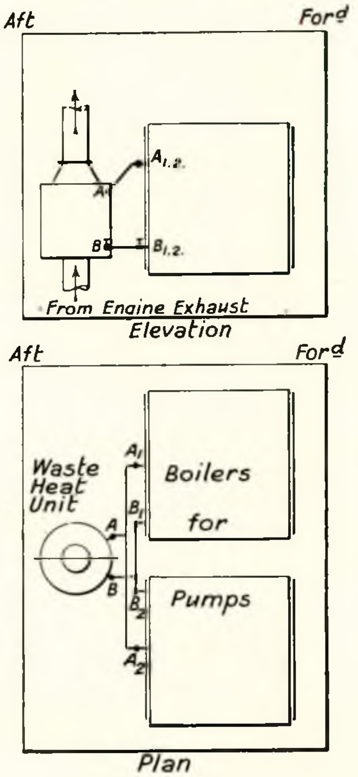


FIG. 31.—View looking down through tube nest.



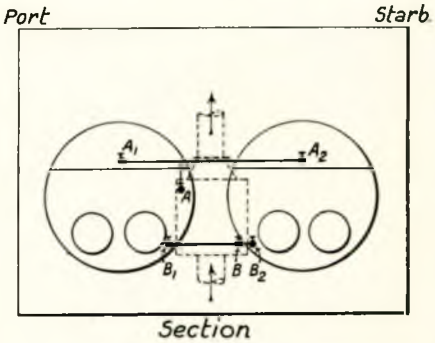


FIG. 32.—Boiler unit arranged to suit tanker requirements; A A_1 A_2 fullway valves on steam and water discharge line to cylindrical boilers; B B_1 B_2 fullway valves on water inlet line from cylindrical boilers; unit also fitted with safety valve, drain valve, mudholes and inspection holes; other boilers are ordinary standard marine type cylindrical oil-fired or coal-fired.

Brief specification for typical tanker, 12,000 tons d.w.:—Steam pressure 150lb. per sq. in., two oil-fired boilers each 11ft. 6in. dia. × 11ft. 6in. for port use, one waste-heat unit 5ft. 9in. dia. × 6ft. high to recover 1,000 B.Th.U.'s/h.p. from engines of 3,000 h.p. four-stroke; if two-stroke the size of unit would be increased approx. 10 per cent.

(6) Lower first cost and more scientific distribution of constructional material.

Incidentally, it might be pointed out that this type of construction deserved a second thought on the part of those responsible for insulated tonnage. Heat leakage would obviously be much less in a vessel having this construction, than in one of normal type.

Turning to the recovery of waste heat from the main Diesel exhaust, as the Author was already aware, the speaker had devoted a good deal of thought to the tanker problem, recognising that there was a specialised field for waste-heat boilers in tanker work, in view of the requirement which existed for a large steam output in port. It was found that the tanker problem was not possible of a simple and economical solution with a thimble-tube boiler of ordinary design, since the need for proper proportioning of the thimble-tube nest, in order to secure effective combing of the gases, inevitably led to this nest being of considerable length.

However, success had now been attained in producing a thimble-tube nest of greatly reduced height, which provided facilities not previously obtainable. A photograph of such a nest was shown in Fig. 31, in which photograph would clearly be noted the presence of two carefully designed spiral directional plates which, in conjunction with a central spiral deflector, caused the gas to swirl through the boiler at high speed under excellent forced convection conditions calculated appreciably to improve heat extraction. The production of a controllable spiral path for the gases in this way shortened the thimble-tube nest by roughly 40 per cent. without any loss whatever in heat recovery. The directional plates could be withdrawn from the tube nest if required.

This shortening of the tube nest was of great importance in the tanker problem and Fig. 32 showed that this new design permitted of a waste-heat unit being put forward to work with two ordinary standard cylindrical boilers. In these days high efficiency and flexibility in operation were sometimes set aside in favour of low initial cost. These new units had been specially designed to suit tanker owners and to enable them to equip their vessels with really efficient waste-heat recovery plant and at the same time to reduce initial cost. Fig. 33 showed the general design of this particular unit illustrated in the layout shown in Fig. 32.

Mr. H. V. Senior (Member) said he would have liked the Author to have dealt at greater

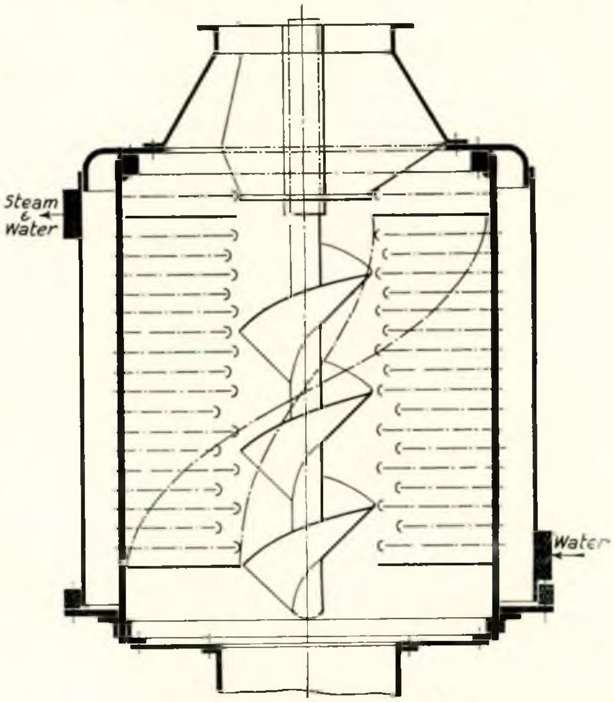


FIG. 33.

length with the auxiliary engines, but as pointed out by several speakers, the Author had covered tremendous ground for one paper.

With regard to the cooling water mentioned, the Author was not quite clear, but the speaker believed he suggested strictly adhering to distilled water. In this connection the speaker would like some idea of the cost, say, for the first fill up, as in his experience on some of the ships with which he had been recently connected, it would be inconvenient and costly to procure distilled water.

The Author mentioned that sea water was preferable to fresh water, but knowing that sea water *had* a tendency to make the outside of the liners and other internal parts soft and spongy, and in view of the Author's statement on page 14 that more wear was found in piston ring grooves when using sea water, on account of the lower working temperatures, the speaker could not quite see his point.

The auxiliary engines which the speaker's firm had supplied to one of the shipping companies used fresh water on a closed system through suitable sea-water tubular coolers, and as it had been found that there was little or no make-up water,

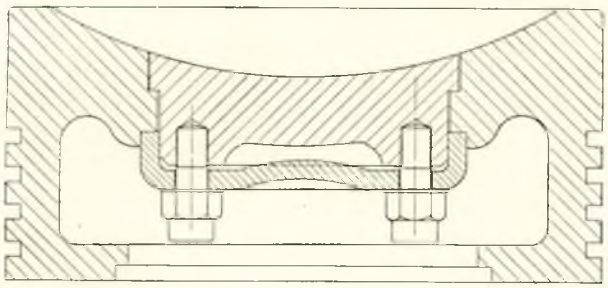


Fig. 34.

the amount of scale therefore would be very little and would not be continuously building up. After the scale was deposited it would be almost as good as distilled water.

With regard to the use of bichromate of potash in the water, which as the Author stated formed an anti-corrosive skin, he agreed that this was advisable; the Author, however, rather cautioned them in this regard in case any salt water should find its way into the system, thus forming hydrochloric acid. It might sometimes be difficult to stop a slight leakage of salt water into the fresh water through tubes in the cooler, and the speaker would like to know if trouble had been experienced in this direction.

On page 14 the Author mentioned heating the jacket water before starting. The speaker thought that all agreed this was ideal, but it was generally inconvenient and added to the cost. Had the Author found this more necessary with Doxford engines, on account of their lower compression, say, for easy starting? It would also be useful if Mr. Humphreys could give the approximate cost and the length of time it took to heat up the jackets to the necessary temperature.

The speaker presumed that the length of time between the grinding of the valves was adopted more as a protection, as he would expect the length of run to be much longer with complete combustion.

Had the Author any experience of using a special high flash point lubricating oil or special graphite grease or even graphited bushes on or for exhaust valve stems, to prevent the exhaust valve sticking, as very often the crankcase oil used was not entirely suitable for exhaust valve spindles, and in the speaker's opinion this oil should be separated from the exhaust valves.

Although the large fillet shown on the Sulzer liner on page 19 brought the pressure more in line with the joints and would assist in preventing distortion at the top of the liner, it had the tendency to increase the diameter of the spigot, thus increasing the diameter and putting more work on the breech end studs. One could see that it helped the foundry by thinning down the metal at the top end.

The screwed plug in the piston crown of the 4 S.C. engine on page 14 had many disadvantages besides being costly. One of the great disadvantages was that the facing would need to be ground, also if the plug was screwed in too tight, then the piston crown was just as liable to crack, more particularly if the co-efficient of expansion of the heat-resisting plug was different from that of the piston proper. He submitted a design (Fig. 34) which got over the above difficulty by using bolts. This enabled the heat-resisting plug to be ground on to its joint. The plate holding it had a slight spring with it, and it would be noted from the design that the face of the underplate made a second joint. This had been used by the speaker on some large pistons with success and he submitted it as an improvement on the one shown by the Author.

Mr. E. P. Paxman (Messrs. Davey, Paxman & Co. (Colchester) Ltd.) said that he had hoped to hear something from the Author on the subject of auxiliaries. If these main engines were providing their auxiliary service from exhaust-gas boilers, it would appear that auxiliary Diesels were only required when in port. From the figures given by the Author it appeared that these vessels were only about 60 days in port per annum, and he wondered why such stringent regulations were laid down for these auxiliary sets. It was not on account of wear as they were not used sufficiently, and he rather thought that Mr. Le Mesurier gave a clue when he stated that they had to use all kinds of fuels. The old engines with air blast were able to do this, but with the coming of solid-injection engines these were dependent upon a good-class fuel unless they ran very slowly. The earlier high-speed solid-injection engines required an exceptionally good class of fuel. To-day they were able to get a type of engine with combustion chambers having a very high and controlled degree of turbulence, and these engines were able to use a much larger range of fuels. He looked forward to a lessening of the stringency of the specifications.

With regard to the repairing of pistons shown by the Author, the speaker had had similar cases. The worst example he remembered was an 18in. bore uncooled air-blast engine. The piston used to get very hot in the crown, and it was not possible to make the pistons last very long until they hit upon a method of fitting a core. Because of the

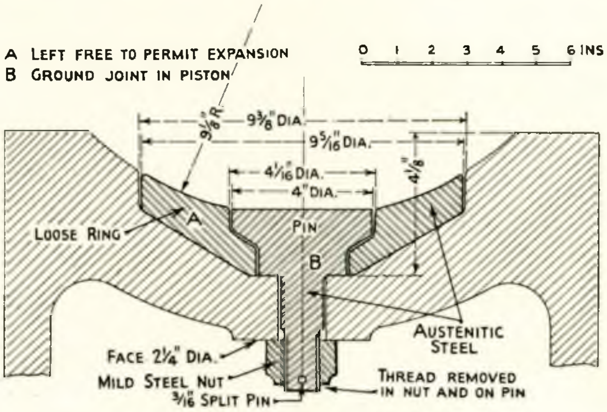


FIG. 35.

high temperature of the metal it was not possible to get the heat away from the centre core and it used to curl up if made in cast iron. They then used an austenitic material and the higher coefficient of expansion of that proved troublesome. The difficulty was overcome by making the core in two pieces. The centre part was ground on and the other part was a loose ring as illustrated in Fig. 35. The same principle when applied to smaller engines had always cured the trouble.

He was glad to see that the Author advocated distilled water cooling and that a good flow of water should be maintained. Only if a fresh-water cooling system was used was it possible to have a good flow of water with a high velocity without unduly cooling the engine. It was distinctly advantageous to have a good flow of water and to keep it as warm as possible.

The Author quoted figures up to about $^{300}/_{1000}$ths in. in connection with liner wear. While he accepted this figure it seemed amazing; it meant that at every stroke the piston rings were going in and out about ⅛in. laterally each side and that the gap of the rings was opening and closing about ¾in. at each stroke. The approximate amount of permissible maximum wear given by the Author appeared to be about $^{10}/_{1000}$ths in. per inch of bore. This was considerably in excess of what would be permitted for smaller engines. The liners of engines of from 15in. diameter down to 4in. diameter were more generally found unfit for service when the maximum wear had come up to about $^{6}/_{1000}$ths in. per inch of bore.

Mr. A. F. C. Timpson, M.B.E. (Member) said that in connection with ventilation the Author mentioned that any pressure or vacuum in the tanks could be observed from the "U" gauge and readily rectified. It would be interesting to know the pressure and vacuum limits he recommended and if there was any reason why these should not be controlled by an automatic valve. Designs were now available for installing at deck level so that they could be easily inspected and kept in working order.

In some tanker fleets relief valves were set to relieve pressure as high as 3lb. per sq. inch with a comparatively low vacuum setting. In other companies it was the practice not to exceed 1lb. per sq. inch. There would appear to be advantages in the higher pressure limit as with low flash point oils which tended to gas freely, the tendency to gassing would be reduced at the higher pressure limit.

It would also be interesting to have the Author's opinion on tank sprinkling with a view to reducing the tendency of the cargo to generate gas. This practice was recently introduced in the Panama Canal Regulations but was apparently previously adopted by some of the American companies for economic reasons.

The Author mentioned flame arresters fitted on the swan necks up the masts. The danger in fitting flame arresters at this point was that they received very little attention and might either choke or burst unless inspected at very regular intervals. There would appear to be advantage in installing flame arresters in the actual gas-freeing line at some point immediately above deck level. Gas mains were satisfactorily protected in this manner.

The Author also referred to the question of mixing through the vapour lines. In some cases this point was given very close attention and designs of relieving valves had been produced to relieve pressure into the gas-freeing line and to relieve vacuum from a separate air line, thus ensuring that once gas had passed the pressure valve it could not condense and be drawn back into another section of the cargo tanks. In other cases with vessels which were carrying mixed cargoes a common gas-freeing line was used and the question of mixing was evidently not considered important. What was the Author's experience on the question of mixing through the gas-freeing lines?

With regard to corrosion in cargo tanks, there were various forms of protective dressing which would satisfactorily withstand oil and spirit, but probably the cost of maintaining a protective dressing with these special compounds in cargo oil tanks would exceed the cost involved in renewing plates when this was necessary, and although the Author's method of spraying with lubricating oil was by no means a solution it was probably the only practical method which could be utilised at reasonable cost.

Referring to Diesel fuel, the Author's suggestions on obtaining efficient separation were extremely sound, in particular the suggestion regarding centrifuging at a slow throughput. The centrifuge, of course, had the limitation that it would only remove impurities in suspension which were heavier than the oil, but it did effectively remove the bulk of injurious impurities. Possibly centrifuging followed by very fine straining would be the ideal solution and this might be economic with tanker vessels burning up to 12 tons a day, but the cost of fine filters for larger-powered Diesel engines would be excessive and at the present stage of development the centrifuge appeared to be the best solution.

