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* Pumping Arrangements.

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Introduction.

The object of this paper is to give an outline of pumping arrangements so that the purpose of some of the numerous pipes, valves, cocks and fittings found in the average vessel may be appreciated.

There is not space for descriptions of the many patent appliances which are fitted in modern vessels; the makers will usually be only too pleased to supply information relating to their own fittings.

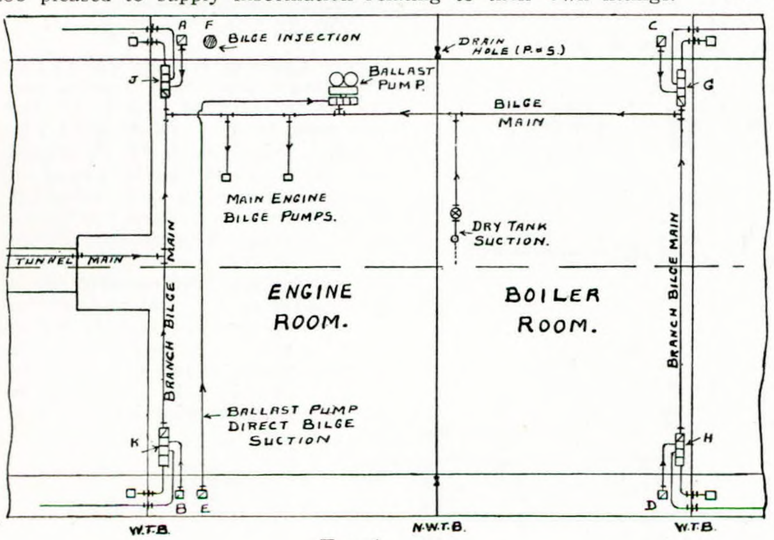


Fig. 1.

General Examination of Pumping Arrangements.

When an engineer first joins a ship, a quick look round the machinery space will soon show him the essential features of the main propelling engines and the auxiliary machinery.

The pipe connections situated above the engine-room floor plates can be examined and traced without undue difficulty, provided each system of piping is dealt with separately until that system is thoroughly understood. A good plan is to keep a reference book in which diagrammatic sketches of each system can be made. A similar procedure can be adopted in regard to the underfloor pipes, but here more difficulty will be encountered unless the owners have adopted the admirable practice of painting each pipe system a distinctive colour.

The builders' plans of the piping arrangements are usually available to all the engineers, and a preliminary look at these plans will be of great assistance, especially if the engineer has the necessary drawing-office experience to enable him to follow and separate readily the various piping systems. In any case, the plans give a good impression of the lay-out of the machinery, and form a valuable check on the engineer's own observations, but it must be borne in mind that in an old vessel alterations and additions may have been made.

When tracing underfloor pipes it is well to start with the bilge system and follow the pipes from the pumps to the main bilge line. A glance at the diagrammatic plan of bilge piping in the machinery space, shown in Fig. 1, will give an indication of the usual arrangement of this piping.

When crawling under the floorplates it frequently happens that obstructions such as auxiliary seatings or groups of pipes prevent one from following a pipe without having to come up above the plates and attempt the search further on. In such cases the pipe should be marked with chalk before leaving it, since it is not always easy to pick out the pipe under consideration when it is viewed from a different position.

The positions of the sea inlet chests for circulating water and donkey pumps should be located, as well as the position of the bilge

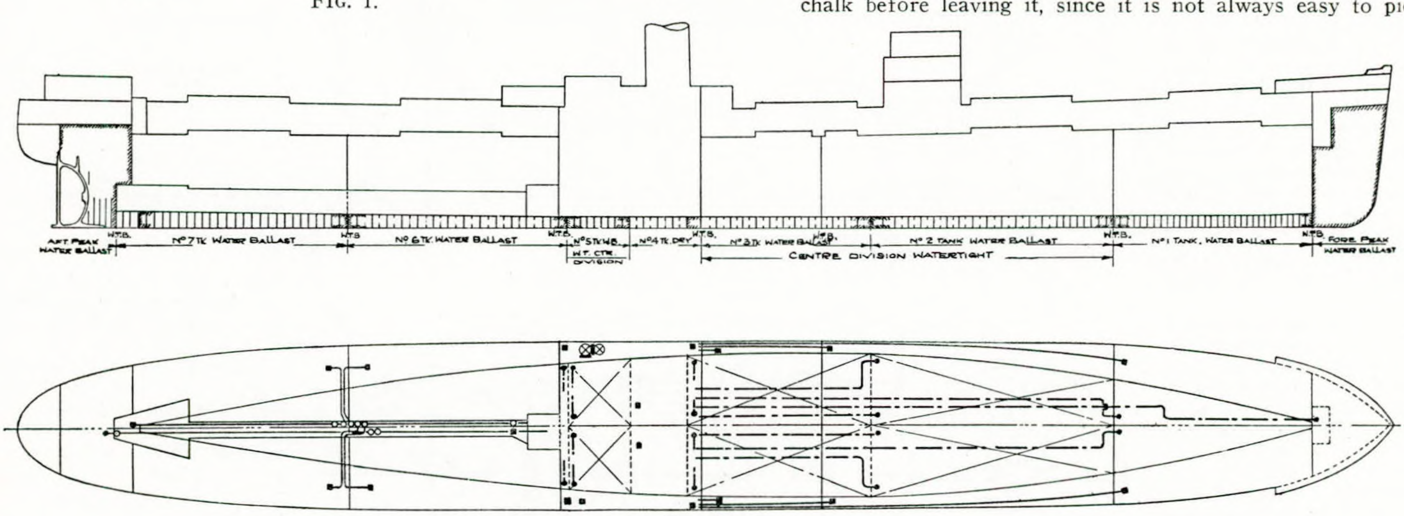


Fig. 2.

* This paper is included in the second edition of "The Running and Maintenance of Marine Machinery" Handbook, published by The Institute at 7s. 6d. plus 7d. postage. Other subjects included in the Handbook are steam turbines, steam reciprocating engines, boilers, Diesel engines, electrical machinery, refrigerating machinery, and steering gears.

injection. This emergency bilge suction is fitted to the suction side of the main cooling-water pump of all steam vessels and of some motor vessels, although in the latter vessels this suction is frequently replaced by an additional direct bilge suction from a ballast pump or other suitable donkey pump.

In order to understand the bilge system of a vessel it is necessary to learn the arrangement of the various compartments in the vessel.

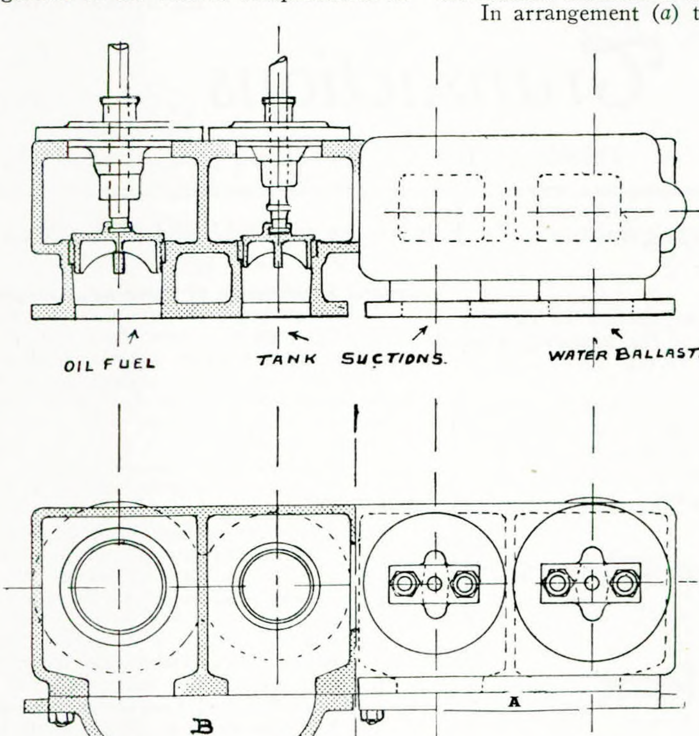
Figs. 1 and 2 show, in outline, the bilge systems of average cargo vessels, and it will be noted in Fig. 1 that the main bilge line in the engine room branches off at each end to connect groups of non-return valves known as bilge distribution boxes. Usually a mud-box is provided at each distribution box, in order to trap any solid matter which may be drawn into the branch suction pipes, and thus avoid choking the main bilge system.

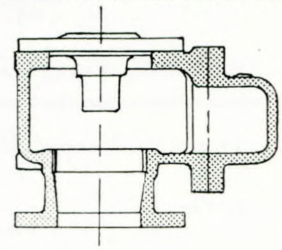
In modern vessels the machinery space bilge suction pipes are straight, leading down from mud-boxes situated at floor-plate level, so that the pipes can be readily cleared by means of a rod or stiff wire, but in older vessels it is usual to have the strainer at the foot of a bent suction pipe. This latter arrangement means "bilge diving" if the strum should become choked, and consequently it is always well to find out the easiest means of access to the strainer or strum, so that the clearing may be quickly carried out before the water in the bilges attains an uncomfortable depth.

Frequently, small hold spaces are constructed for use as deep tanks for the carriage of water ballast, oil fuel or other liquid cargoes, in which case it will be necessary to fit blank flanges in the bilge suctions to the deep tanks when the tanks are used for the carriage of water ballast or liquid cargoes. The blank flanging arrangements may take the form of change-over devices, shown in Figs. 3 and 4 and mentioned in the section dealing with cargo oil.