With reference to lubricating oil, a point frequently overlooked in fitting centrifuges to different types of engine was that some designs were arranged with the cylinder entirely isolated from the crank case and the piston rod working through a wiper gland. The piston-cooling mechanism was also arranged outside the crank case so that the oil in the lubricating system suffered little from contamination. Other designs of engine, however, were arranged with the piston open to the crank case and the piston-cooling mechanism also inside the crank case. Under these conditions the oil suffered to a considerably greater extent from contamination and a much more ample provision should be made for purifying it.

Regarding piston rings, the Author showed a design with a single sealing ring incorporated on the working face of the main ring, generally referred to as a double-seal ring. He also mentioned the triple-seal ring where two smaller section rings were incorporated in the working surface of the main ring. While these designs were satisfactory and ensured better compression, there was the difficulty with breakages and of renewing these smaller rings, which was a factory job. A design of ring was made by the principal piston ring manufacturers incorporating a top section and a bottom section, the axial width being the full width of the ring. A cod piece was fitted in either half of the ring at 180° to the butts, the complete ring being fitted with one cod piece pointing upwards and the other downwards registering with the butts of the adjacent ring. With this design a new half ring could be fitted without difficulty when necessary.

Mr. J. Reid (Visitor) said the statement that "the necessity of good engine seatings need not be stressed" seemed to imply that the Author desired them to believe that enough had been said on this subject. The speaker did not accept this view. Every part of a seating in Diesel tankers should be drilled and the seatings built up and riveted in large sections. All floors should be deepened and carried right out to the bilges, and all the plates on which the Diesels were coming down faired to a tolerance comparable with that obtaining in the Diesel engine bedplate itself. It was not a solution to punch holes in any part of the seating $\frac{1}{8}$in. small in diameter and then reamer them in place. How was it possible to put proper rivets in such confined places? The Author would no doubt confirm that if there was crank trouble and breakage, it could be attributed frequently to improper seating, and certainly any excessive vibration in the ship could fairly be attributed to this cause.

The old Scotch boiler never seemed to be dispensed with. The new exhaust-gas boilers appeared to be Scotch boilers rejuvenated. Some years ago in a certain tanker which carried very volatile oil, there was from time to time an awkward discharge from the funnel of large pieces of flaming carbon. The speaker found that a great deal of combustible oil-gas was being given off from the two engines and that at certain parts of the exhaust this combustible collected until it formed a piece large enough to be blown out. He recommended as a remedy that this combustible should be burnt out, and it was arranged to put the exhaust through a bank of tubes which had been preheated in one of the two auxiliary boilers on board. They managed to get the same boiler tubes to take the exhaust and the gas from the oil burners as well. Everything went satisfactorily for a while; there was plenty of steam, no discharge from the funnel, and then pieces of smokebox doors, furnace fronts, etc. came flying past. After this the owners were satisfied that it would be better to go back to the old scheme in the hope that the soot would go overboard.

He noticed after investigating the speed of the gas flow that builders of exhaust boilers of the Scotch type had a bad habit of discharging the gas into the combustion chamber through a very coarse and badly located cylindrical port. That should never be done, but an effort made to distribute the gas fairly across the tube bank. Every effort should also be made to conserve the gas velocity, so that some form of heat interchanger could be carried in the tubes and more steam generation obtained. Five good furnaces were required to get steam when pumping a very viscous cargo, and it was an annoyance to find that the furnaces had to be started up two or three days before reaching port.

On the proposal of **Mr. J. Carnaghan** (Vice-President) a very cordial vote of thanks was accorded to the Author.

By Correspondence.

Mr. K. O. Keller (Member) wrote that under the heading of "Lubricating Oil" the Author referred to the claim made that colloidal graphited oil reduced cylinder liner wear, and the writer would like to ask whether he had any experience to substantiate such claim.

With regard to the treatment of bearing oil, an improved method to the "trickle of hot distilled water" was the injection of live steam through a spiral jet just prior to the oil entering the centrifuge, a device incorporated in the Sharples machine.

With regard to piston and jacket cooling, the writer was glad to observe the Author's distinction between distilled-water and fresh-water cooling, although the writer believed he was the first to adopt the former in 1920; it was perplexing to observe that even to-day so many shipowners still accepted salt-water cooling when distilled water had every recommendation except a slight extra initial cost.

In amplification of the Author's statement in regard to clean pistons, the writer would add that the same held good for cylinder liners, the surfaces being as clean after four or five years' service as when they left the builders.

The Author's tabulation of liner wear deserved every compliment and should be emulated by other owners.

With regard to pistons, the Author had given a sketch of a piston with a nickel cast-iron centre plug screwed in, and he would like to know whether, in the Author's experience, this had proved satisfactory in service. Not long ago the writer was consulted on how to overcome the difficulties of pistons repaired in this way. This case concerned a great number of pistons of a four-cycle engine repaired with a cast-iron centre using different kinds of threads, but the plugs were invariably found loose after 30,000 miles' service. It was conceivable that the quality of fuel oil and mean pressure had a bearing on the success or failure of such a makeshift.

One important point the Author appeared to have overlooked was in regard to engine seatings and holding-down bolts. The writer had no doubt that in the fleet under his care the Author had to deal with built-up engine seatings and engines bolted direct to the tank top. It would be interesting to hear his experience with the respective methods.

Mr. H. Mackegg (Associate) wrote that he noted with interest that special emphasis was laid upon the fact that both the lubricating and fuel oil should be "carefully centrifuged" and that remark raised the question of the separating capacity of centrifuges. In view of the numerous factors which had to be taken into consideration, such as oil characteristics, temperature, condition of oil, etc., it was impossible to arrive at any definite formula which would produce a given degree of purification, and therefore a large throughput capacity margin should be allowed when the question of the size of centrifuge was being considered. He would also suggest that the question of centrifuge bowl design should always be thoroughly investigated before any such decisions were made.

Regarding Diesel fuel oil purification, the writer had found that because totally-enclosed gas-tight machines were available there was no objection to raising the temperature of viscous fuel oils to 140° F. and under those conditions most efficient purification could be obtained at a high throughput.

Because a centrifuge would effect complete dehydration, there should be no objection to the introduction of a small stream of wash water into the centrifuge with the fuel oil, and the wash water would sludge out the solid impurities from the centrifuge bowl, thus making it possible to run the fuel oil centrifuges for very long periods.

The same remarks applied to the lubricating oil centrifuges, especially in regard to high capacity machines, and it had been found that the use of brine as wash water greatly improved the degree of separation; also, because the specific gravity of brine was nearer to the specific gravity of the separated solids, continuous sludging was maintained throughout the whole running period, and records which had recently been brought to the writer's notice showed that the lubricating oil purifier in a large Diesel ship, running at a capacity of 800 gallons per hour with a brine feed of 25 gallons per hour, ran continuously and was only stopped for cleaning purposes once every four days. It was normally understood, of course, that distilled water or condensate was introduced into the lubricating oil to correct possible acid formation, because the condensate absorbed water soluble acids. The same results, however, were obtained by the use of brine, and over a period the absorption was greater because of the higher capacity which was maintained for long periods.

The question of the speed of centrifuges was of course important, but in these days direct-coupled machines were available, thus ensuring that the centrifuge speed was constant throughout the whole running period.

Mr. P. N. Everett wrote that builders of engines of all types and sizes would do well to take notice of the numerous practical points in the paper, as they were suggestions which arose out of the working of engines in service, which was the only true test. In the running and maintenance of the engine there were many valuable hints to be borne in mind, such as the clamping down of the liner when removing a piston to avoid dirt and scale getting under the joints, the use of draw cards and the recommendation not to remove piston rings unnecessarily. But the most important point of all in the writer's opinion was the Author's recommendation for ensuring clean oil reaching the engine. It was of the utmost importance, as even in tanks taking centrifuged oil it was possible for contamination to occur. No precautions which the engineer took in this connection were too great, and a constant supervision of this work should be maintained as the results of even a small amount of contamination passing through the injection system could be extremely serious and might not show up until some time later when the real reason for the trouble was obscured. It was particularly essential to watch the cleaning of the oil when changing grades or sources of supply.

He would especially recommend to the notice of builders of auxiliary Diesel engines the table of fuels given on page 9. Very few owners of motorships could pick and choose where fuel was concerned; they must take what was offered. Also, very few ships were fitted with special tanks for the auxiliary engines, and these engines must use the same fuel as the main engines. There was a great tendency to speed up auxiliary engines, and while these engines were very successful in the services for which they were designed, it must be remembered that in these services they received the fuel they needed and when selected for ship's service the fuels available must be considered. For the same power, high speeds meant a larger number of smaller cylinders, and the smaller the cylinder and the higher the speed the greater was the necessity for a high-grade fuel. If space was the consideration, then it might be that the horizontal engine could in some cases be usefully employed as, with four cylinders on two cranks, a very convenient unit could be designed.

The Author had said very little about the maintenance of the Doxford sprayer and such information would, the writer thought, be very interesting now that most engines were being built with airless injection. For instance, how many hours did the sprayer run without attention, and during say twelve months' service what, if any, were the cases of choked holes, always providing that clean fuel was used?

Dr. S. F. Dorey (Vice-President) wrote that the Author had given a straightforward account of the care necessary for the maintenance of the Diesel-engined tanker which should prove of special value to the sea-going engineer. A number of points had been raised which were familiar to marine engineers who would appreciate that neglect of them would have serious results, involving perhaps a great deal of expense to the shipowner. While the paper contained a number of points of practical interest which would no doubt be discussed by those most interested, the writer proposed to confine his remarks to the small section dealing with vibration.

The Author rightly mentioned the dependence of vibration stress at the various orders upon (*a*) firing order, (*b*) crank arrangement, (*c*) the form of the normal elastic curve. These were three important points because, in the design stage, consideration had to be given to each one with a view to reducing the vibration stress at orders occurring near the running speed. Alterations to (*a*) and (*b*) were obvious; in the case of (*c*) the form of the normal elastic curve could be altered by choosing the best position of the flywheel, i.e. at the forward or after end of the engine as the case might be. Alternatively, two flywheels might be fitted, one at each end of the engine. In this way the elastic curve could be made symmetrical about the centre line of the engine, in which case so-called major critical speeds would completely cancel out, but on the other hand minor criticals would become important.

If consideration of (*a*), (*b*) and (*c*) did not solve the problem of critical speeds for the desired speed range of engines then any of the following methods could be applied, viz.:—

(i) The stiffness of the shafting increased.
(ii) A friction damper fitted.
(iii) A synchronous vibration absorber fitted.
(iv) A detuner of the Doxford-Bibby type fitted.

The application of (i) above was perhaps the best solution for a large speed range, but it had the great disadvantage of increasing the weight of moving parts and the first cost of the engine.

Friction dampers had the disadvantage of requiring very careful adjustment in order to obtain optimum value of the friction force between the flywheel and the disc attached to the shaft. Small changes from the optimum value led to a very rapid falling off in the efficiency of the damper.

Now the Author had mentioned the synchronous vibration absorber and perhaps he would give more detailed information of experience gained with this form of damper since it had only comparatively recently been adopted for marine engines, although the principle was an old one.

Reduction in vibration stress to a value of one-third and one-half were claimed for the Doxford-Bibbly detuner, and the principle appeared a very recent application. Perhaps the Author would also state any experience he had had regarding changing the form of the elastic curve by the fitting of flywheels or other devices.

The Author's Reply to the Discussion.

The Author, in reply, thanked the contributors to the discussion of the paper for their valuable criticisms.

With regard to the questions raised by Mr. Thomas, it was essential to scale thoroughly the tank bulkheads before spraying. The light lubricating oil was sprayed at about 60lb./sq. inch and appeared to soak into the plates. It was not claimed that this would cure corrosion but it had been observed that the bulkheads did not deteriorate so quickly after treatment by this method. It was a case of taking the half a loaf which was available.

Piston heads should not be repaired by welding since allowance must be made for expansion. The plugged piston shown in Fig. 12 had been commented on by several speakers. The Author had found that the best results were obtained when the shoulder of the plug was made gastight. The threads should not be too tight and allowance should be made for expansion at the periphery of the plug flange, the plug of course being located by dowel pins. The plug should be made of nickel cast-iron since this material did not "grow" with heat.

The Author was indebted to Mr. Youldon for his constructive remarks and agreed with him that cast-iron gave better service than steel for cargo piping, although the former was much heavier. It was also agreed that soft packing produced better results than lead shavings for expansion glands in the tanks. He was pleased that Mr. Youldon called attention to an oversight with regard to the heating of creosote. This was generally heated to between 90° F. and 100° F., since, if allowed to cool, naphthalene salts would deposit; nevertheless the main point was that the heating coils must be used to keep the cargo "pumpable", otherwise some 10 per cent. would settle out and the cargo could not therefore be discharged by the pumps.

In reply to Mr. Hunter, the Author could see no reason for fitting twin-screw machinery in the average size vessels shown in Table 1, since the reliability of Diesel machinery was well established and the powers were well within the range of single-screw machinery. A tanker of, say, 17,000 tons displacement on 27ft. 6in. loaded draft, would be ballasted with three or more tanks of water and trimmed two or more feet by the stern; the draft aft would be usually 18ft. 0in. or more according to weather conditions, so that the propeller was generally immersed and any advantage gained by the increased immersion of twin screws would not be so great as in the case of ordinary cargo vessels in ballast. From his experience he could see little gain from a propulsion point of view in the average class of tankers, but the initial cost and maintenance of twin-screw machinery would be increased. For high speeds and very large capacities and in special cases of shallow draft vessels, twin-screw machinery might be necessary.

Corrosion in the tanks was found to be worse in tanks mostly used alternately for the carriage of benzine and water ballast, irrespective of the position of nodes, and was partly due to corrosion-fatigue as distinct from fatigue. In the Author's

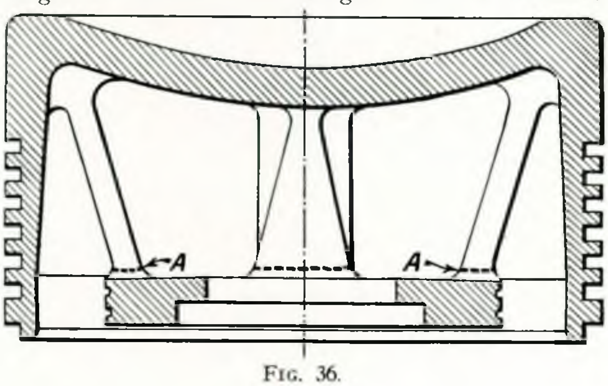


Fig. 36.

experience, the corrosion was no greater in vessels fitted with Diesel engines than in vessels with turbine or steam reciprocating machinery.

The 90° mechanism for indicator diagrams was so simple and the information gained so valuable that the Author considered it well worth while fitting to any type of main Diesel machinery. The Author did not consider it good practice to anneal bottom-end bolts, and with 4 S.C. generator engines running at about 400 r.p.m., it was his practice to scrap these after, say, 10,000 hours' running. The life of the bolts would, however, depend on several factors, e.g. the stress, the number of repetitions of stress, and the designs of the engine and the bolts. The importance of identification marks on all bolts of special material should, however, be emphasised.

Cracks in pistons and cylinder liners and cylinder heads occurred on the side which was subjected to the greatest tension. For instance, it would be observed from Fig. 36 (ribbed piston head —2 S.C.S.A. engine) that the cracks started at the ribs at positions "A", which were caused by increased stresses due to bending when the piston crown expanded with heat. In Fig. 8 (cylinder head—2 S.C.S.A. engine) the cracks invariably commenced at the underside and adjacent to the fuel valve aperture. In some cases, where the cracks were not too deep, the covers had been saved by machining across the underside, thus reducing the thermal stresses, care being taken to leave sufficient metal to provide for the pressure stresses.

Chlorine deposit found in piston grooves might be due to three causes, viz. (1) sea-water leakage where salt-water cooling was employed, (2) salt-laden air drawn into the cylinder, (3) sea-water leakage into the fuel which should have been removed by efficient centrifuging. In the last two cases this might happen irrespective of whether sea-water or distilled water cooling was used. While the Author had no direct evidence from actual analysis of the presence of chlorides in carbon deposits taken from piston ring grooves resulting from sea water, it might be assumed that if sea water was present the chlorides would, under certain conditions, form hydrochloric acid, which was well known to be a potent cause of corrosion. The point the Author wished to stress was that if condensation was not present corrosion would not occur, so that distilled-water was preferable to sea-water cooling as higher temperatures could be maintained.

The Author was indebted to Mr. Le Mesurier for his valuable contribution and would emphasise his remarks that engines for new vessels should invariably be tested on fuels which were at least no better than the fuels they might be called upon to use in service. Mr. Hunter's question with regard to hard asphalt had been dealt with adequately in Mr. Le Mesurier's remarks, which needed no repetition.

Mr. Le Mesurier had commented on hull vibration and a most useful written contribution had been received from Dr. Dorey with regard to vibrations. In reply to Dr. Dorey, the Author had no direct experience with regard to maintenance of dampers or detuners. In some engines a flywheel had been fitted at each end of the engine, and provision was made for fitting a damper in place of the forward flywheel if found necessary.

Investigations into hull vibration were carried out in one tanker. The vessel was 440ft. 0in. length b.p., 57ft. 0in. beam and had a displacement of 15,000 tons on 27ft. 0in. draft. Vibrograph readings were taken in suitable positions and it was found that the worst vibrations occurred at 83 r.p.m. loaded (26ft. 6in. mean draft) and at 86 r.p.m. ballast (18ft. 0in. mean draft). The gravest frequency observed in loaded condition was 332~/min., i.e. four times the revolution speed. These speeds were avoided in service. The vibrations ceased at 2 to 3 r.p.m. away from these speeds.

As to torsional vibration a 4 S.C.S.A. engine was found to vibrate excessively in way of the thrust block at 93 r.p.m., the first degree frequency was 372~/min. In all probability a very short amount of running at this speed would have resulted in shaft failure, and it was therefore imperative to place this critical speed outside the range of service speeds. A short length of intermediate shaft of 24in. diameter was therefore substituted for the original one of 14¼in. diameter, thereby raising the natural frequency to 445~/min. and the dangerous critical up to 111 r.p.m., which was well above the normal revolution speed. In another case in a 2 S.C. engine the service speed of the engine was about 75 r.p.m. and the second degree, 10th order critical was found to be at 73 r.p.m. As the 10th order was a somewhat dangerous one and 73 r.p.m. was too near the service speed, 3 tons of material were machined off the 13 tons flywheel, which brought the frequency up from 730~/min. to 790~/min., so that the 10th order, viz., 79 r.p.m., was well above the service speed. These examples illustrated how, in service, a dangerous critical could be raised by increasing stiffness or by decreasing masses. A critical could, of course, be lowered by the opposite changes.

Mr. Spanner had shown some very interesting drawings and while it was appreciated that maintenance and design were closely connected, the construction details were somewhat outside the scope of the paper. Undoubtedly welding would be more used in future in connection with both hull and machinery. Well-known engine builders were constructing Diesel machinery with fabricated bedplates, crankcases and columns, while rapid strides were being made in welding in connection with ship construction.

Although the Author agreed that a specially designed waste-heat boiler would be a little more efficient than the arrangement shown in Figs. 26 and 27, the additional first cost and maintenance of present designs hardly warranted the very small saving in fuel. The average figures given by the Author were obtained with a boiler pressure of 120lb./sq. inch (steam temperature 350° F.) and with feed water at 100° F., and where the exhaust gases left the boiler at between 425° F. and 450° F. In the Scotch boiler double-pass design, the upper nest of tubes was made about 10 per cent. less in area than the lower nest to compensate for the denser cooling gases and in order to maintain approximately the same velocity through the tubes. The dew point of exhaust gas was less than 212° F., so that there was no sulphuric acid formation above this temperature and no corrosion had been found due to exhaust gases. There was very little soot deposit in the tubes provided that excess lubrication to the engine cylinders and unburnt products, due to incomplete combustion, were avoided. No thermal gain apart from a reduction in radiation losses could be obtained in a boiler specially designed for exhaust gases. Where space was limited and a Scotch boiler was not fitted, there were advantages in fitting a special waste-heat boiler; also where a pair of Scotch boilers were fitted, and if space was available, the layout shown in Fig. 32 would have the advantage that the special exhaust-gas boiler could be used in conjunction with either boiler.

With regard to the question of cooling water raised by Mr. Senior, the Author was in favour of distilled-water cooling. The reduced cost in maintenance and the saving of delay far outweighed the initial extra cost and the maintenance of a cooler and distilled water pumps. Cracked covers and other defects which arose through scale formation when sea water was used were not only an expense but also caused delays. In addition, the Author attached great importance to the higher jacket temperatures which could be maintained when distilled water was used. Distilled water was preferred to fresh water on account of the solid impurities in the latter. Doxford engines required heating, in any case, to facilitate starting on account of their lower compression. The distilled water was tested periodically, and it was, of course, necessary to avoid leakage in coolers. The jackets of a 3,000 b.h.p. engine could be warmed thoroughly in a closed circuit in about three to four hours, and the fuel consumption for this purpose did not exceed half a ton. The capacity of the distilled-water tank and jacket system varied, but assuming this to be about 8 tons, the evaporator and distiller would be capable of supplying the first charge on about 1 ton of fuel. A heating coil was fitted in the distilled water tank, and the tank was periodically cleaned.

Practically no trouble was experienced with exhaust valves of 4 S.C. engines sticking. Light lubricating oil was used sparingly for this purpose.

The Sulzer liner shown in Fig. 22 had the position of the joint in the same place as in Fig. 21.

The reason for the larger fillet was to reduce the mass of metal. The design shown in Fig. 24 was an improvement, seeing that both the bottom of this liner and the exhaust and scavenge port bars were uncooled, and the mass of metal at the top of the liner was reduced, thereby resulting in a more uniform temperature throughout the liner and less tendency for condensation towards the bottom of the liner, with a consequent reduction in cylinder liner wear.

The pistons illustrated in Figs. 34 and 35 were most interesting and the Author was indebted to Mr. Senior and to Mr. Paxman for these. Both designs of plugged pistons served to illustrate the necessity for making allowance for expansion.

In reply to Mr. Paxman, the Author stated that many modern tankers were fitted with Diesel generator engines. In many cases one steam-driven generator and one Diesel-driven generator were installed. The main engines might be self-contained with cooling and forced lubrication pumps. The independent stand-by pumps and other auxiliaries might be either steam or electrically driven, but generally there was a proportion of each. The Diesel generator engines should be able to run on the same fuel as the main engines, and the speeds had increased during the last few years from about 300 r.p.m. to 600 r.p.m.

With regard to cylinder liner wear, the highest figure quoted by the Author was $^{250}/_{1000}$ths in.; this was considered the limit for 4 S.C. engines of about 30in. cylinder diameter and $^{225}/_{1000}$ths in. for 2 S.C. engines of similar size. He agreed with Mr. Paxman's figures for smaller bore cylinders.

Mr. Timpson had raised the point regarding pressure and vacuum limits in the cargo tanks. With the "U" gauge method these could be observed in the wheelhouse by the officer of the watch, and when any appreciable change took place in the level of water in either leg of the gauge, the vapour valve at the foot of the mast was opened which released pressure or broke the slight vacuum in the tanks as the case might be. With regard to mixing through vapour lines, the Author had experienced this due to condensation of gases in a common vapour line. In modern practice, separate vapour lines were generally fitted to each section of tanks. In any case, when "mixed" cargoes were carried in vessels having only one vapour line, care should be taken to see that the vapour valves of tanks containing different parcels of cargo were not opened at the same time. The flame arresters were cleaned periodically with the vapour lines.

In tankers used for special cargoes, e.g. for the carriage of casinghead products which gassed freely, it might be desirable to carry a pressure in the tanks and to fit relief valves set to 3lb./sq. inch. It might also be advantageous to fit tank sprinkling to reduce gas generation. Otherwise the Author considered this pressure somewhat high. He would, of course, fit tank sprinkling when trading through the Panama Canal in order to comply with the Regulations, but would not do so for economic reasons.

The Author was pleased that Mr. Reid had raised the matter of engine seatings. The words which had been quoted from the paper were certainly not intended to convey his interpretation of them. Well-constructed engine seatings were essential, but more was not said as the Author felt that in a paper of this description, construction details and design could only be touched upon.

Colloidal-graphited oil had been tried on a small experimental engine and the results had justified a trial in a Diesel tanker. The Author regretted that he could not give Mr. Keller any results at present, as, although he had a case under observation, the colloidal-graphited cylinder oil had only been in use for a few months and he did not wish to jump to hasty conclusions.

With regard to the service of plugged pistons, these had been in service a good deal longer than the time stated by Mr. Keller and had so far remained tight. It should, however, be pointed out that the engines were run with mean indicated pressures corresponding to those given in Table II.

There was no doubt that the best arrangement was to bolt the engines direct to the tank top, rather than on box-girder built-up seatings. Recessed double-bottom tanks were sometimes fitted, but the deeper tank in way of the bedplate holding-down bolts had the effect of reducing the engine room measurement and might cause inconvenience in obtaining the 32 per cent. tonnage deduction for machinery space.

Mr. Mackegg's remarks regarding oil purification and brine water washing of lubricating oil were very interesting. The Author had, so far, kept to distilled water washing as he did not take too kindly to the possibility of adding sea water to the lubricating oil system in the event of faulty separation.

Mr. Everett's remarks regarding fuels for high-speed Diesel generators were sound and endorsed what had been said on this subject.

As regards the Doxford sprayer, the fuel valve was examined about every 150 days' running, when opportunity was taken to examine the spray plug. It was only on rare occasions that a sprayer hole became partly choked and there was practically no carbon formation. The spray plug was fitted with five angled jets; each jet was generally $^{22}/_{1000}$ths in. in diameter and should be renewed when the bore exceeded $^{27}/_{1000}$ths in. Tapered instruments should never be used to clean the spray holes as they tended to open them and shorten their life. The jets should only be cleaned by means of a spiral drill of the same diameter as the bore of the jet. The reason the spray plug gave such little trouble was, no doubt, due to the excellent method of cooling the fuel valve; also a very efficient duplex filter was fitted between the fuel pump and fuel valve.

Fuel Combustion Adjustments of Diesel Engines.

By JOHN LAMB (Member).

Although combustion efficiency primarily depends upon factors over which the operating engineer has no control, such as the design of the fuel injection device, the shape and dimensions of the combustion space and other matters governed by thermodynamical considerations, yet factors over which he has control such as the degree of atomisation, penetration and, to a certain extent, turbulence as well as final compression pressures and temperatures and the correct timing of the injected fuel charge, are all equally essential considerations. As operating engineers, they are not required to deal with the former factors, which, it may safely be assumed, are sufficiently correct to burn any Diesel fuel efficiently, their work being mainly to see that the various devices provided to ensure proper combustion function properly.

To ensure proper combustion in oil engines the fuel must enter the cylinders in such a state and at such a rate that it will burn and maintain more or less the pressure in the cylinder during the process. If the fuel burns too slowly after-burning will take place. In addition to increasing the fuel consumption per unit power developed by the engine, the result of after-burning is to cause abnormally high cylinder temperatures throughout the working and exhaust strokes of the cycle, which may lead to cracking of pistons and cylinder covers, excessive cylinder liner and piston ring wear, burnt exhaust valves and other equally serious operating troubles. The effect of the fuel burning too readily is to cause overheating of the fuel injection nozzle. This may result in carbon depositing on the nozzle and affecting the fuel spray to such an extent that after-burning and the serious consequences which result therefrom will take place.

At one time low viscosity gas oil was generally used as fuel for Diesel engines. Such fuel requires a comparatively low degree of atomisation, and in most engines no special effort is required to obtain the desired degree of penetration and so maintain the normal high combustion efficiency. This grade of fuel is, however, expensive and although still used to a small extent, most ships are now supplied with more viscous grades and the trend is toward even heavier and cheaper fuels. Airless-injection engines still require to be supplied with a slightly better fuel than air-injection engines, but the rapid progress being made with the former type of engine leaves no doubt that the time is not far distant when the inferior grades of fuel will be burnt efficiently.

The more viscous fuels are naturally more difficult to break up into the very small particles so essential to ensure instantaneous ignition and complete combustion. With an atomising device suitable for, say, gas oil, the particles of a more viscous fuel would be of comparatively large size and consequently take a longer time to burn to the core. In a slow-speed engine the difference in the rate of combustion would perhaps not be noticeable in the running of the engine, but if the speed of the engine is such that the time allotted for the process is insufficient, the particles of fuel would smoulder, as it were, and emit heat during the expansion and exhaust periods of the cycle, thereby causing an abnormally hot and probably discoloured exhaust. Even moderately excessive gas temperatures during the working and exhaust strokes will reduce the thickness of the lubricating oil film on the cylinder liner and result in greater wear of the liner and piston rings, while particles of partially burnt carbon will lodge in the exhaust valve faces and eventually cause them to leak.

Besides the possibility of the foregoing serious mechanical troubles arising from after-burning, the drop in thermal efficiency with consequent increase in fuel consumption per unit power developed can be very considerable without any outward indication apart from an increase in the temperature of the exhaust gases. And the temperature of the exhaust gases in conjunction with the m.e.p. developed is not always a reliable guide to the combustion efficiency, since the temperature of the exhaust gases is affected by the amount of injection or supercharge air admitted into the cylinder, viz. the greater the amount of air admitted the lower the exhaust temperature. For instance, to take a simple case, suppose the combustion efficiency of a particular engine is below the normal figure and the amount of injection air admitted to the cylinders is greater than usual owing to the pressure being too high or the lift of the fuel injection valve too great, the former fault would have the effect of increasing the exhaust temperature while the latter would tend to decrease it, so that it is quite possible for one to balance the other and so have a low thermal efficiency with a normal exhaust temperature.