It will be appreciated that, with each watertight section of the ship connected to a common bilge main, bilge valves of non-return type are required in order to avoid communication between one watertight compartment and another in the event of any compartment being flooded and the valve spindles in the relevant distribution boxes being left in open position. Similarly, it is necessary to avoid the possibility of the sea being run into the vessel through bilge connections at pumps which have got both sea and bilge suctions, and this safeguard can be effected by the use of non-return valves or by the correct use of cocks. The latter fittings may, however, be a source of danger if the correct type is not used. To illustrate this point, the reader's attention is drawn to the arrangements shown in Fig. 5.

In arrangement (*a*) the use of a T-ported cock instead of an L-ported cock or an open-bottom cock having one port in plug, could lead to the direct admission of the sea to the

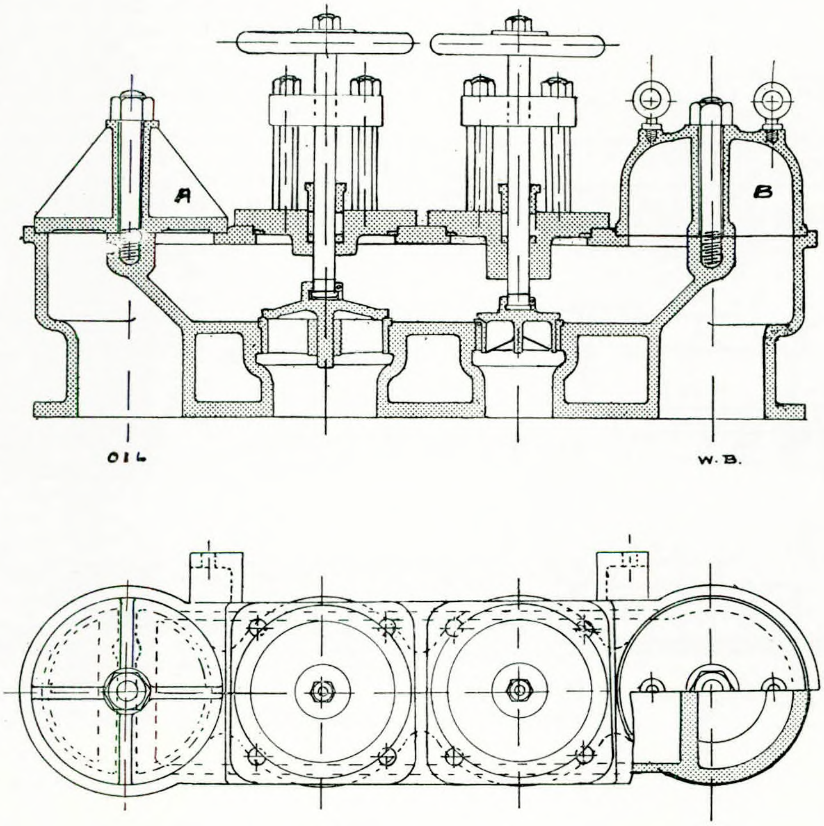


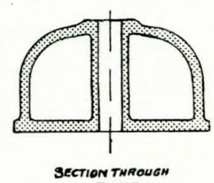


A & B ARE INTERCHANGEABLE.

FIG. 3.

bilges. Cases of flooding due to this arrangement have been known, and can be avoided if either of the arrangements shown in (*b*) and (*c*) be adopted. Similarly, in (*d*) the tunnel and the afterhold can be placed in communication with each other, if the T-ported cock is turned as shown in the sketch. The alternatives (*e*) and (*f*) do not possess this defect. Where, however, arrangement (*d*) is fitted, great care should be taken to see that the cock is turned with the ports—thus ⊥ when drawing from the tunnel well and thus ⊢ when drawing from the hold





A & B ARE INTERCHANGEABLE.

FIG. 4.

well. Fig. 5 (*g*) shows a suitable arrangement when three suctions are led to one cock.

An item in bilge drainage arrangements which deserves attention is the use of self-closing cocks or valves fitted on scupper pipes which drain one watertight compartment into another.

If cocks are used they should have parallel plugs so that they cannot be locked in open position by

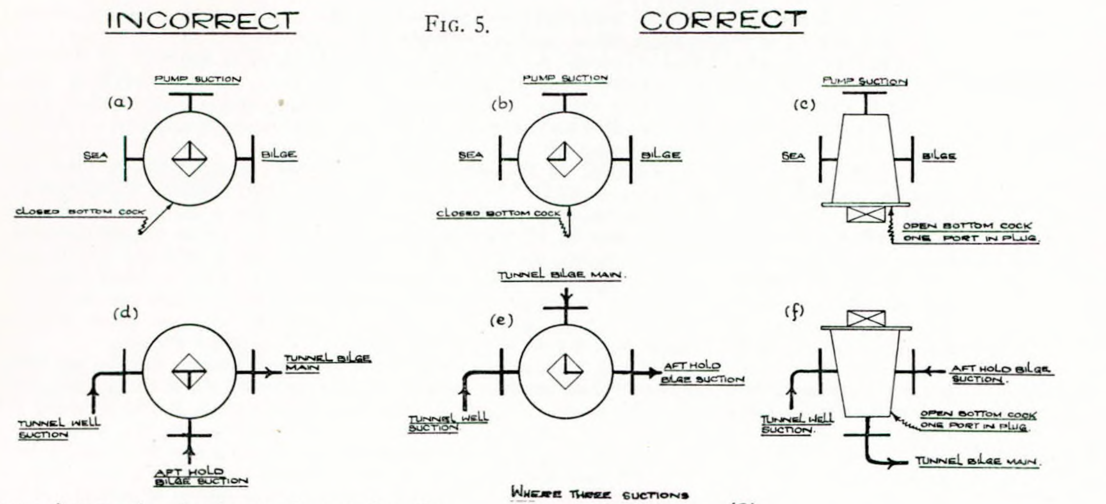
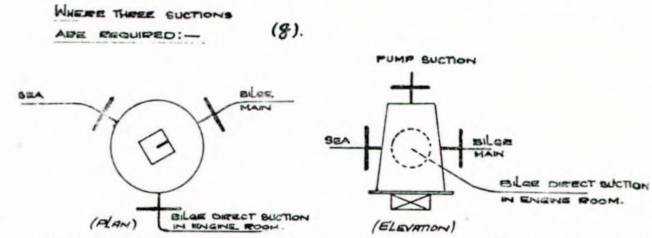


Fig. 5.



screwing up the gland nut. In practice, the plugs are usually provided with a slight taper, so that it is essential for the cocks to be worked from time to time in order to ensure their freedom of action. When valves are used they should also be tested, and care should be taken that they are not wedged open. Otherwise it may for instance be possible, in the event of such a casualty as collision, for several compartments to be flooded although only one was initially damaged.

A fitting which should receive more attention than it generally gets is the bilge injection which, as previously mentioned, is the bilge direct suction fitted to the suction inlet pipe of the main cooling water pump.

Of necessity the bilge injection valve is of non-return type, since normally the top of the valve is open to the sea through the circulating water suction. Since the valve is only used in cases of grave emergency, it may become encrusted with marine growths which would prevent its opening, and in order to prevent this defect arising, it is desirable to open up the valve for cleaning and inspection when opportunity offers. For the sake of security this should be done only in dry dock, unless it is carried out under the supervision of the chief or 2nd engineer, who can satisfy himself regarding the closing of the ship's side valves.

Water Ballast Systems.

The rate at which ballast tanks are to be filled or emptied is usually a matter for the owners to decide in relation to the needs of the vessel's particular trade. Thus, a vessel loading or discharging bulk cargoes in ports where efficient mechanical loading or unloading devices are employed may require to have its water ballast dealt with in 8 or 10 hours, whilst a vessel carrying a general cargo, involving a more prolonged stay in port, may have a week or more for the working of the water ballast. Generally, it is preferable to "run" the tanks up from the sea and finish off by pumping to ensure that the tanks have been completely filled or "pressed-up" as it is generally called. Peak tanks and deep tanks can, of course, only be run-up to the depth of water the vessel is drawing, and must be completed by pumping or by hose from the wash-deck line.

If a "tank injection", that is to say a valve on the ship's side for placing the ballast line in direct communication with the sea, is not provided, the running-up can be done by opening the ballast pump suction valves for sea and tank connections. Sometimes this is accomplished without interfering with the use of the pump for other services, if the suction chest is made as shown in Fig. 6.

In this arrangement, which is not very common, the sea is run into the tanks through the water ballast main by opening valve *A* and the valves in the distribution chest controlling the suctions to the tank which it is desired to flood. When pumping out the tank, the valve *B* should also be opened and the ship side valve on the sea inlet pipe shut.

When pumping tanks, care must always be taken that the plugs or other closing appliances fitted to the end of the tank air pipes are removed, in order to avoid serious damage to the structure of the tanks—especially deep tanks and peak tanks.

It is always advisable to slow down the pump when a tank is nearly full and to arrange for someone to keep a watch on the tank in order that word can be passed to the man at the pump when the tank is full.

Water ballast connections to deep tanks which are also used as hold spaces for the carriage of cargo must be blanked off when cargo is carried in these compartments, in order to prevent leakage or accidental flooding.

Provision against possible damage to the structure of water ballast tanks can be made by means of pressure relief devices on the ballast pump discharge to the tank. The use of spring-loaded relief valves for the low pressures essential for the protection of the structure of the tanks is not so effective as the arrangement shown in Fig. 7.

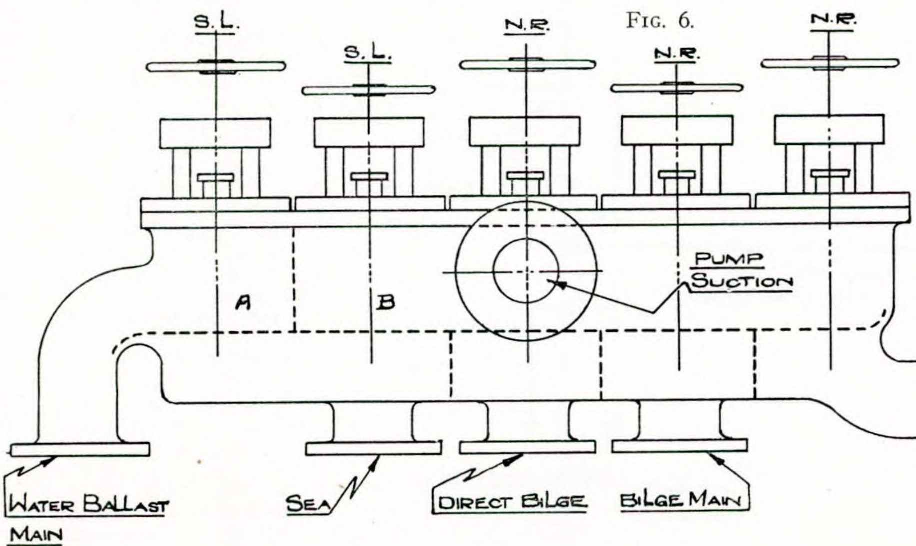
Whatever form of device is used, it should be fitted on the discharge pipe leading to the tanks rather than on the pump, since the pump may be required to exert much higher pressures when used for fire or wash-deck services.

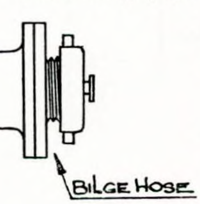
Oil Fuel Systems.

The oil fuel main is usually a single pipe line with branches led from convenient points to the various groups of valves controlling the tank suctions. A typical arrangement is shown in Fig. 8 and it will be seen that since the filling pipe is connected to the oil fuel main, this is really a combined suction and filling main, the tanks being emptied and filled through the same connections. In some vessels separate lines are used for filling the tanks, but such cases are not common.

The taking on board of oil fuel bunkers is an important operation, and every care should be taken to prevent the oil overflowing either on the deck or into the dock. To ensure this it is essential

Fig. 6.





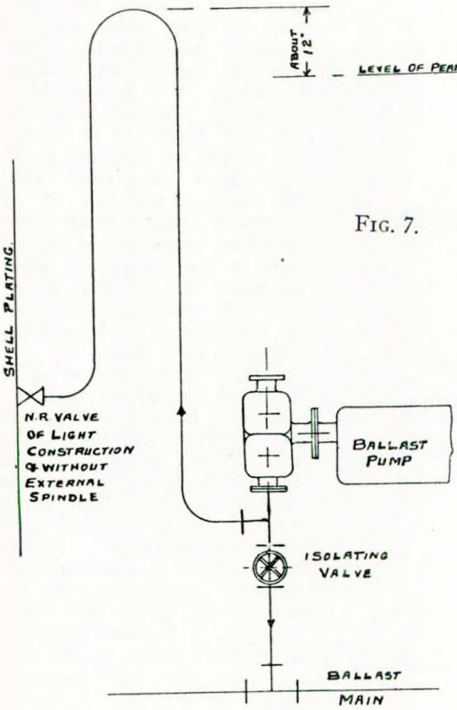


FIG. 7.

that the air and overflow systems to the oil fuel tanks should be clearly understood.

In the smaller class of vessel it is usual for the tanks to overflow through their individual air and overflow pipes led to the weather deck, terminating in the familiar swan neck vent with wire gauze diaphragm, but in larger vessels overflow systems may be used. Their arrangement will be according to the number and disposition of the storage tanks.

A simple and direct system of overflow pipes is shown in Fig. 9. The deep bunkers of the forward boiler room overflow into an overflow tank situated within this compartment, and a similar arrangement is followed in the case of the after boiler room.

It is not always practicable to have an overflow tank situated in each watertight compartment of the vessel which contains oil fuel storage tanks, and accordingly there may be a group of bunkers and double-bottom oil fuel tanks situated within a number of different watertight compartments, but all connected to one overflow tank. Sometimes there is one battery of tanks complete with its own overflow tank situated forward of the machinery space, and a similar arrangement aft.

Looking at the structure of oil fuel storage tanks a false impression is apt to be gained of their strength, whereas the maximum pressure head for which the tanks are suitable is frequently 8 feet of water (that is, 3½lb. per sq. in.) above the crown or top of the tank. It is therefore essential that undue pressure should not be allowed to come upon the deep storage tanks, whether they be filled from the shore or by pumping up from another tank by means of the ship's transfer pump, and it is always advisable to slow down the oil supply to the tank before the oil has quite reached the top.

It is a good plan to adjust the rate of filling so that as one tank is nearly full another tank is three-quarters full, a third tank half full and so on. Filling of the port and starboard tanks should be carried out simultaneously in order to avoid giving the ship a list.

When completing the filling of tanks which overflow on to deck or overboard through non-return valves on the ship's side plating, special care is required to slow down the oil supply to avoid sudden rushes of trapped air and oil fuel, which would result in an objectionable mess on the deck or in the dock.

Where vessels are engaged on long voyages and require to carry a large supply of oil fuel, storage tanks are sometimes built into the ship's structure in way of the tunnels. Fig. 10 shows a typical arrangement, in which the full lines indicate an overflow ring main system led along the upper portion of the tunnel to two separate overflow tanks, whilst the dotted lines show the larger separate overflow mains required where only one overflow tank is fitted. This is an example of a battery of tanks situated within one watertight division of the vessel, say No. 4 hold. Often, however, deep oil storage tanks extending on each side of the tunnels from the engine room to the aftermost hold are utilized for the storage of oil fuel.

In such cases the use of an overflow main below the load water line will necessitate the provision of non-return valves in order to prevent inter-communication between these large storage tanks in the event of any one tank being bilged, *i.e.* placed in communication with the sea due to a casualty such as collision or heavy stranding.

Alternatively, the overflow main may be placed in a position well above the load water line close to the deck to which the watertight bulkheads extend.

Some owners make the non-return valves of screw-down type to facilitate tank testing during survey, but this introduces a risk of damage to the tanks during filling operations. In these cases it is advisable that a warning notice should be fitted at the filling station and transfer pump to the effect that all valves must be opened on the overflow pipes of the tanks affected before pumping operations on the tanks are commenced.

It is particularly important that the overflow arrangements for settling tanks are understood, since these tanks are frequently pumped up. The overflow pipe may be led to (1) a deep overflow tank, (2) a double-bottom tank, (3) deck, or (4) oil wells in the engine-room