From what has been written it will be apparent that careful consideration of the combustion efficiency is essential for more reasons than that of reducing the daily fuel consumption to a minimum. Moreover, it is not possible to state here just what adjustments should be made to Diesel engines in general in the event of imperfect combustion, because the construction of the atomising and injecting devices in general use varies so widely, as well as the physical properties of the fuels commonly used. The best results from any particular engine and fuel can only be obtained by making a very thorough study of the effect which the various adjustments have upon the combustion efficiency, and carefully observing the effect of any adjustment that may be carried out. The remainder of this article will, however, doubtless serve as a guide to the adjustments necessary for the solution of the combustion difficulties generally encountered.

Air Injection. When an engine of this type will not start or work satisfactorily even though the fuel is first heated, and it is known that the timing of the fuel injection valves is correct and their working conditions all that may be desired, then it may be assumed that the atomiser adjustments are not suitable for the fuel in use. Difficulties of this nature where the cause is high viscosity of fuel can sometimes be got over by simply heating the fuel and the water in the cylinder jackets to a temperature of about 120° F. and 100° F. respectively. It is, however, very rarely indeed that an engine in which the proper compression pressure and consequently temperature is present fails to start, so that when this occurs the first step should be to take an indicator diagram to make sure that the air is being compressed to the correct pressure before the fuel is condemned or the fuel atomiser adjustments altered. The most common cause of an engine failing to begin working on fuel is that the fuel is not injected into the cylinders as supposed due to the injection valves and pipe system not being properly primed, or to the fuel being blown back through leaking fuel pump valves by the high-pressure injection air. Water in the fuel is another probable cause of an engine failing to start.

If an engine fails to start working on fuel when the change in the grade of fuel is from one of high viscosity to one of a lower viscosity, the cause is always due to the fuel pump valves leaking, even though the engine previously worked with apparent perfection on the higher viscosity fuel. The explanation of this is that the lower viscosity fuel is generally thinner and more "searching" and will naturally leak through the small openings between the faces of the valves much more readily than the higher viscosity, thicker fuel.

In the event of an engine failing to begin working on fuel when the change over is from a light to a heavy fuel, and all the factors previously mentioned which affect the injection and ignition of the fuel are in order, the cause will be due to the different properties of the fuels and certain adjustments will have to be made.

Any grade of commercial Diesel fuel, no matter how viscous it may be, will ignite if injected into air at the normal compression temperature of from 1,000° to 1,200° F. and at the correct time relative to the position of the piston in the cylinder. If, therefore, the engine fails to begin working because the fuel is too viscous, it means that the fuel is entering the cylinder too late rather than that its degree of atomisation is not high enough to bring about ignition. Degree of atomisation is, of course, a very important factor, but it affects the combustion efficiency after the engine starts working rather than the starting qualities of the engine. No one would be so unwise as to start an engine with a quantity of liquid fuel lying on the top of the piston, because it is generally known that the oil would be ignited, probably before the piston reached the end of the compression stroke, and dangerously high pressures may be produced in the cylinder. This proves that if an engine fails to begin working on fuel because the fresh supply of fuel is more viscous, the cause of the failure is not that the fuel is being insufficiently atomised. And since the ignition point of the fuel is bound to be less than the normal compression temperature, the only cause of the failure to start can be that the fuel is not entering the cylinder early enough. The obvious remedy for this is to advance the cam toe piece operating the fuel injection valve. This would cause injection to begin earlier but have no effect upon the rate at which the fuel charge enters the cylinder.

The fuel must, therefore, be injected earlier and at a greater rate without reducing the degree

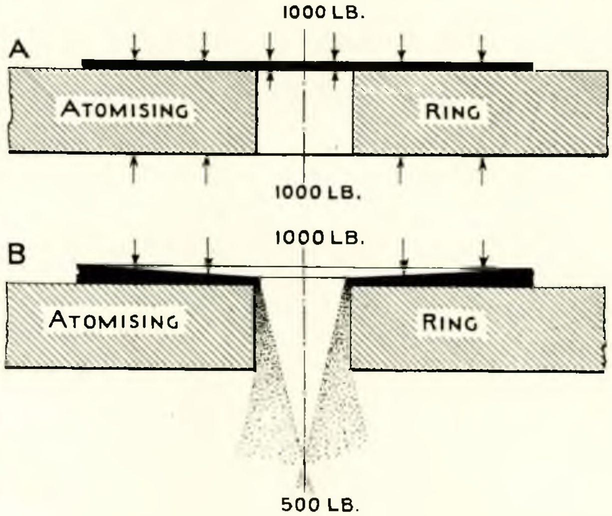


Fig. 1.

of atomisation if the more viscous fuel is to be ignited and the engine started to work efficiently on fuel. This can be achieved to a small extent by increasing the injection-air pressure and/or the lift of the fuel injection valve. This will cause the valve to open and injection of fuel to begin earlier and the rate of entry to be slightly increased. If this adjustment proves insufficient to have the desired effect then the resistance to the flow of fuel through the atomiser must be partly removed. To understand the reason for this one must visualise the charge of fuel lying on the top atomiser ring as at "A" in Fig. 1, where it awaits the opening of the valve in order that it may be blown into the cylinder by the injection air. So long as the valve is closed the only force tending to carry the fuel through the atomiser rings is that of gravity. This force is so small and the resistance of the small passages so great that the entire charge of fuel would still be on the top atomiser ring when the valve opens, even though it was placed there five minutes before instead of one-fifth of a second before, which is about the actual time in a marine engine. Therefore the passage of fuel through the atomiser does not begin until the valve opens, and if some of the obstruction to the flow of fuel is removed the fuel charge will enter the cylinder earlier and at a greater rate. In the most common type of fuel atomiser, this effect can be produced by fitting atomising rings having larger holes or reducing the number of rings. If the holes in alternate rings only are in line, the desired effect may be obtained if the difference in the properties of the fuels is not very great, by bringing the holes in all atomising rings in line.

Because a particular fuel causes some difficulty in starting an engine from cold it does not imply that there will be any trouble due to it when the engine is once started. It is therefore wise to keep a small tank filled with the fuel previously used, which is known to be satisfactory, separate from the fresh supply if the analysis of the latter cannot be obtained or differs very materially from the recommended specifications. The fuel from the small tank can then be used to start the engine, after which the supply can be changed over to the less satisfactory fuel. One of the gravity tanks can be used for this purpose if no other is available. When this procedure is necessary it is advisable before stopping for any length of time to run the engine for the last five minutes or so on the lighter fuel in order to clean the pump valves and fill the system with the lighter oil ready for when the engine is next required. At a time when one had to be far more discriminate in regard to the quality of fuel shipped than is necessary to-day, the Author always procured a 40-gallon oil drum and connected it by means of a charge cock to the fuel pump suction of one of the auxiliary Diesel engines. The practice was to run a portion of the new supply of fuel into this drum and test it in the auxiliary engines before the ship left port. If the auxiliary engines start and work satisfactorily on the fuel, then no anxiety need be felt with regard to the behaviour of the main engines. If, on the other hand, the fuel is not entirely satisfactory for the existing valve timing and atomiser adjustments of the auxiliary engines, experiments can be made and the necessary re-adjustments to the main engine carried out before the new supply of fuel is brought into use.

In changing over from a heavy to a less viscous fuel no difficulty will be experienced in starting an engine, although the cylinder relief valves may lift violently. The cause of the excessive cylinder pressures is that the fuel, being thinner than that previously used for which the injection valve and atomiser adjustments were correct, enters the cylinder too early and at too great a rate. An engine must not be allowed to run under these conditions, as apart from the possible ill effects of abnormally high cylinder pressures, the fuel charge entering the cylinder in a lump, as it were, at the early part of the combustion period instead of being spread over the whole period, will result in imperfect combustion and local overheating of the pistons, which may eventually fracture. To retard the entry of the fuel charge the injection-air pressure and the lift of the injection valve should first be reduced, the former to not less than 100lb. above the compression pressure and the latter not more than one-fifth of the designed lift. If this should not have the desired effect the obstruction to the flow of fuel through the atomiser will require to be reduced.

The injection-air pressure must always be well above the terminal compression pressure in the cylinder, in order to ensure the whole of the fuel charge being injected each cycle. Should the pressure be allowed to fall below that necessary to inject the fuel charge, there is a danger of some of the hot compressed air in the cylinder flowing into the fuel injection valve when it opens and igniting the fuel therein, with very serious consequences. If the fuel injection valve lift is reduced to less than the amount recommended, the whole of the fuel charge may not have time to enter the cylinder, in which case a proportion of it would remain in the valve and, after a few revolutions, so much fuel would have accumulated in the passages between the atomiser and the seat of the valve that "solid" fuel would be injected and excessive cylinder pressures produced. The combustion efficiency under these conditions would be so low that the engine would soon stop with the cylinder relief valves lifting violently. It is a matter for surprise to some that the engine should stop with such unmistakable signs of pressure in the cylinders. The explanation is that an abnormally large quantity of unatomised fuel enters the cylinder immediately the fuel injection valve opens. This more or less "solid" fuel ignites and produces a very sudden rise in pressure, which causes the relief valves to come into action. Although the fuel ignites, it is not in a condition to burn properly, so that the power produced is relatively small and some of this escapes through the relief valves. An indicator diagram taken under these conditions would be very high but of small area.

Reference to Fig. 1 will help to explain how the fuel charge becomes broken up or atomised. The two illustrations, greatly enlarged, are of one of the several holes in the atomising rings of the most common type of fuel injection valve. In the view indicated by "A" (Fig. 1) the injection valve is closed, and since the spaces above and below the atomising ring are under the same pressure, i.e. the injection-air pressure, the fuel charge enters and is evenly distributed on the top of the ring. When the valve opens and is put into communication with the cylinder, the pressure underneath the ring is considerably reduced and the fuel is forced through the holes in the ring as shown at "B" (Fig. 1). The fuel flows towards the holes and when it reaches the edge it is chopped off into small pieces as it were, and the pieces as minute particles mix intimately with the fast moving air and are carried into the cylinder.

These sketches will also serve to illustrate how the rate of entry of the fuel charge into the cylinder is affected by the viscosity of the fuel and the injection-air pressure. It will be clear from sketch "B" that the more viscous the fuel the longer time it will take to flow to the hole, and the lower the injection-air pressure the less will be the tendency for the fuel to be flattened out and be forced towards the hole. The rate of entry can therefore be increased by heating the fuel and thereby reducing the viscosity before it is delivered into the fuel-injection valve, or by increasing the injection-air pressure. Increasing the diameter and/or the number of holes in each atomising ring will, of course, also increase the rate of entry.

When making adjustments with the object of increasing the rate of flow into the cylinder so that starting is facilitated, care must be taken not to alter the degree of atomisation to any material extent, otherwise after-burning will result. Raising the injection-air pressure has the effect of increasing the rate of entry and the degree of atomisation, while increasing the number of holes in each atomiser ring increases the rate of entry only. The effect of increasing the diameter of the holes in the atomiser rings is to increase the rate of entry and decrease the degree of atomisation, while if the number of atomising rings is reduced the rate of entry will be increased without affecting to any material extent the degree of atomisation.

Airless Injection. The chief development of recent years in connection with marine oil engines has undoubtedly been that of airless injection, and the success obtained in practice with this system of fuel injection leaves little doubt that a great many of the engines now employing the older system will be converted. Although the airless-injection system is simpler in construction, it is more sensitive to variations in the characteristics of fuels, and in consequence requires even more thought and attention than engines working on the air-injection system.

In the airless-injection system atomisation of the fuel depends, so far as operating engineers are concerned, mainly upon the size and number of holes in the nozzle and the velocity and direction of the fuel sprays as they enter the cylinder. To ensure proper combustion it is essential that the fuel, in addition to being atomised to the required degree, should penetrate to the furthermost parts of the combustion space and so mix thoroughly with the hot compressed air. The spray penetrability is particularly important, since the degree of turbulence with this system of fuel injection is much less owing to the absence of expanding high velocity injection air which, in the air-injection system, carries the atomised fuel well into the cylinders and ensures thorough mixing of the combustibles.

The fuel sprays entering the cylinder take a conical form, the angle of the cone being mainly governed by the nozzle design and the pressure of injection. As a wide angle of cone will bring the fuel into contact with a greater proportion of the combustion air than will a small angle, it is only to be expected that better distribution will be obtained with a wide angle. The cone, however, must not be too wide, otherwise the fuel particles will strike the piston and cylinder head before they penetrate sufficiently far into the cylinder and poor distribution and, consequently, combustion will result.

The air density in the cylinder also has an effect upon the form of the sprays as well as the degree of atomisation. It has been found that a small change in density has a much greater effect upon the spray than has a relatively large change in injection pressure, so that of the two systems of fuel injection it is more essential to maintain the correct compression pressure in engines working on the airless-injection system. The dissipation of the fuel sprays caused by their friction in travelling through the compressed air has a certain atomising effect, so that the lower the compression pressure in the cylinder the lower will be the degree of atomisation. On the other hand, the greater the density the greater will be the friction between the fuel sprays and the compressed air, and the lower will be the degree of penetration.

A noticeable difference between the air-injection and airless-injection systems is that with the former the desired combustion efficiency can be obtained with a very moderate rise in cylinder pressure upon injection of fuel, whereas to obtain the same good results with airless injection the increase in pressure must be very substantial. The actual figures in lb. per sq. inch are from 500 to 530 in the case of air-injection engines and from 500 to 600 in the case of airless-injection engines. If the pressure rise is reduced 25lb. in each case by delaying the beginning of injection, the effect is hardly noticeable in air-injection engines, but an appreciable increase in the fuel consumption of airless-injection engines will result, indicating of course a corresponding reduction in the combustion efficiency.

Since the combustion efficiency depends upon atomisation and penetration of the fuel charge and turbulence of the combustible mixture, delaying the beginning of injection must affect one or more of these conditions when the airless-injection system is employed. So far as atomisation or penetration are concerned, the conditions affecting same remain unaltered when injection is made to take place a little later, so that such an adjustment must affect the degree of turbulence with airless injection only. It will be quite evident that if the same amount of injection air at the same pressure is admitted just a little later the degree of turbulence in air-injection engines will be unaffected.

It would therefore appear that at the very beginning of the injection period of airless-injection engines, something occurs which creates turbulence. A possible explanation is that the formation of the conical-shaped fuel sprays is such that ignition takes place at the centre of each cone and a slight explosion occurs. The effect of such an initial explosion would be to burst the cones and scatter the fragments about the combustion space as it were, and so cause the atomised fuel and air to mix sufficiently thoroughly to ensure rapid and complete combustion. Evidence of this explosion is the very rapid rise in pressure which results even when injection begins at the same time relative to the position of the crank as in the case of an air-injection engine of corresponding size and speed. Now if the beginning of injection is delayed so that the volume of the combustion space is increasing when the explosion occurs, the pressure rise will not be so great and the force available to violently agitate the cylinder contents and bring about intimate mixing of the atomised fuel and air will be less, with a corresponding reduction in the combustion efficiency.

Late injection will reduce the combustion efficiency in air-injection engines also but, as already stated, to a lesser extent. In such engines the reduction is due not to a lower degree of turbulence, but to a reduction of the time in which the fuel must burn. Airless-injection engines will of course be similarly affected, but the greater reduction in the combustion efficiency with engines of this type for an equal delay in the injection period would certainly appear to be due to reduced turbulence.

In most cases the quality of the fuel and the capabilities of the engine will be such that any slight difference in any of the factors referred to will not have any noticeable effect, but in a great many cases it has been found that an engine is just capable of burning certain grades of fuel without giving any serious trouble until, through normal wear and tear or lack of attention to some of the parts, one or more of the conditions so essential for combustion efficiency are affected and an epidemic of mechanical trouble is the result.

The formation of carbon in the fuel valve nozzles is about the only combustion trouble likely to be experienced with the airless-injection system of fuel. When all is not well, the carbon sometimes begins to form on the inside but generally it forms on the outside of the nozzle. When it forms on the outside, that is the cylinder side, the only effect it is likely to have is to build up and deflect the fuel sprays which may, of course, have very undesirable consequences. When the carbon begins to form in the inside of the nozzle one or more of the holes will very soon become choked; when carbon forms on the outside and some of the holes become choked, the cause of the holes choking will be due to pieces of grit in the fuel.

The formation of carbon on the atomiser nozzle generally results from the temperature of the nozzle or of the fuel being too high and causing ignition and burning of the fuel to take place too close to the nozzle and in an insufficient amount of air. For this reason it is very necessary to maintain a good circulation of water, or whatever the cooling medium may be, in the fuel valve housing, and take other available steps to keep down the temperature of the nozzle. Carbon formation can also result from the employment of inferior grades of fuel, and when this is the cause the deposit is generally of a tarry nature and accumulation takes place on the outside of the nozzle. When carbon formation is the result of an insufficient degree of atomisation, the deposit takes place on the piston top and rarely, if ever, on the nozzle.

When the temperature of the fuel is raised unduly, either as a result of the fuel valve being worked at too high a temperature or to the nozzle being overheated, the effect is to reduce the viscosity of the fuel. Reducing the viscosity in this way will bring about a higher degree of atomisation, and if this condition was correct prior to the temperature of the fuel being raised it means that the fuel will burn too close to and overheat the nozzle. When dealing with combustion troubles it should always be borne in mind that the essential function of the nozzle is not only to atomise the fuel charge but, at the same time, produce sprays with sufficient penetration to ensure the fuel being brought into contact with the air necessary for its proper combustion. As these two requirements are in practice incompatible, all nozzles are a compromise based upon results obtained in practice.

The fact that carbon is formed on the nozzle does not indicate that the fuel is being insufficiently atomised as is generally supposed. On the contrary, the cause is most likely attributable to the degree of atomisation being too high, especially if the exhaust gases are clear and of normal temperature. For instance, if the particles of fuel are made so small that the pressure on the fuel is unable to drive them well into the cylinder, the bulk of the fuel will ignite and partly burn in the vicinity of the nozzle, and since a proportion of the air only makes contact with the fuel, the local temperature produced is much above the normal and the nozzle is overheated. The fuel valve closes very quickly and the cut-off is almost instantaneous, but there can be no doubt that the pressure forcing the last of the fuel charge into the cylinder is greatly reduced, and that it is this part of the fuel charge which carbonises on the overheated nozzle. The reason why the exhaust gases remain clear and the temperature normal is that the highly atomised fuel and air have time to mix and burn completely during the early part of the expansion period, thus no other ill-effect, apart from carbon deposit on the nozzle, results from over atomisation of the fuel.

Although this is the only direct ill-effect from too high a degree of atomisation, it must be remembered that if the carbon is allowed to accumulate to such an extent that the fuel sprays are affected, carbon may eventually deposit on the piston, the exhaust gases be discoloured and increased in temperature, and other symptoms of improper combustion appear. In such an event it is necessary to determine the primary cause of these irregularities before any adjustments to the injection device are made, although as a rule, if carbon deposits on the piston top or the exhaust gases are affected and, in addition, carbon deposits on the nozzle, the trouble is due to over atomisation causing carbon deposit on the nozzle which ultimately affects the fuel sprays and produces imperfect combustion.

If the engine is allowed to run a few days during which it is found that carbon deposits on the nozzle and not on any other part, it proves that too high a degree of atomisation is the primary cause of the carbonisation and that the carbon deposits on the piston are due to the fuel sprays being affected by the accumulation of carbon on the nozzle.

Another cause of carbon depositing on the nozzle is a leaking valve or sluggish closing of the valve due to too fine a fit or the presence of grit, both of which faults result in the formation of drops on the nozzle which quickly carbonise. The carbon in such cases deposits on the outside of the nozzle only.

Until fairly recently it was customary to supply Diesel engine-driven ships with fuel of more or less constant chemical composition and physical properties, with the result that it was seldom necessary for those in charge of the machinery to make any re-adjustments to the parts which affect the combustion efficiency upon changing over to a fresh supply, the engine builders' adjustments being suitable for the fuel obtainable at the numerous bunkering ports throughout the world.

INSTITUTE NOTES.

ELECTION OF MEMBERS.

List of those elected at Council Meeting held on Monday, 3rd February, 1936.

Members.

John Austin, 13, Adelaide Terrace, Waterloo, Liverpool 22.

Edwin Percy Bamford, Superintendent Engineer, Messrs. Anglo-Saxon Petroleum Co., Ltd., St. Helen's Court, Great St. Helen's, E.C.3.

Robert Samuel Brown, 5, Ormerod Road, Stoke Bishop, Bristol.

William McLaren Forwell, 17, Estcourt Terrace, Headingley, Leeds.

John Courtney Vaughan Horton, 43, Billiter Buildings, Billiter Street, E.C.3.

Colin Ferguson Kerr, 22, Woodcroft Avenue, Broomhill, Glasgow, W.1.

Robert Rawlings, 42, Addison Drive, St. Giles, Lincoln.

Arthur Thomas Wall, The Hayes, Cowes, Isle of Wight.

George Wilson, 8, Northwood Road, Prenton, Birkenhead.

Lester Arnold Wright, 128, Colman Road, Norwich, Norfolk.

Associates.

William Gibb, No. 2 Roboni Flats, Bogota Avenue, Cremorne, New South Wales, Australia.

Henry Carmichael McCormick, 44, Aintree Avenue, East Ham, E.6.

Transfer from Associate Member to Member.

Arthur B. Brown, 57, Cromwell Street, Glasgow.

Kaikobad P. Ghandy, Scindia S.N. Co., Sudama House, Ballard Estate, Bombay.

ADDITIONS TO THE LIBRARY.

"Boiler Feed Water Treatment", by F. J. Matthews, B.Sc. Hutchinson's Scientific & Technical Publications, 256pp., illus., 12s. 6d. net.

Scale formation, corrosion and priming are not new problems in the technique of steam boiler operation but, with the development of the modern boiler in the direction of increased working pressure, temperatures and ratings, they have become of much greater importance, demanding improved standards of boiler feed water treatment. Valuable research work has recently been carried out upon these and allied problems, much of which is contained in technical articles and the proceedings of technical societies, and as such inaccessible to many operating engineers.

The Author of the work under review has performed a timely service in collecting and arranging in convenient form the more important of these researches, etc. The book will appeal to the practical engineer who is solely responsible for the feed water conditioning of the boilers under his charge. The opening chapter entitled "Natural Water Supplies" is followed by one dealing with each of the principal operating troubles, viz.: "Scale Formation", "Corrosion" and "Foaming and Priming", and the book concludes with a section entitled "Analysis and Routine Testing".

Of the subjects of particular interest to the marine engineer at the moment may be quoted the following which, among many others, are dealt with very fully in the text:—

Caustic Embrittlement.—The cracking of boiler plates as a result of caustic embrittlement has become one of the most serious problems of modern steam plant operation. In the past numerous conflicting theories in regard to the cause of this have been advanced. As a result of the researches of Parr and Straub at the University of Illinois, however, it is now possible to determine whether the composition of boiler water is such that embrittlement may occur. Together with inhibiting agents this problem is treated exhaustively.

Hydrogen-ion Control.—Hydrogen-ion (pH) control of boiler feed water is a comparatively recent development, but as it simplifies routine testing its value will be appreciated by engineers.

Colloidal Conditioning.—This treatment of boiler water is in use in a number of modern vessels and is dealt with fully in the section on "Scale Formation".

In general, the work is to be thoroughly recommended to engineers in charge of steam plant who wish to be conversant with modern methods of boiler feed water treatment.

"Relativity", by F. W. Lanchester, LL.D., F.R.S. Constable & Co., Ltd., 222pp., illus., 12s. net.

In so far as the modern theory of relativity represents a mathematical instrument for the examination of physical phenomena, a study of the subject is not without interest to engineers, whose work implies a knowledge of the ways of Nature. It has long been recognised that Newtonian mechanics are sufficient for the investigation of laws relating to the behaviour of gross matter, but additional factors must be taken into account in cases where electrons and atoms form the objects of study. This arises from the fact that an electron or ultimate particle of matter is surrounded by a field of force, which affects the motion of adjacent particles, and thus modifies the motion of the complete system of electrons. Consequently, a quantity of matter, such as, for example, a piece of metal along which heat is propagated, may be represented uniquely by a complex system of fields of force. In this manner the relativist substitutes fields of force for the "material" of the universe, and regards that intangible substitute as the "real" world.

The picture thus presented may at first appear to differ greatly from that revealed to us as engineers, but in this connection it should not be forgotten that our experience is usually based on effects, rather than on the cause of those effects. For instance, in the case of the transmission of electricity along a wire, our chief concern may well be that of transmitting from one point to another a given quantity of energy under specified conditions, but the physicist may be more interested in the means or methods by which the current is propagated during the process, an investigation into which involves a study of the transmission of energy from atom to atom of the conductor. On the other hand, the problem may be that associated with the atomic mechanism that lies behind the phenomenon of the strength of materials, which, again, involves the propagation of stress in a material.

An examination of this type of problem in general necessitates the use of the theory of relativity enunciated by Einstein, so that many engineers will welcome Dr. Lanchester's volume. This is so because the Author has brought to bear on the matter a mathematical equipment that has earned for him an international reputation in the sphere of aerodynamics, which has resulted in a noteworthy contribution to the subject. The matter is discussed from first principles, and in a manner that can scarcely fail to appeal to all who possess nothing more than a knowledge of Euclidean geometry. Moreover, the value of the book is enhanced by the fact that the Author has refrained from discussing the related philosophical aspects, in consequence of which the work is a concise statement of an important problem in natural science.

Dr. Lanchester's easy style makes for a book that can be read with pleasure and profit by engineers, and it is manifest that the Author possesses a profound knowledge of his subject, which cannot always be said in reference to scientific literature devoted to recondite matters.

This original treatment of the subject can be recommended without any hesitation, for it belongs to that special class of book the value of which increases with repeated reading.

"Steam Plant Operation", by E. B. Woodruff and H. B. Lammers. McGraw-Hill Publishing Co., Ltd., 368pp., illus., 18s. net.

The little time at the disposal of the student attending university courses for the study and inspection of actual working conditions of necessity affords only the briefest glimpse of the infinite number of details and difficulties that confront the practical man. This book takes the reader rapidly over the whole gamut of steam plant management and gives a practical insight into the working of almost everything that has to do with steam boilers and steam engines. It gives "thumbnail" sketches of the equipment that may be found in any industrial plant. Boilers, combustion of fuels, boiler accessories, valve settings, mechanism and governors, operating characteristics of steam engines, turbines, pumps and auxiliary power equipment are all covered.

The Authors, who are both practical engineers and also instructors at the Ohio Mechanics Institute, have presented a combination of practical and technical information for the help of those who intend to take the Engineer's license in the United States, and have produced a useful book for home study for those attending technical school classes. At a time when specialisation has become a matter of considerable importance it is as well not to forget to pay attention to many other details which may be a little more "round the corner" than the daily duties.

In their endeavour to miss nothing that may be of interest to steam engineers the Authors have been somewhat over-ambitious, but at the same time the variety of questions given at the end of each chapter will tend to assist the reader in mastering the subject matter. A number of simple formulæ are included, but the reader outside America must not forget that the American ton and gallon are used.

"Examples in Heat and Heat Engines", by T. Peel, M.A. Cambridge University Press, 146pp., 5s. net.

This book is the second edition of a collection of questions which, with few exceptions, have been taken from papers set for the courses of lectures at the Cambridge Engineering Laboratory and from the A and B papers of the Engineering Tripos.

The contents are arranged under very suitable headings beginning with heat and work and laws of gases, followed by the various theoretical cycles, entropy, indicator diagrams, steam engine trials, combustion and explosion of gases, boiler trials, compressors and refrigerators, steam nozzles, steam turbines and elementary conduction. The book terminates with 40 pages of general examples and 15 pages containing the answers to the whole of the questions.

The examples on heat and work and laws of gases are varied and generally more interesting than the examples found in the average textbook, since they are not confined to engine application. The indicator diagram section containing five questions on steam engines and one only on internal-combustion engines is disappointing, as much useful information can be obtained from internal combustion engine diagrams. The papers containing air, gas and oil engine trials are a good representative selection, although Question 6, page 57, and Question 5, page 60, are practically identical. There are eight good questions on steam nozzles and another eight on turbines, and these are augmented by further examples in the general section.

Although the Author has selected most of the questions from lectures given at Cambridge, there is a general atmosphere of direct practical application throughout the book. The absence of any particular reference at the end of each question as to the exact examination paper from which it has been taken, leaves the origin of the questions rather vague.

The book is a useful one to have for reference when studying for examinations in applied thermodynamics, as a considerable amount of useful practice can be obtained in working out questions of a varied character, and it can be thoroughly recommended.

"The Motor Ship Reference Book, 1936". Temple Press, Ltd,. 255pp., illus., 5s. net.

The present edition of this book is the 12th annual volume to be published. The usual thorough annual revision has been carried out, and all statistics and other details have been corrected up to 1st January, 1936. A modification has been made in the arrangement of the ships completed, so that all vessels, excluding those constructed in 1935, are now arranged alphabetically, whilst those built in 1935 are also set out alphabetically in a separate table. Thus, every motor ship of over 2,000 tons deadweight capacity is listed with all the main particulars. There has also been a very complete revision of the chapters on Diesel propelling engines and auxiliary Diesel machinery, so that the most recent types are included. The following list of contents of this valuable work of reference indicates its wide scope, viz.: history and progress of the motor ship; motor shipbuilding in 1935; principles of operation of oil engines; four-cycle single-acting engines: two-cycle single-acting engines: double-acting Diesel engines; Lloyd's Rules for the construction of motor ship machinery; motor ships of the world; list of motor ships completed in 1935; builders of marine Diesel engines; general motor shipbuilding statistics, engine weights, fuel consumptions and cargo capacities; auxiliary engines; and fuel and lubricating oil for motor ships.

There seems to be no other work on this subject which affords such comprehensive reference facilities, and it is a book which the marine engineer can ill afford to omit from his library.