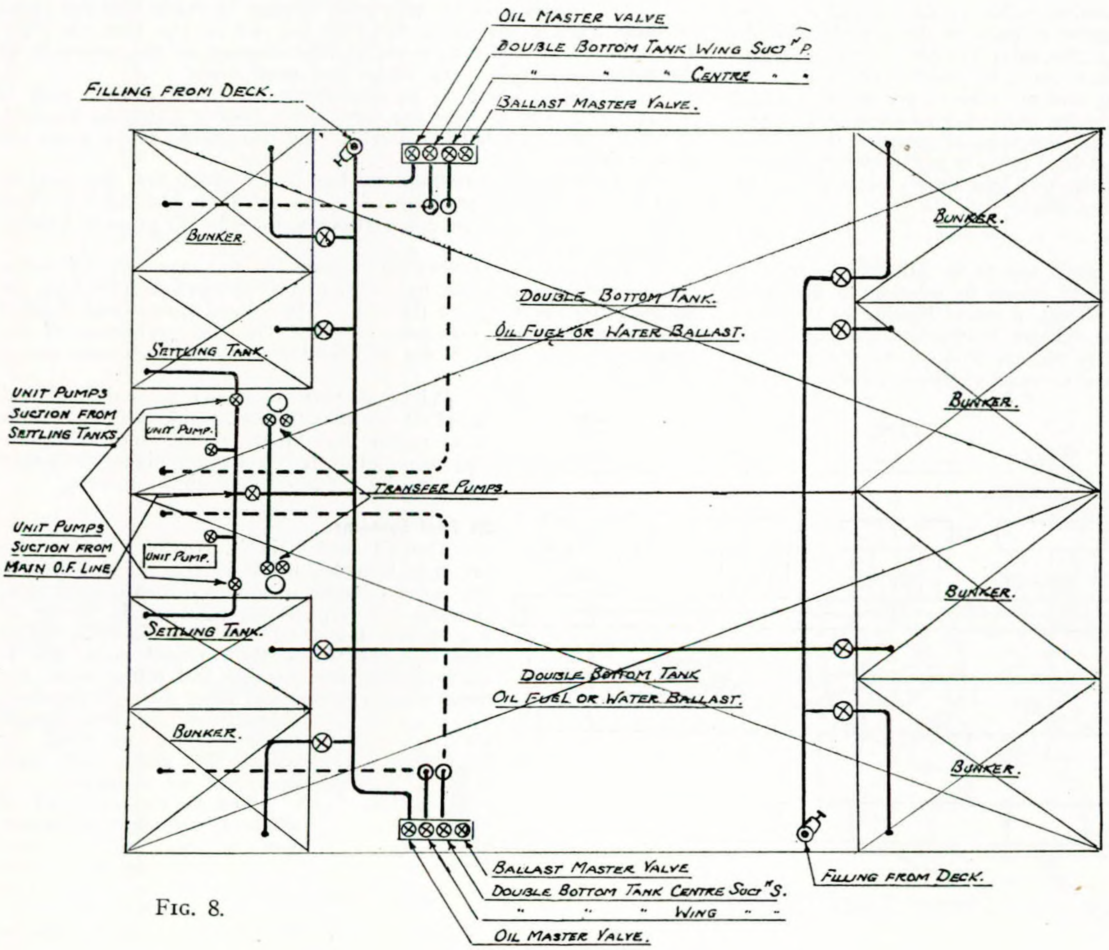
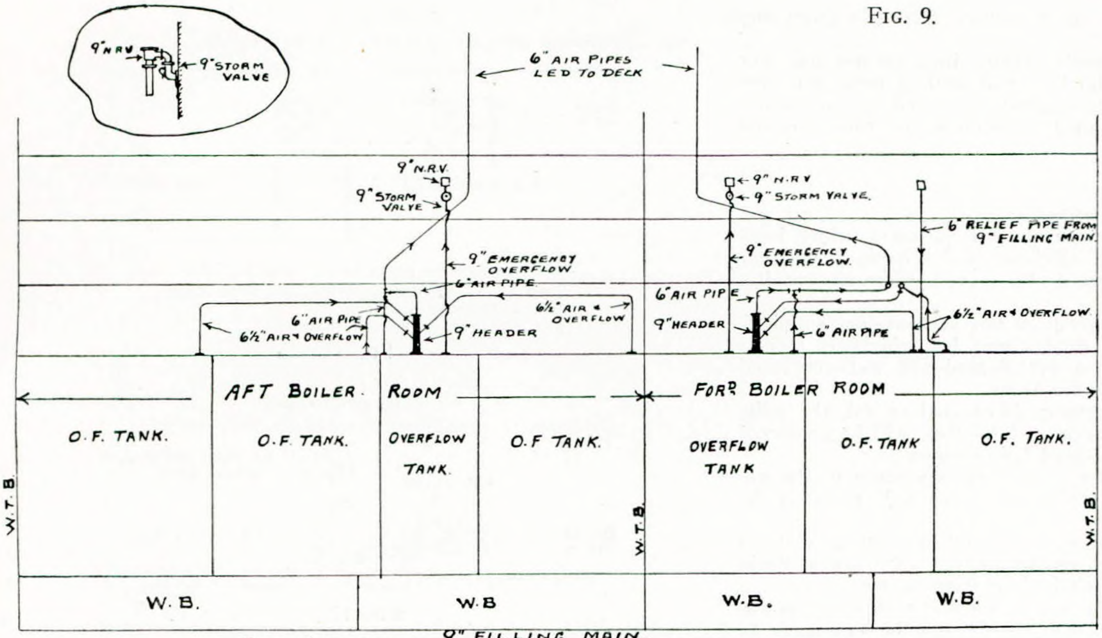


FIG. 8.

FIG. 9.



ONE 6" OIL FUEL SUCTION & FILLING CONNECTION TO EACH TANK.
(AIR PIPES TO W.B TANKS NOT SHEWN).

bilges. Of these arrangements the least desirable is No. 4, since vapour is bound to be discharged into the machinery space, and this, coupled with any oil which may be discharged, constitutes a danger from the point of view of fire.

The "open" overflow to the oil well has the advantage of both oral and visual warning of the tank overflowing, but it is incomparably better to have a "closed" overflow system leading to an overflow tank or to a storage tank utilized as an overflow tank. In some vessels the overflow pipe is led near to the transfer pump and a well-lighted sight glass provided so that there is no difficulty in ascertaining when the tank overflows.

When the overflow pipe is led into a double-bottom tank which is also used for the carriage of water ballast, a non-return valve should be fitted to the overflow pipe to prevent the possibility of water being introduced into the settling tank when the double-bottom tank is being pumped up with water ballast. Frequently the settling tank is too high to be affected thus, but this is not always the case.

It is not unusual for oil fuel and water ballast tanks to be connected to the same overflow main as other tanks which are reserved exclusively for oil fuel. If the former tanks can be pumped up with water ballast it will be necessary for their overflow pipes to be provided with change-over pieces or with ring and blank flanges, so that when used for water ballast they can overflow through shipside non-return valves to the sea or on deck instead of into the oil fuel overflow system. An arrangement of settling tank overflow which is not uncommon in older vessels is to lead the overflows into the drip tray under the tanks, which is usually provided with open scupper pipes, or with a suction pipe to the oil fuel transfer pump, so that the oil which overflows may be recovered.

The danger of this arrangement is that overflows of oil are accidental, due to neglect or to the engineer on watch having his attention directed to some other duty, and consequently the overflow may continue at full bore without being noticed until the drip tray overflows. The oil may then flow on to any heated surfaces situated below the tanks, such as boilers, steam or exhaust pipes, and cause a serious fire.

ALARM DEVICES.

These are sometimes fitted to settling tanks to give warning when the tanks are, say, 95 per cent. full.

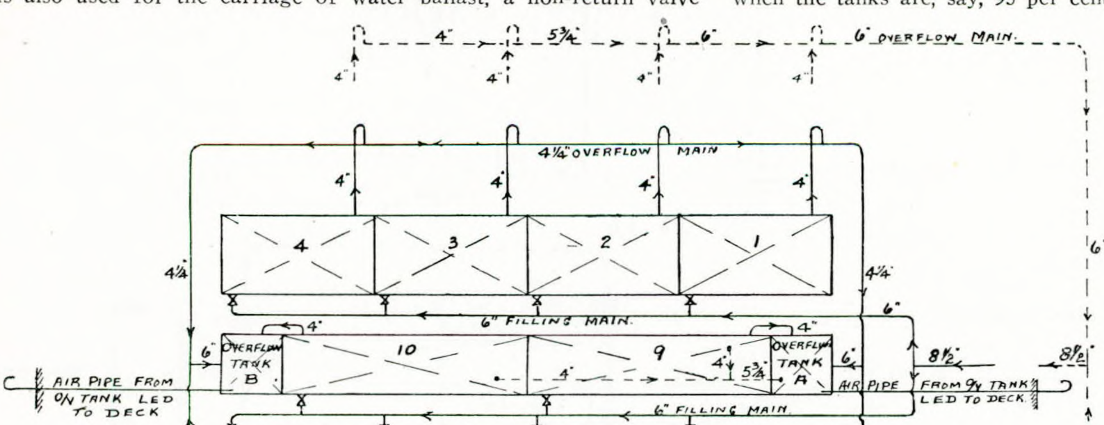


FIG. 10.

FILLING BRANCH TO EACH TANK 4" BORE.
——— OVERFLOW SYSTEM OF RING MAIN TYPE.
- - - - " " WITHOUT RING MAIN & TANK B.

Automatic devices are, however, apt to go wrong now and again, and it is always advisable to pay careful attention to the whole operation when pumping up a settling tank.

LEVEL-INDICATING DEVICES.

In order that the level of oil in settling or daily service tanks may be ascertained at a glance, it is customary for some sort of level-indicating device to be employed. This may take one of the three following forms, viz. float gear, gauge glass or pneumatic gauge.

The first device is simple and reliable, but it is sometimes overlooked that the tube through which the float rod or chain emerges is not oil-tight, in which case care should be taken not to "press" the tank unless the tube is led above the level of the overflow pipe.

The use of gauge glasses on oil fuel tanks is regrettable. For reasons of safety they should be fitted with self-closing cocks and will therefore only give an indication of the true level of oil in the tank when the cocks are held open, whereas the indicator of the float type gives an immediate record without any manipulation.

Pneumatic gauges are deservedly coming into greater use, not only for the deep tanks such as bunkers and settling tanks but also for double-bottoms. A hand-book is usually supplied by the makers of pneumatic gauges, giving detailed instructions on the care and upkeep of such outfits.

Oil Burning Arrangements.

OIL FUEL BURNING UNITS.

There is little scope for comment about these units, which have been developed to a high state of efficiency and reliability, whether they include the steam-driven pumps or the more modern electrically-driven pumps.

There is much to be said in favour of two separate or "simplex" units in lieu of the less accessible duplex unit, but whichever type is employed it should be situated in a well-lighted and well-ventilated position in the engine room or boiler room.

If the vessel is fitted for burning either coal or oil, the units should be situated in the engine room, where they will be protected from the dust and dirt of a coal-fired boiler room.

To an engineer who has had no previous experience in the use of oil burning installations, the following notes will probably be helpful :—

(1) Before changing over on to the stand-by settling tank, it should be ascertained by means of the self-closing drain cock or valve that the tank is free from water.