"Standard Methods for Testing Petroleum and Its Products". 3rd edition. Institution of Petroleum Technologists, 228pp., illus., 7s. 6d. net.

Marine engineers receive their supplies of fuel oils according to a specification and their responsibility is more one of quantity than quality. Samples are taken and retained—with perhaps an odd check test on specific gravity and other similar simple test—until the bunkers are clear of that supply.

For many years now a committee has been engaged on investigating the methods of testing, their standardisation, and also the revision of previous standards. Such revision has been mainly with a view to securing international standardisation, a valuable objective particularly for the marine engineer who at times has to face disputes in foreign ports.

The considerable progress which has been achieved in the formulation and establishment of standard methods is clearly indicated by the size of the new edition of this work, which contains nearly 100 pages more than the second edition.

The book is the authority on the methods of testing various chemical and physical properties of petroleum products and as such it represents a valuable reference book.

"Practical Boiler Firing", by H. C. Armstrong and C. V. Lewis. Charles Griffin & Co., Ltd., 132pp., illus., 4s. net.

This is a well written and instructive book giving in practical form the principles of scientific and economic control of coal-fired boilers which can be put into everyday operation.

Scientific investigation is of little use if the results are not put into practice, and this volume contains many useful hints to the fireman, the engineer, and to others who may be responsible for the upkeep of boiler plant. It indicates how the thoughtful study of firing, use of the most suitable coal, and correct air supply and distribution will result in fuel economy, less expenditure on cleaning and maintenance, and added life to the boiler plant, at the same time making the fireman's duties less arduous and more interesting.

Both hand and mechanical firing are dealt with. The Authors have devoted a short chapter to the upkeep of steam mains—an important component of the boiler plant that is so often neglected—and there are other chapters on combustion, fuel, stoking, boilers, boiler mountings, measuring instruments, feed water treatment and boiler efficiency. Mathematical calculations have been avoided, except for a few examples of a very simple nature in the chapter on boiler efficiency. The book is a useful contribution on boiler firing and will appeal to the practical man.

"The Preparation of Engineering Reports", by Thomas R. Agg, C.E. and Walter L. Foster, C.E. McGraw Hill Publishing Co., Ltd., 187pp., illus., 10s. 6d. net.

This book has been written primarily as a basis for a brief course dealing with the content and form of engineering reports, by two members of the Iowa State College, Thomas Agg, C.E., Dean of Engineering and Walter Foster, C.E., late Professor of Engineering. The Authors have been at great pains to produce a book which is equally useful as an occasional work of reference and as a guide in the actual writing of engineering reports. The subject is introduced broadly at the commencement and the study differentiates between the informal and the formal report.

Throughout, innumerable illustrations of actual existing reports are supplied, being entirely quotations from American civil and mechanical engineering proceedings. A chapter is devoted to the problems involved in collecting data, including suggestions as to the available sources of report data, and one on the general style, arrangement and choice of suitable type for the clear presentation of ideas.

An excellent chapter follows on the uses of diagrams, charts and graphs. This chapter is profusely illustrated and shows the many possible ways of presenting data in simple design which can be readily appreciated by the reader of the report.

Appendix A on suggestions on style is essentially a leaf taken from any book on English composition and includes punctuation, use of capitals, use of italics and numerals as particularly applied to engineering terms. Appendix B gives a very comprehensive list of accepted abbreviations of relevant technical terms. Appendix C has been written for the use of the teacher of the subject and suggests a course of study to be followed from the book in an outline of 30 contacts, and finally Appendix D provides a list of suitable books and reports for reference in connection with Appendix C.

The art of clear, concise expression, so important in presenting the solutions of engineering problems on paper requires careful study and here is a book filled with helpful suggestions to assist those who find it necessary to resort frequently to the written report.

"Unsolved Problems of Science", by A. W. Haslett. G. Bell & Sons, Ltd., 317pp., 7s. 6d. net.

The achievements of science as manifested in terms of industry seem sometimes to border on the miraculous. Perhaps it is to this cause that the irritating superstition in the minds of a section of the lay public that the scientist is to be regarded as almost omniscient is to be attributed, and why when eminent scientists indulge in speculations outside the field in which they can be regarded as authoritative, their views are accorded a respect which is quite unwarranted. Mr. Haslett's review of some of the more important of the many unsolved problems of science, if it is read as widely as it deserves, will have a very salutary effect.

The problem of the creation of the universe first engages the Author's attention, and he gives a brilliant exposition of the current theories of the "Concertina Universe", the "Explosive Creation Universe" and what is termed "Creation by Toppling". He then presents the evidence for and against the possibility of there being inhabited worlds other than our own, followed by a discussion on the changing face of the world. In connection with the latter, an interesting theory of "continental drift" put forward by Professor Wegener is clearly explained.

Meteorology and cosmic rays are next discussed, and the Author expounds in simple language the difficult theories associated with the latter phenomena. The scientific evidence regarding the origin of man and the allied mystery the "beginnings of civilisation" is then reviewed, and is followed by chapters on the mechanism of the human body and the future possibility of predetermining sex. Special mention must be made of the Author's successful attempt to describe, in homely analogy, present knowledge of atomic physics and the impasse arising out of the implications of the quantum theory. Perhaps the chapter of most direct interest to engineers is that dealing with what the Author terms "the secret of strength". The use of X-ray analysis, as well as fatigue, "crystal slip" and atomic structure are briefly discussed.

Written with a pleasing smoothness of style, this able review of the researches, significant discoveries and problems of the scientific world of to-day can be read with little more concentration than is required for an average novel. It cannot fail to interest the engineer.

"Electric Arc Welding Practice", by H. I. Lewenz. 128pp., 108 illustrations and figures. Crosby Lockwood & Son, Ltd., 8s. 6d. net.

The primary object of this volume is to be of assistance to those who actually have to carry out the work of arc welding, and whilst no attempt has been made to treat the subject exhaustively, the Author endeavours to show, by practical examples, some of the difficulties that the welder has to surmount.

The introductory chapter refers briefly to the various welding processes from forging and burning to carbon arc and metallic arc welding. In a chapter on the characteristics and weldability of metals, the absence of essential information, not given in subsequent chapters, is disappointing; for example, the Author's statement that "the welding of high carbon steels calls for a very high degree of skill on the part of the operator and the most careful control of all conditions" clearly calls for amplification.

A fair proportion of space is devoted to types of electrodes and it is interesting to note that on the grounds of instability of the arc the use of a.c. is not recommended, and again in a later chapter on equipment, the same point is stressed, in this case from the aspect of welding efficiency and quality, and safety, although no facts to explain the advantages of d.c. over a.c. are given.

In the chapter on stresses and deformation, an interesting and commendable feature is the use of red lines to amplify the diagrams. The inclusion of stress diagrams without a full explanation of their purpose is of little value to the average welder for whom this book is written, and he would probably prefer fewer illustrations relating to the "Application of Welding" and more detailed information on such matters as the influence of hammering on welds, the question of polarity, and advice on the more fundamental problems that beset him.

The book should prove useful as an introduction to the subject; it is clearly written, with the various sections brought out in the margin, a useful idea for this type of book.

JUNIOR SECTION.

Steam versus Electrical Auxiliaries.

A debate on the above subject was held in the Lecture Hall of The Institute on Thursday evening, 16th January. The protagonist of steam auxiliaries was Mr. E. H. Gibson (Associate Member) and he was opposed by Mr. D. M. Cathcart, B.Sc., who took the electrical side. Mr. H. R. Tyrrell, B.Sc. (Associate Member), occupied the Chair.

Mr. Gibson in a well-reasoned summary of the advantages of the steam auxiliary stressed its simplicity and economy in use, and pointed out the losses incurred in the electrical system, which could not compare with steam on the score of efficiency. He maintained that the evidence which he put forward undoubtedly indicated that electrical auxiliaries tended to increase weight, fuel consumption and the power required, while it was well known that the first cost of this type of plant was greater.

In a vigorous statement of the case for electrical plant, Mr. Cathcart showed that this type could now be used for every auxiliary duty on board ship. Horse power for horse power, the electrical auxiliary was lighter, and on the score of reliability and ease of dismantling for repair it excelled its steam rival. After a review of the uses and advantages of the electrical auxiliary, which he declared was simple and robust, Mr. Cathcart presented evidence to show that electrical gear would give a saving in fuel costs, salaries, upkeep costs, insurance and loss due to damage.

Following a very keen discussion, in which many of the senior members present joined, the two principals replied to the various points raised. On a vote being taken, the result was an almost two-thirds majority in favour of electrical auxiliaries.

On the proposal of Mr. E. F. Spanner (Member) a very cordial vote of thanks was accorded to Mr. Gibson and Mr. Cathcart for their able advocacy of the two types of machinery.

ABSTRACTS.

The Council are indebted to the respective Journals for permission to reprint the following abstracts and for the loan of the various blocks.

The Propelling Machinery of the "Tannenberg".

A Double-reduction Geared-turbine Propelling Installation at 880lb. per sq. in. Working Pressure.

Abstract from an Article by Regierungsbaurat von Rohr. "The Shipbuilder and Marine Engine-Builder", January, 1936.

Early in September, 1935, the twin-screw passenger steamship "Tannenberg", built by Stettiner Oderwerke A.G., of Stettin, to the order of Reichsverkehrs-ministerium for service along the south coast of the Baltic, was handed over after satisfactory trials.

In Fig. 1 an interesting comparison of the space and weight requirements of three modern engine-room layouts is given. In spite of the fact that the essential main-engine auxiliaries of the Diesel installation have been taken at their least possible values, the combined weight of the turbine propelling machinery, auxiliaries and steam-generating plant installed on board the "Tannenberg" is less than the total weight of the Diesel plant for developing the same power. Furthermore, the engine-room floor area necessary for this new steam installation is considerably less than that required in the case of either Diesel or turbo-electric propulsion. The low space and weight requirements were some of the principal deciding factors in the selection of the type of machinery for the "Tannenberg". A further important consideration was the two diverse speeds at which the vessel has to operate. The service speed is about 16·5 knots, which, for a vessel of about 4,500 tons gross, requires about 6,000 s.h.p. The ship, however, has to be capable of continuous propulsion at 20 knots, which means that the main turbines must be able to develop continuously more than double the power named. This large reserve of speed enables the ship to run to schedule despite possible delays caused by bad weather, and thus ensures that passengers will make their connections at the terminal ports.

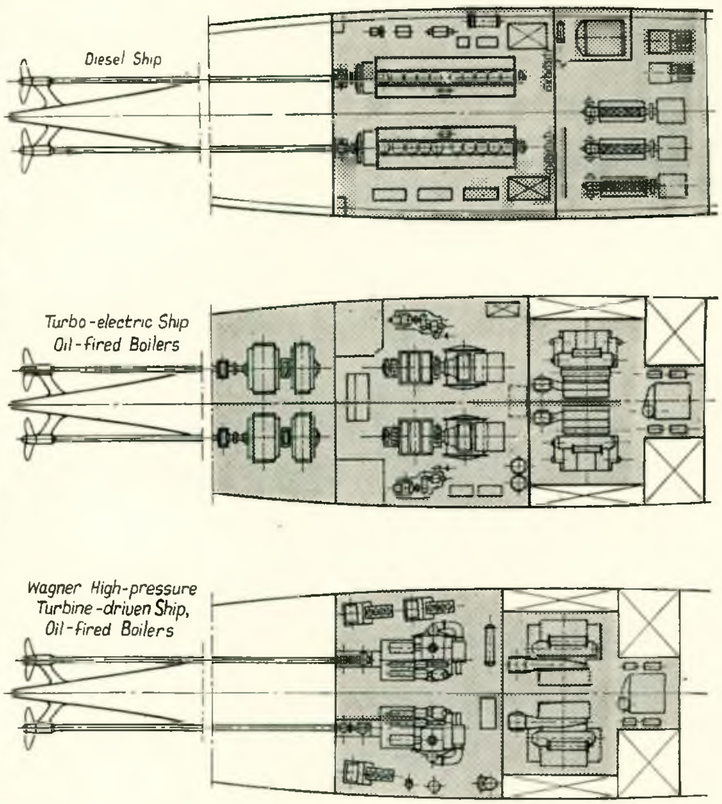
The general arrangement of the machinery spaces will be apparent from the plans reproduced in Fig. 2. The main propelling machinery consists of two sets of double-reduction geared turbines, supplied with high-pressure steam by two Wagner-type water-tube boilers.

Boilers.

The two steam generators built by Firma F. Schichau, are of the Wagner water-tube type. The final steam conditions are 880lb. per sq. in pressure and 860° F. total temperature. The boiler scantlings are sufficient to enable pressure of 1,030lb. per sq in. to be worked. Each boiler is capable of supplying all the steam required by both main turbines in order to maintain the service speed. Under these steaming conditions, one boiler is, of course, shut down, and this makes for very economical running. Sectional drawings of the boiler are given in Fig. 3. The drums are seamless and forged from one piece out of Molybdenum steel, the following being the properties of this material at 275° C. (530° F.):—

Ultimate tensile strength	28·5 tons per sq. in.
Yield stress	15·3 tons per sq. in.
Elongation (length of test piece = 10 × diameter) ...	18 per cent.

For both the downcomers and upcomers, cold-



	Diesel Engine (Sulzer).	Turbo-electric Propulsion.	Wagner High-pressure Geared Turbines.
S.H.P. for service speed...	5,850	6,400	5,800
S.H.P. for maximum speed	12,500	13,400	12,500
Engine-room floor area (approximate)	380 sq. m. (4,100 sq. ft.)	445 sq. m. (4,800 sq. ft.)	300 sq. m. (3,250 sq. ft.)
Weight of main engines only, English tons ...	817	550	342
Weight of main-engine auxiliaries (including boilers in Columns 2 and 3), English tons ...	136	246	206
Total weight of propelling machinery, English tons	953	796	548

Fig. 1.—Comparison of engine-room layouts.

drawn seamless tubes of Molybdenum steel have been adopted having the following mechanical properties at 275° C. (530° F.):—

Ultimate tensile strength	24·1 tons per sq. in.
Yield stress	12·7 tons per sq. in.
Elongation (on specimen of length = 10 × diameter) ...	20 per cent.

The superheater elements are arranged in two groups, the tubes in each being so expanded into the headers that complete and ready drainage is possible. The air preheater is arranged in the uptake, and the elements are of stream-lined section. In this way, the resistance to the flow of funnel gases is reduced to a minimum. Soot-blowers for the air-heaters are fitted, while the soot-blowers for the main-boiler tubes are arranged between the first and second rows of tubes (downcomers) counted from the inside of the combustion space.

The following are the principal heating surfaces of each boiler:—

Total heating surface of tubes	3,230 sq. ft.
Superheating surface	1,240 sq. ft.
Air-heater surface	6,460 sq. ft.

The tube area exposed to the flame is 254 sq. ft., or some 8·2 per cent. of the total tube-heating surface. This relatively large proportion of the heating surface absorbing radiant heat makes for high rates of evaporation and lower furnace temperatures.

Each boiler is equipped with two Saacke-type burners, one at either end of the combustion space. The capacity per burner is about 2,000lb. per hour. The volume of the combustion chamber is 670 cu. ft., and the heat release is 100,000 B.Th.U. per cu. ft. per hour.

The fuel-oil units are electrically driven, as are also the two primary-air forced-draught fans. The secondary air is delivered by two fans, driven through reduction gear by two small fast-running steam turbines. These units are arranged at the after end of the boiler-room on a flat some distance above floor-level.

The overall steam consumption at the service speed is 22·1 tons per hour and the rate of evaporation is 15·3lb. per sq. ft. of heating surface, while at a speed of 20 knots the figures are 45·2 tons per hour and 15·7lb. per sq. ft. of heating surface.

Main-Turbine Units.

Each set comprises one high-pressure, one intermediate-pressure and one low-pressure turbine, together with a separate high-pressure astern turbine, and a low-pressure astern turbine incorporated in the low-pressure ahead casing. These turbines are of the multi-stage impulse type. A longitudinal section of the high-pressure unit is given in Fig. 4.

At the highest ship speed of 20 knots, the propeller speed is 250 r.p.m., and the revolutions per minute of the various turbines are as follow:—

High-pressure	18,000
Intermediate-pressure	15,700
Low-pressure	6,500

These unusually high speeds were selected for two principal reasons—(1) to keep the steam consumption as low as possible by reduction of clearance losses, and (2) to keep the sizes of the wheels as small as possible. With regard to the second reason, experience has proved that small fast-running turbine wheels are more reliable and safer than larger slow-running wheels. This is a particularly important consideration in the installation under discussion, as the heat drop in the high-pressure turbine is large. Further, the high speeds of the high-pressure and intermediate-pressure units have resulted in these turbines being small. The low-pressure ahead turbine is a combination of the single-flow and double-flow types; and while the first four stages are of the single-flow type, the last two stages are double-flow.

The high-pressure astern turbine, as mentioned previously, is housed in a separate casing, and is coupled to an extension of the intermediate-pressure turbine shaft. The high-pressure astern turbine is of the overhung type. The total astern power developed is some 60 per cent. of the service ahead power.

The control of each main-turbine set is carried out by means of a handwheel, which operates three cam-actuated valves. One-valve is open for a speed of 16·5 knots and two valves for 20-knot propulsion, while the third valve controls the steam to the astern turbines.

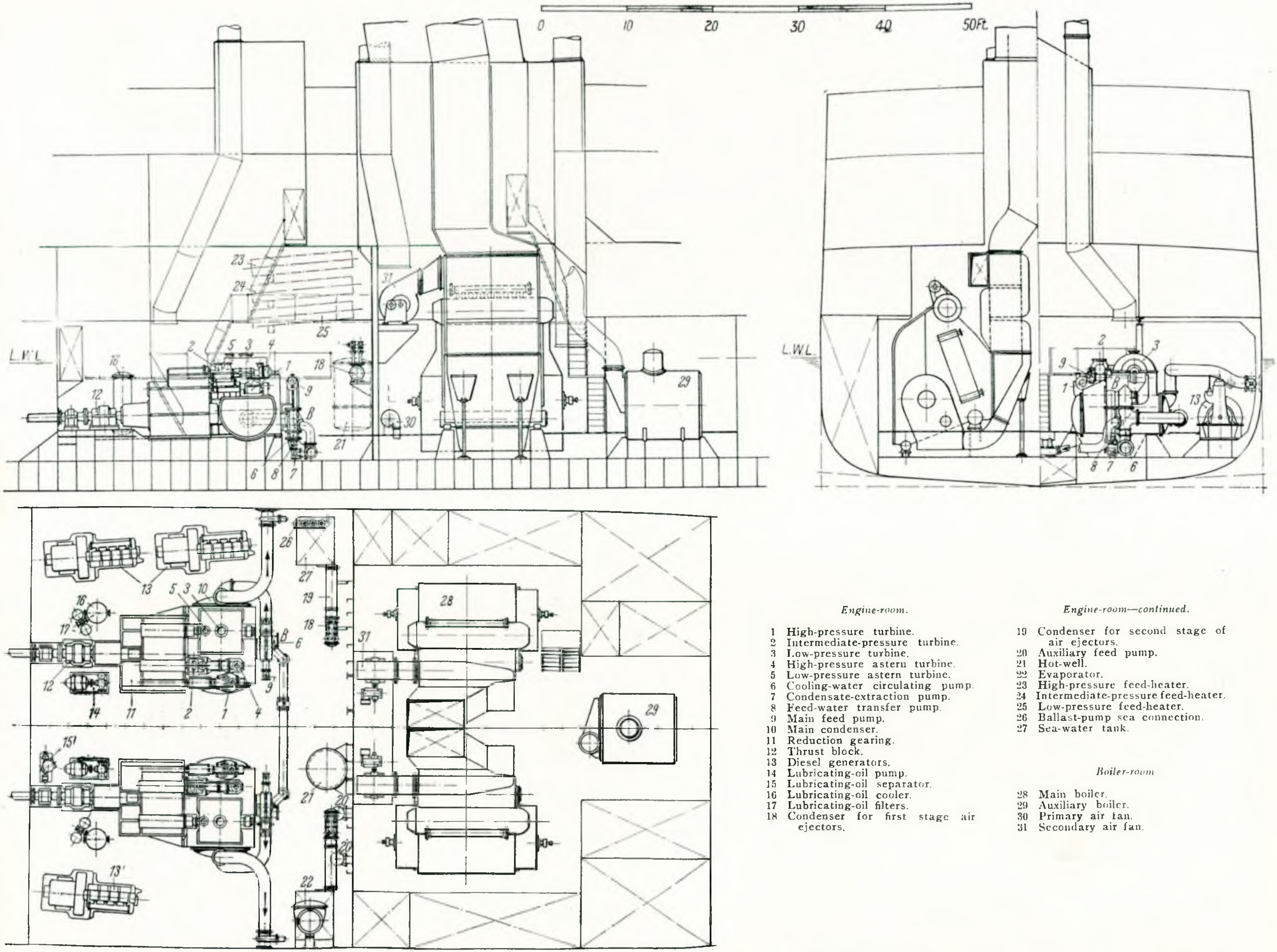
The high-pressure ahead, high-pressure astern and intermediate-pressure turbine casings are of Molybdenum cast steel. Cast steel has also been adopted for the lower halves of the ahead and astern exhaust casings of the low-pressure turbines.

The main condenser is integral with the low-pressure turbine casings.

Power is transmitted to the propeller shaft from the double-reduction gearing through a gear-wheel-type flexible coupling, fitted just aft of the gearcase. Abaft this coupling is the main thrust-block. The object of the flexible coupling is to prevent shocks from the propeller reaching the main gears, with a consequent reduction of wear and tear of the wheels, and, further, it is a definite aid to noiseless running.

Nickel-chrome steel has been adopted for the pinions, while the rims of the gearwheels are of manganese steel. The gearcase of the low-pressure turbine cover and condenser shell are of the fabricated welded steel-plate type.

Rigidly attached to the main propelling set at the forward side of the condenser is a pump group. A multi-stage impulse turbine is direct-coupled to a two-stage main-boiler feed pump, which runs at 15,500 r.p.m. Mounted on the shaft between the turbine and pump is a pinion meshing with a gear-wheel, which in turn drives two vertical shafts through the intermediary of bevel gears. One of these vertical shafts drives a propeller-type cooling-water circulation pump at 1,375 r.p.m.; while the other drives at 2,500 r.p.m. a single-stage condensate-extraction pump, and a two-stage feed-water transfer pump which delivers the feed to the main



Engine-room.

1 High-pressure turbine.
2 Intermediate-pressure turbine.
3 Low-pressure turbine.
4 High-pressure astern turbine.
5 Low-pressure astern turbine.
6 Cooling-water circulating pump.
7 Condensate-extraction pump.
8 Feed-water transfer pump.
9 Main feed pump.
10 Main condenser.
11 Reduction gearing.
12 Thrust block.
13 Diesel generators.
14 Lubricating-oil pump.
15 Lubricating-oil separator.
16 Lubricating-oil cooler.
17 Lubricating-oil filters.
18 Condenser for first stage air ejectors.

Engine-room—continued.

19 Condenser for second stage of air ejectors.
20 Auxiliary feed pump.
21 Hot-well.
22 Evaporator.
23 High-pressure feed-heater.
24 Intermediate-pressure feed-heater.
25 Low-pressure feed-heater.
26 Ballast-pump sea connection.
27 Sea-water tank.

Boiler-room

28 Main boiler.
29 Auxiliary boiler.
30 Primary air fan.
31 Secondary air fan.

FIG. 2.—General arrangement of the machinery spaces in the "Tannenberg".

feed pump. For stand-by duty, two turbo-feed pumps of the directly-driven vertical-shaft type are provided, and each of the feed pumps is fed by a transfer pump operated from the vertical spindle through gearing.

Boiler-feed Circuit.

The condensate from the feed-heaters drains to the hot-well, and the turbine condensate passes through a mixing nozzle into the hot-well. From the latter, the feed-water transfer pumps draw the condensate and deliver it to the high-speed main feed pumps. Regulation of the feed circuit is effected in the hot-well. The head of water in this tank controls an overflow valve, which by-passes the surplus feed to a storage tank; or, alternatively, if the head is low, an automatic valve allows a suitable quantity of feed to enter the circuit.

The boiler feed is pumped through the three-stage feed-heaters. The low-pressure heater is fed with exhaust steam from the auxiliary machinery, and the intermediate-pressure and high-pressure heaters are fed with steam bled from the intermediate-pressure and high-pressure turbines respectively. The control of the feed-heating apparatus is so interconnected with the main-turbine regulation that the intermediate-pressure and high-pressure turbine stages bled are higher for travel at 16·5 knots than at 20 knots, and consequently the best temperature of feed-heating is always obtained.

Automatic control has been adopted for the main feed pumps. The regulator is of the differential-pressure type, and controls the turbine driving the pump in such a manner as to always maintain a fixed difference between the pressure in the pump-delivery line just before the feed-check valve and the pressure inside the boiler.

At the service speed only one pump set is in operation, and this is sufficient to circulate both condensers, to feed the one boiler in operation, and to deal with the condensate from both condensers. To allow for "working" of the hull, all large pipe connections to the pump group are provided with flexible couplings; and for the large diameter cooling-water circulating pipes, rubber connections are utilised.

The lubricating-oil service for the main turbines and gearing incorporates three lubricating-oil pumps of the electrically-driven spur-wheel type, one of which acts as stand-by. Two lubricating-oil coolers are provided, each of 430 sq. ft. cooling surface. The cooling medium is sea-water tapped from the condenser-circulating service.

The make-up feed is supplied by a surface-type evaporator, fitted with a de-aerator and having a capacity of about 20 tons per day. Special care has been taken to eliminate the oxygen from the feed water, and, further, the feed is kept slightly alkaline, which tends to keep the boiler heating surfaces free from scale and to prevent corrosion.

Electrical Equipment.

The only engine-room auxiliaries on board which are electrically driven are the lubricating-oil pump, primary-air fans and oil-fuel sets; and hence the electrical load is small.

Three M.A.N. Diesel generators of 110 kW. output each are provided, mainly for ship's use. For cooking and heating purposes, steam is used, and this is generated in a small oil-fired cylindrical boiler having a heating surface of 560 sq. ft. The working pressure is 60lb. per sq. in. The secondary-air fan turbines are specially arranged with suitable nozzles to take this low-pressure steam when the main boilers are getting up steam, and with this arrangement it is possible to raise steam in the high-pressure boilers in 25 minutes.

. . . . . .

It is interesting to note that the main turbines do not need to be warmed through when starting up. This is due to the small size of the turbine wheels; and, further, as the live steam can be put immediately through the turbine, all possibility of rotor distortion through uneven warming up is obviated.

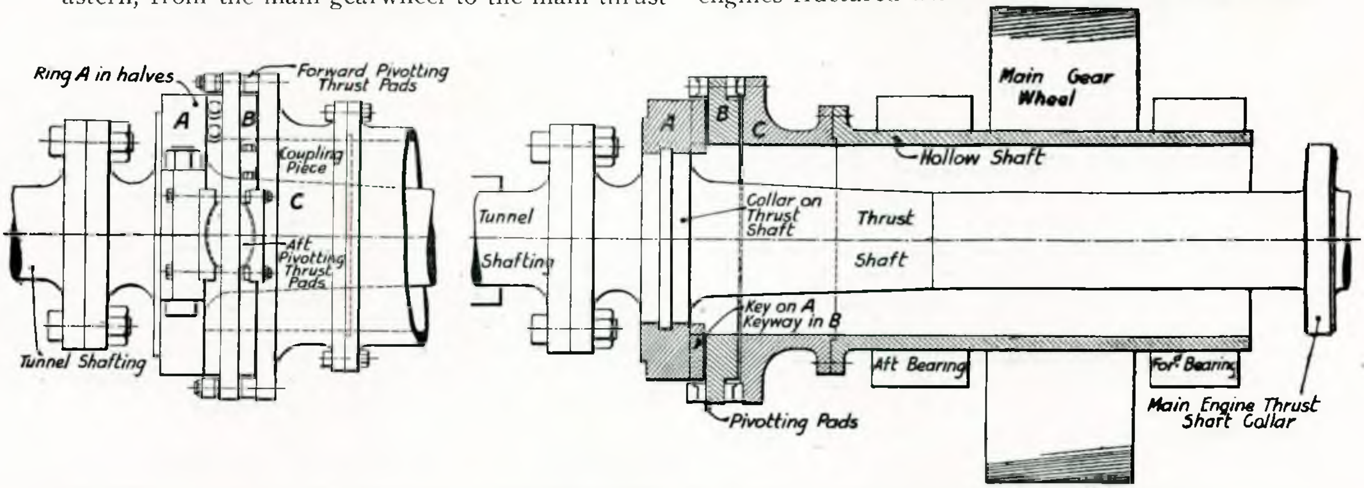
The "Tannenberg's" machinery is the first high-pressure turbine installation of its type to be used in the mercantile marine, and the high overall efficiency attained is due to careful consideration of the individual efficiencies of the separate units.

New Coupling for Bauer-Wach Installations.

"The Marine Engineer", October, 1935.

This coupling is primarily intended for steamships fitted with an exhaust turbine connected through gearing to the reciprocating main engines. In the usual arrangement with the Bauer-Wach exhaust steam turbine, it will be recalled, the main exhaust turbine speed reduction gearwheel is mounted on a hollow shaft supported on bearings. The main thrust shaft for the reciprocating engine is carried through the hollow shaft and at the after end is connected rigidly to the hollow shaft. The main line of shafting is carried on bearings independent of those supporting the hollow shaft. A small difference in the wear-down of the two sets of bearings is accommodated by a slight flexing of the thrust shaft. The new coupling is designed to allow for considerable difference in the alignment of the main shafting, compared with the hollow shaft without imposing any stresses on either, all movement, angular, horizontal, or vertical, being taken up by the arrangement of keys and pads in the coupling; the development should thus make for improved reliability over a period of years.