(2) The tank in service should not be allowed to become nearly emptied before changing over, as it may be necessary to change back again to this tank if any hitch should occur to prevent the pump obtaining oil from the full tank.

(3) The best temperature and pressure for burning oil of a certain type and quality should be found by experiment.

(4) Where suction filters of the basket type are fitted it is essential that these should be regularly and carefully cleaned.

(5) If carbon forms at the back of the furnace it should be raked well forward at the beginning of each watch; it will then burn away.

(6) Smoke may be caused by too high or too low temperature of the oil, too high oil pressure or an insufficient supply of air caused by the air baffles not being correctly regulated. Smoke is, however, more often than not due to bad combustion, caused by lack of attention in keeping burners clean. The ideal condition is that where a faint haze shows at the funnel top, indicating that the air supply is just right. Panting or vibrating of air casings and smoke-box doors generally points to insufficient air admission.

(7) The stokehold platforms, tank tops, bilges and units should all be kept

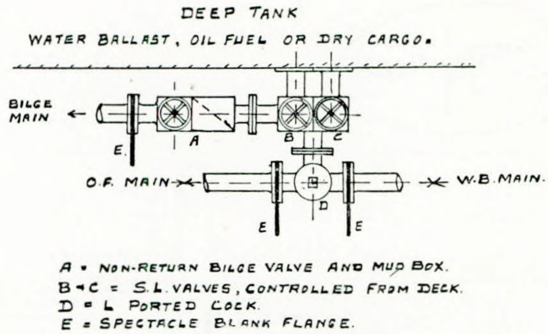


FIG. 11.

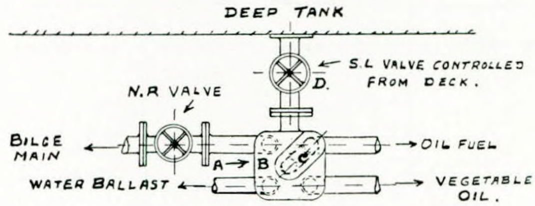


FIG. 12.

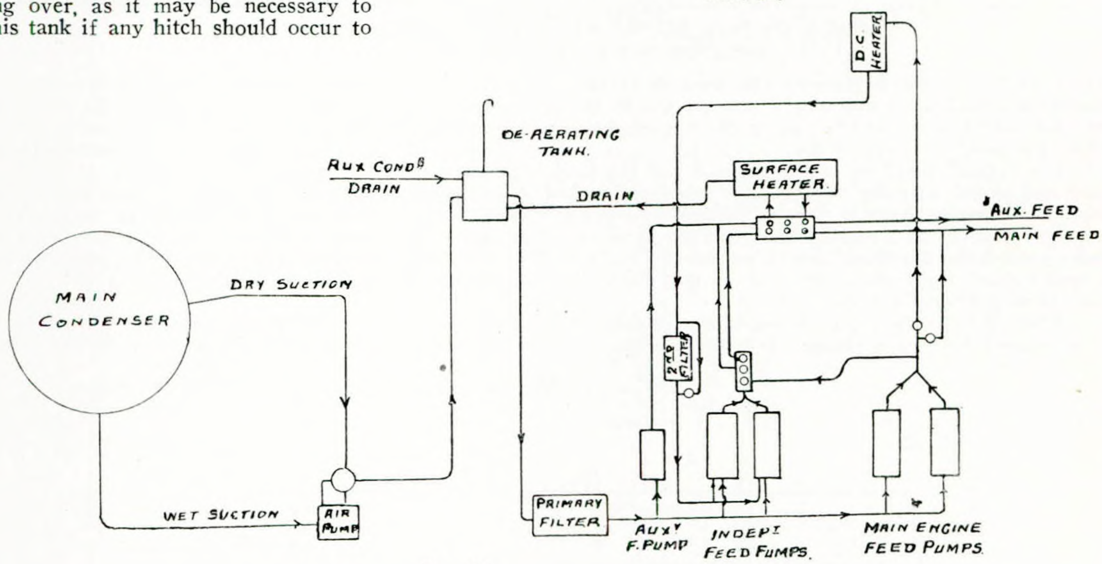


FIG. 13.

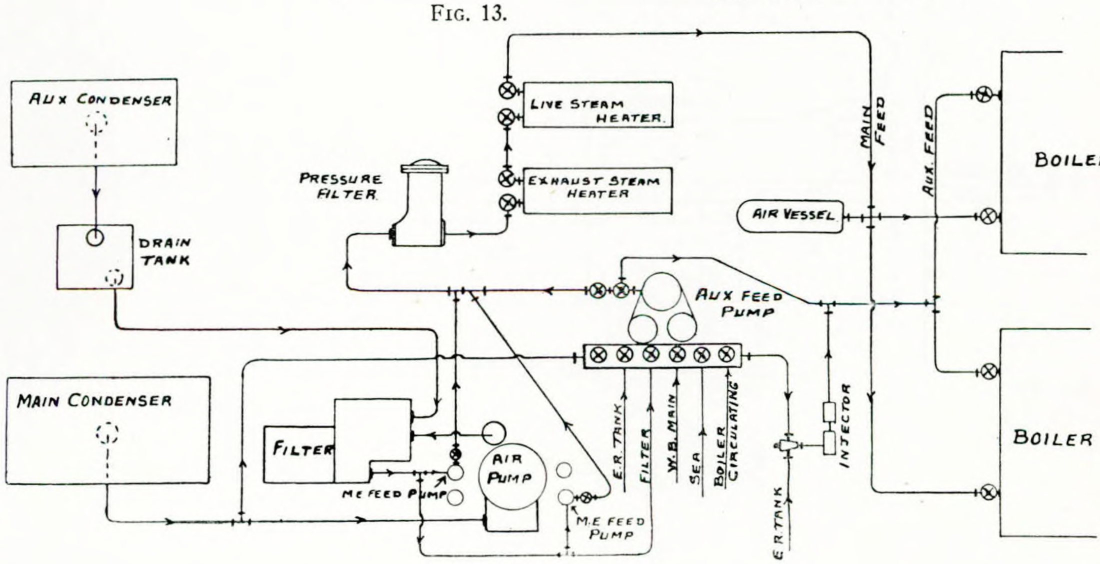


FIG. 14.

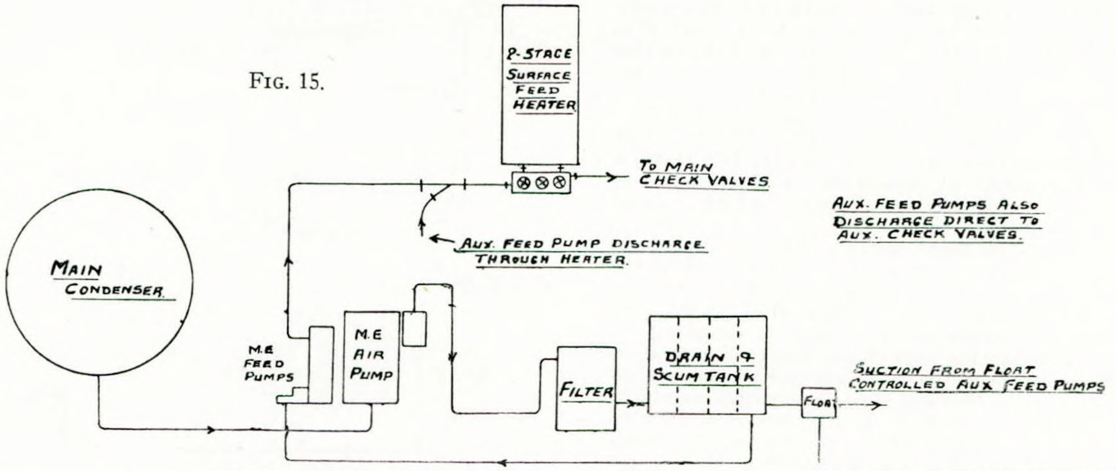
scrupulously clean.

(8) Should there be any sign of leakage of oil fuel, especially on the discharge side of the pressure pumps, the matter should at once be investigated and steps taken to remedy the defect.

Cargo Oil Systems (other than those on oil tankers).

Increasing use is being made of general cargo vessels for the carriage of oil cargoes which have a flash point above 150° F., such as vegetable oils, wood oils, molasses, heavy oil fuel, and latex.

FIG. 15.



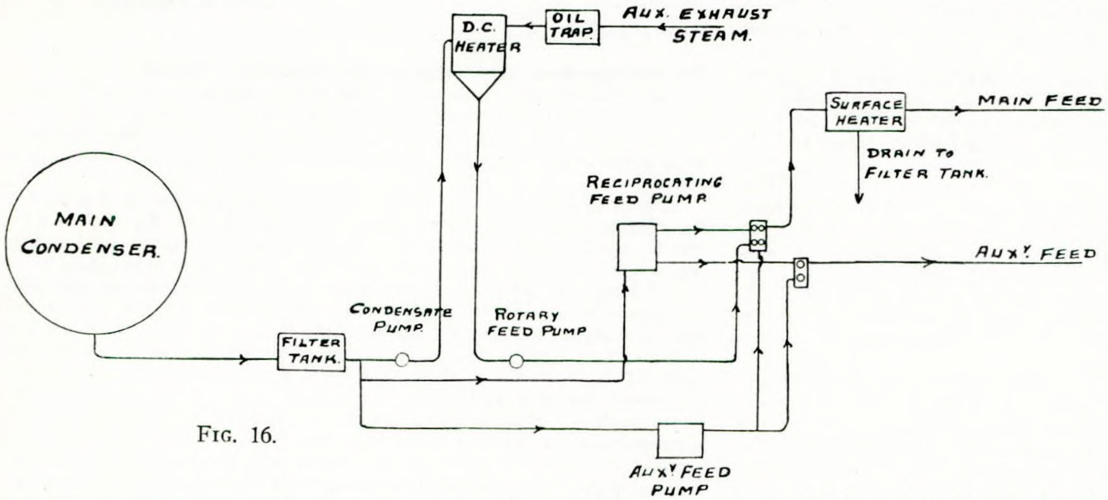


FIG. 16.