The coupling consists of three rings marked *A*, *B* and *C* on the accompanying sketch. Ring *C* is rigidly attached to the after-end of the hollow shaft, and has two main driving keys. Ring *A* is rigidly attached to the thrust shaft and also has two main keys similar to those on ring *C*. Ring *B* is fitted between members *A* and *C* and has four keyways to take the main driving keys on *A* and *C*, the keyways facing ring *A* being at right angles to those facing ring *C*. With this arrangement the power can be transmitted from the hollow shaft to the main line even if the shafts are not concentric. In addition, provision is made for angular misalignment of the shafts. The face of ring *B* is not actually in contact with *A* or *C*, but there is a clearance to allow for angular displacement. Pivoting pads or rocking centres are provided at right-angles to one another so that the whole coupling can adjust itself during rotation. Thrust surfaces are combined with the rocking pads to enable the coupling to transmit thrust either ahead or astern, from the main gearwheel to the main thrust



Sketches showing construction of the special Bauer-Wach combination machinery coupling evolved by Clan Engineering Patents, Ltd.

block. The coupling as a whole runs in oil, and provision is made to apply oil jets to the key faces and thrust pads. In the event of a mishap to the exhaust turbine, ring *A* can be removed from the thrust shaft, and the reciprocating engine run quite independently of the turbine when the latter is out of action.

Couplings of this character have recently been fitted in new Clan Line Bauer-Wach installations, constructed by Swan Hunter & Wigham Richardson, Ltd., Neptune Works, Newcastle-on-Tyne; the reciprocating engines were built by the North Eastern Marine Engineering Co., Ltd., Wallsend-on-Tyne.

Torsional Oscillation.

"The Engineer", 20th December, 1935.

Of the practical consequences of torsional oscillation in the torque-transmitting members of engines and other machinery there have been numerous demonstrations. One of the most striking instances on record is that provided by the German airship "Graf Zeppelin" on what was intended to be her second voyage across the Atlantic in May, 1929. Following her first successful voyage, her five Maybach engines had been fitted with superchargers and brief flights with the engines so modified had been carried out. There can be no doubt that the addition of the superchargers converted the dynamical conditions of the engines from safe to dangerous and that the added masses altered the natural period of torsional oscillation of the crankshafts to a value closely in tune with the running speed of the engines. Resonance being present, failure by fatigue became a question only of time. Within twelve hours of leaving Friedrichshafen, while the airship was over Spain, the crankshafts of two of her engines broke in succession. It was decided to return to Germany but over the Rhône Valley the crankshafts of two more of her engines fractured within five minutes of each other. After she had, with great skill, been landed near Toulon, it was found that the crankshaft of the fifth engine was just on the point of fracturing. There have been corresponding occurrences elsewhere, notably in some motor ships, of which no doubt our readers have heard. Even the recognition of possible danger and the fitting of vibration dampers as a precautionary measure, has, in certain instances, failed to avert fracture by fatigue induced by torsional oscillation. Within the past few years the subject has been extensively studied both mathematically and experimentally, but it still remains a source of anxiety to designers. It is scarcely possible that any engineer to-day would, as in the past, blindly incur the conditions requisite for disastrous torsional oscillation. But between the recognition of the danger and the taking of adequate measures to avoid it there is still a considerable gap across which existing scientific knowledge of the subject is not fully sufficient to carry us.

Any resilient system subjected to an external cyclically varying load must oscillate in some degree. In an internal combustion engine torsional oscillation of the crankshaft must always be present under the rhythmically varying torque applied to it by the piston forces. Torsional oscillation, if the torque is not absolutely constant, as it never is, is not some accidental defect which is capable of being suppressed but is in all cases a necessary accompaniment of the transmission of power. In many cases the amplitude of the oscillation is very small and as a consequence the stress in the part concerned fluctuates only through a small range about the mean value corresponding to the power being transmitted. In such cases the maximum value of the stress is below the fatigue limit of the metal and safety ensues. In some cases the amplitude of the oscillation and the range of stress variation may be satisfactorily small at all speeds except one, or one of a series of particular speeds. The shaft is capable of oscillating torsionally, like a torsional pendulum, in a certain natural period. It may, depending upon its construction, be capable, like certain other bodies such as a circular disc held round its periphery, of oscillating in several different natural periods. If the frequency of the cyclically varying load applied to it is in tune with one or other of its natural frequencies the amplitude of the torsional oscillation will show a marked increase over what it is when the frequency of the varying load is something above or something below the "in-tune" value. The factors which determine the extent of the amplitude —and therefore of the stress range induced by the oscillation—are, when resonance occurs, the magnitude of the load, the stiffness of the shaft and the magnitude of the damping influences acting to restrain the oscillation. The effect of the damping influences is critical. If the damping influences are powerful the amplitude of the oscillation at resonance will be relatively small and the actual point of resonance will not be sharply defined. If the damping is slight the amplitude will be relatively great and the speed at which resonance occurs will be sharply defined. So much elementary theory teaches us. The need of the practical designer is clearly some means of calculating the amplitude at resonance in order that he may determine therefrom the corresponding stress range with the object of ensuring that such range is within the safe limit with respect to fatigue. To calculate the amplitude a knowledge of the amount of damping is essential. The position at present is that not only do we not know how much damping to count upon in any given case; we do not even know with certainty what characteristics of the engine or system give rise to the damping, or, to be more precise, how much various possible sources contribute to its total. The damping influences, whatever they are, have one and only one function to perform, namely to rob the oscillation of energy and thereby limit its amount. On the Continent a commonly held view is that the main element contributing to the damping is the variation which change of speed produces in the resistance of the driven unit—say the generator or propeller or pump. In the United States Professor F. M. Lewis some ten years ago popularised the belief that the chief source of damping lay in the hysteresis of the metal composing the part in oscillation. In this country Dr. J. F. Shannon in his paper "Damping Influences in Torsional Oscillation" presented before the Institution of Mechanical Engineers on the evening of 13th December, and in certain earlier writings has sought to establish the claim of the lubricant in the bearing and journals of a shaft to by the main source contributing the damping force.

We have now briefly indicated the technical history of this important subject from the early days of mishap to the enlightened—but still not fully enlightened—present. It is evident that the rationalisation of design with respect to torsional oscillation turns first upon the settlement of the factors responsible for the damping of the oscillations at resonance and secondly upon the establishment of a sound means of assessing the magnitude of the damping effect which they contribute. It seems now almost certain that the American view concerning the origin of the damping can, for the most part, be disregarded. The results of Dr. S. F. Dorey's investigation into the elastic hysteresis of crankshaft steels have been applied to show that the loss of energy through hysteresis in the steel is in general too small to account for more than a fraction of the damping required to obviate high amplitudes at resonance. It should be noted however that speaking in the discussion on Dr. Shannon's paper, Dr. Dorey conveyed the impression that he is not fully in agreement with this deduction from his results. Other considerations, nevertheless, support the view that hysteresis plays only a minor part, if any, in the damping of the oscillations at resonance. What we have called the Continental view cannot be similarly disregarded. It seems certain that in some cases the rise of resistance with speed of the machinery driven must exert a powerful damping influence although equally other instances might be named in which it seems doubtful whether it could be sufficient by itself to provide the amount of resistance requisite to suppress excessive oscillation at resonance. Dr. Shannon's belief in the efficacy of the lubricant in the shaft bearings as a damping agency is well supported by his experimental investigations and does no violence to our preconceived ideas when it is studied in full detail. Its adoption by designers would not necessarily require them to discard the Continental or the hysteresis theory. There is no reason why the damping influence should reside only in one factor. There is every reason to expect that in many cases it owes its origin to two or all three of the suggested causes acting simultaneously.

The Turbine-Driven Cargo Steamer "Hopestar".

"Engineering", 31st January, 1936.

On Wednesday, 22nd January, there was launched, without ceremony, in view of the national mourning, from the Wallsend shipyard of Messrs. Swan, Hunter & Wigham Richardson, Limited, a vessel which may prove to have inaugurated a new era in the propulsion of steamships of what is known as the tramp class. The vessel, named the "Hopestar", and built for Messrs. Wallsend Shipping Company, Limited, is of interest as being engined with geared turbines and high-pressure boilers with superheaters, a combination which was alluded to by Mr. John Johnson in his recent paper on "The Future of Steam Propulsion" as probably presenting a more formidable competitor to the Diesel engine for this class of ship than the machinery hitherto generally employed. The hull of the "Hopestar" is itself somewhat of a departure from the accepted practice of her class, a raked stem and a cruiser stern being embodied. The dimensions are 410ft. long between perpendiculars, by 57ft. 3in. in moulded breadth, by 38ft. moulded depth to the shelter deck. The deadweight-carrying capacity is about 9,800 tons on a draught of 26ft. The holds are planned for the stowage of large machinery and fitted for carrying grain cargoes. The whole double bottom, with the exception of a feed-water compartment, is arranged to carry water ballast, as are also the fore and aft peak tanks. A considerable amount of electric welding has been done in the internal structure. The deck equipment comprises 12 cargo winches, a warping winch, and 12 derricks. Electric light and steam heating are fitted throughout the accommodation and a refrigerated store chamber is provided. The classification is Lloyd's 100 A1.

The propelling machinery is of 2,000 shaft horse-power on a single screw. The engine consists of one high-pressure and one low-pressure turbine working in series and connected by flexible couplings to the opposite ends of a single high-speed pinion. Astern turbines are incorporated in both ahead turbines. The high-speed pinion gears with a wheel on a secondary shaft, the pinion on which gears with the wheel on the main shaft. The turbines and main condenser are mounted on the gear-case, an arrangement not only resulting in a strong and rigid construction, but enabling the alignment of the whole unit to be carried out in a single operation. The engine, which was constructed by Messrs. Parsons' Marine Steam Turbine Company, Limited, Turbinia Works, Wallsend-on-Tyne, was described and illustrated in some detail in "Engineering", Vol. 138, page 706 (1934), on the occasion of an exhaustive test made in the shops. When developing 2,323 brake horse-power the steam consumption was 8·6lb. per b.h.p.-hour. As this corresponds to the low figure of 7·6lb. per i.h.p.-hour, there appears to be every possibility that the vessel will run considerably more economically than one of similar power with reciprocating engines, and so realise the expectations of the operating company. As regards maintenance, the unit should also prove economical judging by the way in which the turbines of the T.S.S. "King Edward" have stood up to their work, it being stated that these turbines, fitted in the year 1901, are still in as good a condition as when originally installed.

Of the auxiliaries in the engine-room, a combination unit made by Messrs. Drysdale & Company, Limited, Yoker, Glasgow, is of interest. This unit, which comprises the main circulating pump, forced-lubrication pump, bilge pump, sanitary pump and oil-coiler circulating pump, is driven by a chain from the main turbine unit in normal steaming conditions. When the vessel is running at a slow speed, e.g., on entering or leaving port or in foggy or stormy weather, the unit is disconnected by means of a friction clutch and driven independently by a high-speed reciprocating steam engine. A full range of other auxiliaries is provided, including what is rather unusual on this class of vessel, viz., a lubricating-oil purifier and an oil separator on the auxiliary-exhaust range.

Steam is generated in three single-ended Scotch boilers arranged in line athwartship and made by Messrs. The Wallsend Slipway and Engineering Company, Limited, Wallsend-on-Tyne. Two of these are main boilers 14ft. 6in. in diameter by 12ft. 3in. long, constructed for a working pressure of 285lb. per sq. in. and having a combined heating surface of 4,306 sq. ft. and a grate area of 118 sq. ft. There are three furnaces to each boiler with separate combustion chambers, in each of which is a superheater manufactured by Messrs. The Superheater Company, Limited, Aldwych, London, W.C.2. The equipment is capable of raising all the steam generated to a final temperature of 750° F. A circulating connection is made to all the superheater elements from the donkey boiler, so that no damage can occur when the main boilers are being lighted. When superheated steam is not required—for instance, during standby periods—the steam is led through a de-superheater having its elements suspended between the nests of smoke-tubes, and thence to the auxiliary steam range or to the auxiliary condenser. The de-superheaters were also made by Messrs. The Superheater Company, Limited. The boilers are fitted with Howden's system of forced draught, and each smoke-box outlet is fitted with an air-preheater of the tubular type arranged for three passes of air. Diamond soot-blowers are fitted in the combustion chambers for cleaning both the main tubes and the superheater elements. The third boiler is a donkey boiler and is 12ft. 6in. in diameter by 10ft. 9in. long. It is constructed for a working pressure of 120lb. per sq. in., and supplies saturated steam only. There are two furnaces arranged to work under forced draught. The tubular air heater has in this case only one air pass. The heating surface of the boiler is 1,495 sq. ft., and the grate area is 39½ sq. ft. All the boilers are coal-fired, the bunkers being situated in the wings of the boiler-room. Reserve bunkers are provided in the 'tween-deck machinery space, and one of the holds can also be used for this purpose. The steam-generating plant and installation has been under the supervision of Messrs. Wilson & Burletson, Newcastle-upon-Tyne.

The late Sir James Mills, K.C.M.G. (Past-President).

[Portrait by Elliott & Fry.

OBITUARY.

Sir James Mills, K C.M.G. (Past-President).

It is with deepest regret that the death is announced of Sir James Mills, K.C.M.G. (Past-President), which occurred at Bournemouth on Thursday, January 23rd, 1936.

Sir James Mills, who was the founder and original managing director of the Union Steam Ship Company of New Zealand, was born in Wellington on July 30th, 1847. He was the third son of Mr. William Mills of Dunedin, where he was educated. He entered the service of Mr. John Jones, the Otago pioneer, who was then in business in Dunedin as a merchant and shipowner, and soon took over the active management of Mr. Jones's Harbour Steam Company. On Mr. Jones's death in 1869 he was entrusted by his partners with the sole direction and control. In 1874 he came to Great Britain and made the arrangements which resulted in the formation of the Union Company.

Mr. Mills, as he then was, was appointed managing director of the company, an office which he held until 1913. He had been chairman of directors since 1906, and was the only remaining member of the original board.

He rendered service to the community in various ways. He was Member for Waikouaiti in the Provincial Council of Otago from 1873 to 1875, and represented Port Chalmers in the General Assembly from 1887 to 1893. In 1907 he was appointed by the Government of New Zealand to represent it at the Imperial Navigation Conference. In the same year his services in assisting to develop the trade and resources of New Zealand were recognised by the honour of knighthood, in his case conferred for the first time upon one born in that Dominion, and in 1909 he was made a K.C.M.G. Sir James was also a Knight of Grace of the Order of St. John of Jerusalem.

Those who knew Sir James in his prime will understand how he came to build up from nothing the great business over which he presided for so many years. He had vision, capacity, industry, and the gift of securing and retaining the loyalty and affection of his staff. An instance of this occurred when, two or three years before serving as President of The Institute in 1916-17, the engineers of the company presented to him on his retirement, as a mark of their high appreciation, a handsome service of plate suitably inscribed. The presentation took place at a meeting of The Institute at 58, Romford Road, Stratford. Sir James had great charm of manner and a wide circle of friends, and with splendid health and vitality was able to enjoy life to the full.