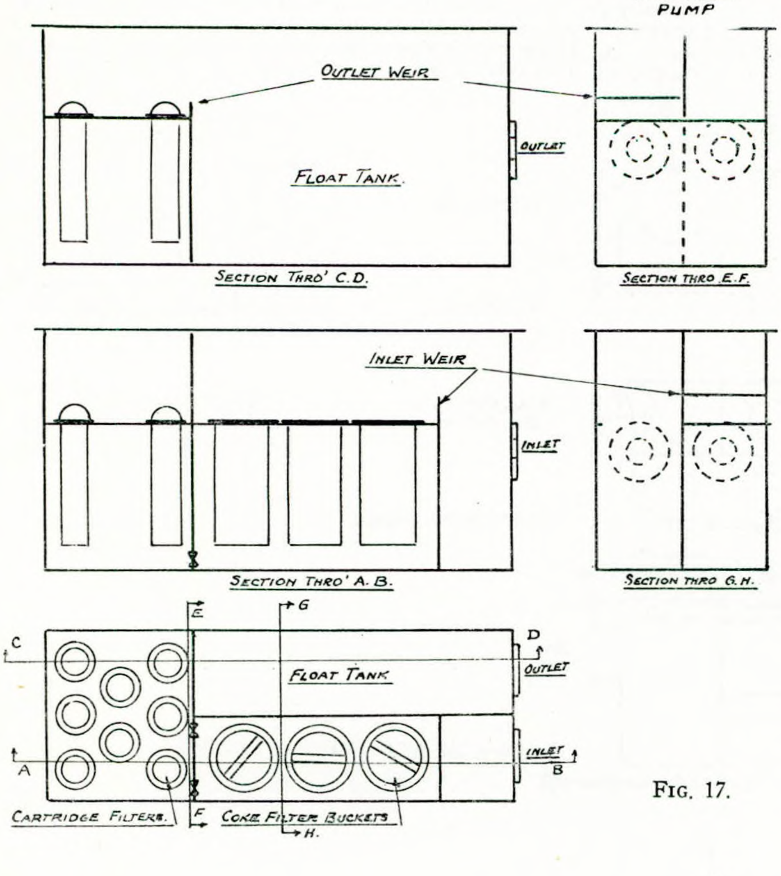


FIG. 17.

Generally, a special cargo oil pump is fitted in the engine room, but in some cases the ship's ballast pump or transfer pump is used for dealing with cargo oil, and in others shore pumps only are employed.

With valuable oils liable to contamination, absolute cleanliness of tanks, pipes, fittings and pumps is essential and great care is required in the use of the change-over devices or the fitting of open or blind flanges, in order to separate the cargo oil from other liquids such as bilge water, oil fuel, or sea water.

In order to make clear the references to change-over devices, the reader is referred to the arrangements shown in Figs. 3, 4, 11 and 12. Fig. 12 shows the most foolproof arrangement, and consists of a chest (*A*) having a square cover (*B*) with hollow passage (*C*) from one corner to the centre aperture, which is connected directly to the deep tank valve (*D*). By changing the cover round, the centre aperture, and therefore the tank, can be placed in communication with any one pumping system, whilst the other systems are automatically blanked off.

When cargo oil other than oil fuel is carried immediately above double-bottom tanks used for the carriage of oil fuel, care should be taken to ensure that the double-bottom tank is not pressed-up, in case of possible leakage into the cargo oil tank.

Feed-water Systems.

In superheated steam installations the amount of lubricating oil required in the cylinders of reciprocating engines, whether main or auxiliary, makes it essential to provide efficient means for eliminating the oil which is eventually found in the condensate. For this purpose gravitation filters are the most efficient, but in many installations pressure filters are also employed. Various arrangements of feed-water systems are shown in Figs. 13, 14, 15 and 16.

It should be noted that where exhaust steam is used in direct-contact heaters, the oil in the steam will inevitably find its way into the feed water, and it is necessary that the feed water from the direct-contact heater should be passed through an effective filter, or the exhaust steam supply to the heater be passed through an oil separator. It is desirable that the filters, whether of the gravitation type on the suction side of the pumps or of the pressure type on the discharge side of the pumps, should be in duplicate, so that one filter may always be in use and thus ensure the continuous filtration of the feed water. Where this is not the case, the spare filtering cartridges or other material should be handy, so that a quick renewal of the filtering materials may be effected. Incidentally, preparations of this sort reduce to a minimum the time spent on a hot steamy job.

When a scum tank forms part of the gravitation filter a scumming ladle should be attached to the tank by means of a light wire or chain, so that the oil floating on the surface of the condensate may be removed as often as found necessary. A sketch of a gravitation filter is shown in Fig. 17.

The elimination of lubricating oil from feed water has been dealt with briefly, but there is another source of contamination which is not so obvious and is due to ingress of oil fuel or cargo oils when these are carried on board.

The most usual source of ingress of oil is by way of oil fuel heaters and the heating coils in the oil tanks, due to the failure of a tube or pipe or to a defective joint. So that all the condensate from the heaters and coils may be examined before entering the hotwell an "observation tank" is therefore necessary. As a precaution, it is preferable that this heating steam condensate should be passed through a filter. If oil is observed, the condensate should be discharged to the bilge by means of a cock or valve provided for this purpose, and should not be allowed to enter the hotwell until the leaky connection has been traced and shut off and the system purged of oil.

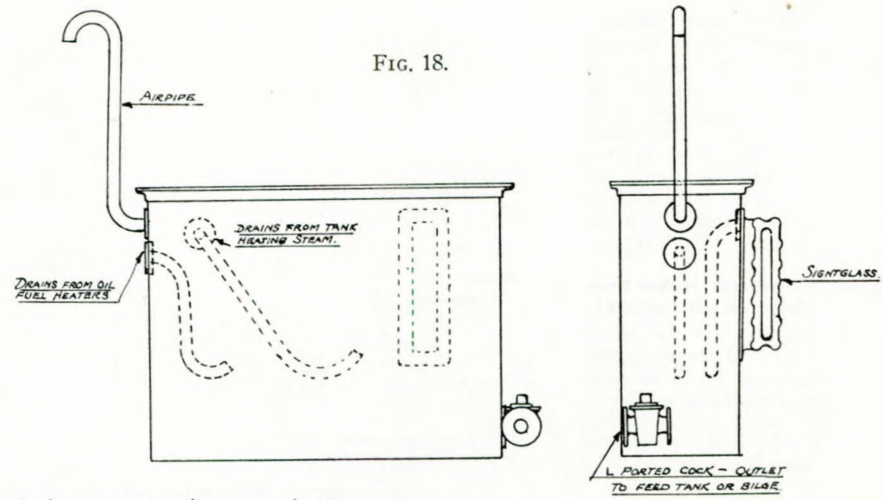


Fig. 18.

In tankers carrying low flash oil, particular care should be given to the observation tank if heating coils from the cargo oil tanks are connected to it, and on no consideration should a naked light be brought near the tank. A sketch of a typical observation tank is shown in Fig. 18.

Another possible source of feed-water contamination by oil is the connection of an auxiliary feed pump to a ballast line which may be used for pumping water ballast out of tanks which were previously used for the carriage of oil fuel or cargo oil. The ballast line thus becomes a "dirty ballast line", and connections from it to pumps which are used for boiler feeding should be blanked off. Similarly, a feed pump should not be used for pumping from the bilges of vessels carrying oil fuel, since the bilge water is almost certain to contain oil.

If, however, the connections mentioned above are provided and are not blanked off, care should be taken to wash out the feed pump by pumping from the sea to overboard before changing over, on the suction side, to the hotwell tank or to the direct connections to the feed-water storage tanks and, on the discharge side, to the boilers.

Pumps.

The smooth working of a steam pump of either duplex or simplex type is largely dependent upon the adjustment of the steam valve gear, and any close attention to this point is amply repaid. In simplex types valve gears of special and sometimes complicated design are used, and it is always advisable to look out for the makers' adjustment marks on the valve spindle and to work to the firm's instructions in this respect.

In rotary pumps fitted with self-priming devices it is necessary that these devices should be kept in good order, or trouble may arise when the vessel is in a heavy sea-way and the bilge suctions are exposed from time to time, thus admitting air into the bilge suction pipes. The internal arrangements of suction and discharge chests of pumps may not always be ascertained by outward inspection (as can be seen from Figs. 6 and 19) and a suitable opportunity should be taken to open up the chests for inspection if there is any doubt, care being taken that the valves on the other ends of each of the connections are shut.

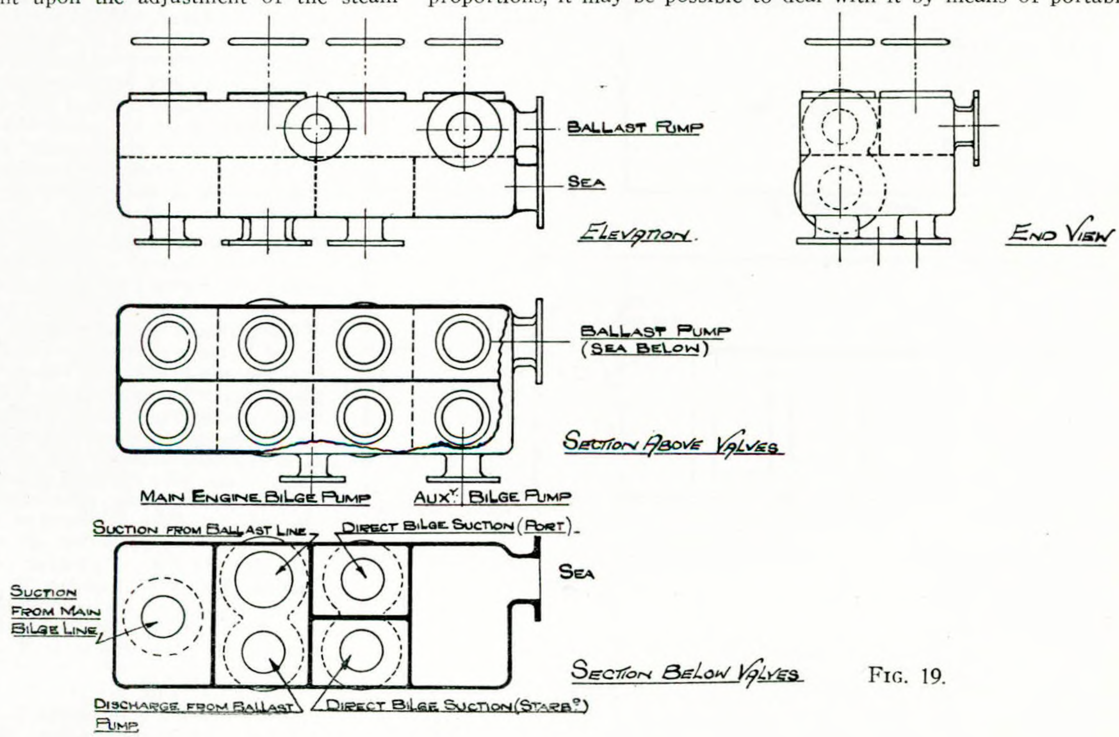


Fig. 19.

Fire-extinguishing Apparatus in the Machinery Spaces.

In coal-burning vessels any fire which occurs is most likely to take place in the bunkers, due to spontaneous combustion of the coal; especially is this so in bunkers situated in very hot positions, such as a cross bunker situated between the boiler room and the engine room.

Fires of this nature are difficult to deal with at sea, and probably the closing of bunker doors, vents and hatches, and the use of CO_2 gas if available to smother the fire, are the best means of tackling the job, provided there is coal available from some other bunker.

Where oil fuel is used, adequate provision for fire-extinguishing is necessary, and in cargo ships it usually consists of steam smothering. The steam is taken direct from the boilers or from the auxiliary steam range, and is led to a series of horizontal perforated pipes situated between the tank top and the floor plates.

Since the idea underlying the injection of steam into the boiler room is the exclusion of a supply of oxygen to the burning oil, arrangements should be made to close doors, fiddley gratings and ventilators. The stop valve for admitting the steam to the perforated pipes must necessarily be controlled from outside this compartment, and is usually fitted with an extended spindle led to the casing top. If the fire is discovered before it has reached serious proportions, it may be possible to deal with it by means of portable chemical fire extinguishers, and to ascertain the source of the fire so that immediate steps may be taken to shut off the oil supply.

For this purpose it is necessary that the suction valves on the oil fuel settling tanks should be controlled from outside the compartment in which they are situated, and that the pump or pumps handling the oil fuel should be stopped, if desired, by means of similar distant controls. One point to bear in mind is the necessity for prompt and decisive action. It is of no use stopping the pump which supplies the burners if immediate use is not made of the available steam supply for smothering the fire and driving the pump used for supplying water to the fire main, since the steam pressure will fall rapidly once the burners are extinguished. This point illustrates the advantage of CO_2 for fire-smothering, or of the various patent systems for covering the burning oil on the tank top with a coating of air-excluding foam.

Fire Prevention.

However peculiar it may sound, it is probably true that the best way to deal with a fire is to prevent its occurrence. Accordingly, scrupulous attention should be paid to the cleanliness of the compartments containing the oil fuel apparatus and storage tanks, and to the avoidance of any accumulation of loose oil. Failure to see that the jointing material of oil fuel pipes is suitable for the purpose may result in a leaky joint and possibly a destructive fire, especially in the case of pipes conveying heated oil to the burners.

Another source of trouble may be the opening up of some part of the oil fuel system, while the remainder of the system is in service. For instance, in the case of two settling tanks having a common air pipe, if a joint were being remade on one of the tanks, the filling of the other tank would require close supervision, since in the event of its overflowing, oil could pass into the tank under repair and might then run out through the incompleted joint.

Occasionally, the extended spindles of the valves supplying steam to the fire-extinguishing pipes and to the oil fuel pumps, also the rods to the suction valves of the settling tanks or other deep tanks, may be in bad working order or even entirely removed, so it is always advisable to make sure that these controls are in good order, since their effectiveness may at some time be the means of saving the vessel.

Steam Pipes.

By far the most important matter in connection with these pipes is the question of drainage, so that water hammer may be avoided. The engineer on board a vessel must accept the existing arrangements and make the best of them until alterations can be made if found necessary.

Long lengths of horizontal pipes, whether horizontal in a transverse or a fore-and-aft direction, are specially liable to accumulate water if the ship should take a list or alter trim, and this factor should be taken into account before opening up the stop valves.

When one boiler is in use and the other boilers are idle, water is sure to accumulate in the branch pipes leading to the boilers which are not in use, unless isolating valves are provided at the junction of the branch pipes with the main steam pipe. Even then there may be some water present in the branch pipe if the isolating valves are not perfectly tight, and it is always advisable to drain each pipe carefully before easing the stop valves.

In installations employing superheated steam, the arrangement of the pipe lines is often confusing, and a sketch of a typical arrangement is shown diagrammatically in Fig. 20. Usually there is

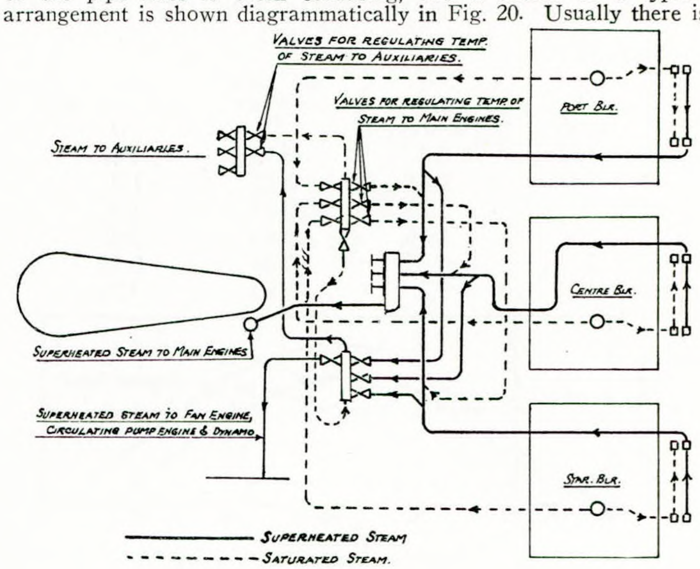


FIG. 20.

some provision for circulating steam through the superheaters when the main engines are not working, in order to avoid the superheater tubes being overheated due to the reduced flow of steam.

In earlier installations this circulation was effected by keeping open the superheater header drains, a practice which involved the loss of a considerable quantity of fresh water if the drains were not connected by piping to a drain-collecting tank.

Finally, if a "blow" is heard on the boiler tops or in the steam pipe line, it is always wise to investigate. If it is due to a defective gland it can usually be remedied by nipping down the gland, but if the noise persists an inspection should be made of the mountings to make sure that the cause is not a fracture of a valve or pipe fitting—a defect which might well lead to a fatal accident.

JUNIOR SECTION.

Naval Architecture and Ship Construction (Appendix contd.)

By R. S. HOGG, M.I.N.A.

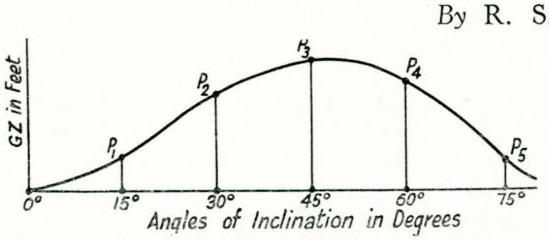


FIG. 202.—*Statical curve of stability for 8,000 tons displacement.*

To obtain the ordinary statical curve of stability, draw a vertical line on the cross curves at the appropriate displacement and "lift" the ordinates. Then set them up on a base of angles. Thus, Fig. 202 is a statical curve derived from Fig. 201 for a displacement of 8,000 tons. The heights of P_1, P_2, P_3, etc., have been measured from the base line in Fig. 201, and set up at 15° intervals in Fig. 202. A curve through p_1, p_2, p_3, etc., is the required statical curve.

The work involved in producing the cross curves is carried out by means of an integrator, which is a mechanical device capable of recording area, moment of area and 2nd moment of area. Its theory and *modus operandi* are clearly set out in Attwood's "Theoretical Naval Architecture". In these notes the method known as *calculation of GZ direct* is implied, and in this connection the position of the centre of gravity of the ship has to be assumed. The curves derived therefore are true only for the assumed position of G. The correction necessary when G occupies any other position is very easily made.

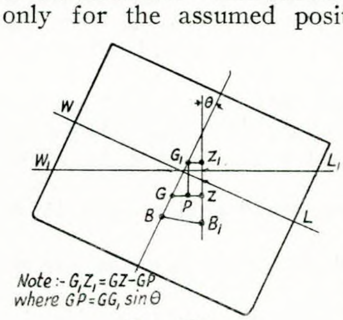


FIG. 203.

Suppose in Fig. 203 the correct position of the centre of gravity was at G_1 whereas in obtaining the cross curves it had been assumed at G. Then the true value of the righting lever would be G_1Z_1 and not GZ. The error is $GZ - G_1Z_1 = GG_1 \sin\theta$. This latter quantity should be deducted from each ordinate in the statical curve. An example will make this clear :—

Example 80.

In a certain vessel of 8,000 tons with its centre of gravity 24ft. above keel the values of GZ are given in the accompanying table. Plot the curve of statical stability, and show the effect of raising the centre of gravity 6in.

Angle in degrees θ	0°	15°	30°	45°	60°	75°	90°
GZ in feet ...	0	0·6	1·45	1·70	1·36	0·65	−0·2

Solution.

θ	GZ	$GG_1 \sin\theta = \cdot 5 \sin\theta$	$G_1Z_1 = GZ - GG_1 \sin\theta$
0°	0	0	0
15°	0·6	0·13	0·47
30°	1·45	0·25	1·2
45°	1·70	0·35	1·35
60°	1·36	0·43	0·93
75°	0·65	0·48	0·17
90°	−0·2	0·50	−0·70

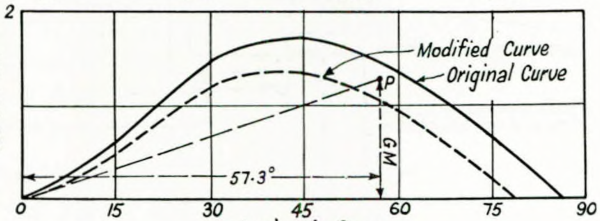


FIG. 204.

Reduction in *range* = 8° (approx.).

" " maximum $GZ = 0{\cdot}35$ft. (approx.).

Cross curves have the great advantage that they are comparatively easily determined, and from them it is possible to obtain with but little labour a statical curve for any desired condition of the ship.

The information which can be gleaned from a statical curve may be summarized as follows :—

(1) The range of stability.
(2) The value of the maximum lever, and the angle at which it occurs.
(3) The angle at which the deck edge submerges.
(4) A rough approximation to the metacentric height.
(5) The dynamical stability at any specified angle.

(1) and (2) are obvious, but the other points need some consideration.

(3) *The angle at which the deck edge submerges.*

At this angle, a little thought will perhaps indicate that although GZ is still increasing, the rate at which it is increasing is commencing to decrease. This is of course a definition of a *point of contraflexure.* If, therefore, the point of change of curvature is estimated from the diagram, the required angle can be determined. It occurs at about 15° in the example.

This is not necessarily the only nor the best method of estimating the angle at which the deck edge submerges.

(4) The equation to the tangent to the curve at the origin is $GZ = k\theta$.

It is known that for very small angles $GZ = GM.\theta$.

$\therefore\ k = GM$,

i.e., the slope of the curve at the origin is a measure of the metacentric height.

These facts may be usefully employed to enable the curve to be sketched in near the origin.

Thus :—Set off along the base line 1 radian (57·3°). Set up vertically a distance $= GM$, obtaining the point P (Fig. 204).

Join P with the origin, and sketch the curve in tangentially to it.

A rough estimate of GM might be obtained by reversing this process. It is not recommended however, if only because of the difficulty in drawing in the tangent accurately.

(5) *Dynamical Stability.*

Dynamical stability is a measure of the work which must be done in order to heel a vessel up to some specified angle.

It can be calculated from Moseley's formula, which is easy to derive but rather laborious to evaluate.

Moseley's formula states :—

Dynamical stability

$$= W\left\{\frac{v \times (gh + g_1h_1)}{V} - BG(1 - \cos\theta)\right\} \text{ ft. tons}$$

where W = displacement of ship.

$v(gh + g_1h_1)$ = vertical transference of the wedges [see Fig. 200]

V = underwater volume of the ship

BG = vertical distance between the centre of buoyancy and the centre of gravity

θ = angle of heel.

The simplest method of computing dynamical stability will be understood from the following investigation.

Consider the vessel to be heeled over to an angle θ, and at that angle let the moment of statical stability be $W \times GZ$ tons feet. Then the work done in heeling the vessel from θ to $(\theta + d\theta)$ is $W \times GZd\theta$ ft. tons, and the work done in heeling from 0 to ψ

$$= \int_0^{\psi} W.GZ.d\theta \text{ ft. tons.}$$

Now consider a vertical strip of the statical stability curve (assuming the ordinates to represent $W \times GZ$ and not GZ), see Fig. 205.

The area of this strip $= W \times GZ \times d\theta$, and $\therefore$ the total area under the curve up to some angle ψ

$$= \int_0^{\psi} W \times GZ \times d\theta$$

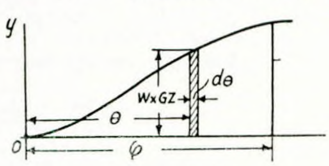


FIG. 205.

which is the same as the expression for dynamical stability.

Whence:—

The dynamical stability up to any specified angle ψ is equal to the area under the moment of statical stability curve up to the same angle.

Example 81.

Find the dynamical stability of the vessel in example 80 up to 60° of inclination. We may use the values of GZ at 15° intervals as Simpson's ordinates, but it will be necessary to multiply the result by the displacement (8,000 tons) to correct for the fact that the ordinates of the curve should be moments of stability, not just levers.

θ	Ordinates GZ	S.M.	Product.
0°	0	1	0
15°	0·6	4	2·40
30°	1·45	2	2·90
45°	1·70	4	6·80
60°	1·36	1	1·36
			13·46

Dynamical stability $= 8{,}000 \times \frac{1}{3} \times 15^{\circ} \times 13{\cdot}46$

[Note θ must be in circular measure.]

$$=8{,}000 \times \tfrac{1}{3} \times \frac{\pi}{12} \times 13{\cdot}46$$

$$=9{,}396 \text{ ft. tons.}$$

When θ is small the following approximate formulæ may prove useful for calculating dynamical stability.

As already stated the dynamical stability at any angle ψ

$$= \int_0^{\psi} W.GZd\theta$$

If it is true to say $GZ = GM \sin\theta$

$$\text{then dynamical stability} = \int_0^{\psi} W.GM \sin\theta d\theta$$

$$= W.GM\Big[-\cos\theta\Big]_0^{\psi}$$

$$= W.GM(1-\cos\psi) \text{ ft. tons.}$$

Aliter:—Let $GZ = GM.\theta$ [θ in circular measure]

$$\text{then dynamical stability} = \int_0^{\psi} W.GM.\theta d\theta$$

$$= W.GM\left[\frac{\theta^2}{2}\right]_0^{\psi}$$

$$= \frac{W.GM\psi^2}{2} \text{ ft. tons.}$$

Example 82.

Two exactly similar ships 5,000 tons displacement have a metacentric height of 2ft.

(a) is fitted with bilge keels;

(b) has no bilge keels.

Each is heeled to 10° and released.

(a) Goes to 6° in the opposite direction;

(b) „ „ 8° „ „ „

Find the work absorbed by the bilge keels.

(a) Dynamical stability at $10^\circ = \dfrac{5{,}000 \times 2}{2} \times (10^\circ)^2$

„ „ „ $6^\circ = \dfrac{5{,}000 \times 2}{2} \times (6^\circ)^2$

Difference equals work absorbed in overcoming resistances (including that of the bilge keels)

$$= 5{,}000 \left\{(10^\circ)^2 - (6^\circ)^2\right\} \text{ ft. tons.}$$

(b) By similar reasoning, work absorbed in overcoming resistances (excluding bilge keels)

$$= 5{,}000 \left\{(10^\circ)^2 - (8^\circ)^2\right\} \text{ ft. tons.}$$

$\therefore$ the difference of these results gives the work absorbed by the keels

$$= 5{,}000 \left\{(8^\circ)^2 - (6^\circ)^2\right\}$$

$$= 5{,}000 \left\{ (8+6)(8-6) \right\} \times \frac{1}{57{\cdot}3^2}$$

$$= 42{\cdot}6 \text{ ft. tons.}$$

Wall-sided Stability.

If the sides of a ship may be regarded as vertical in the region of the waterline, an exact expression for the moment of statical stability can be derived from Atwood's formula.

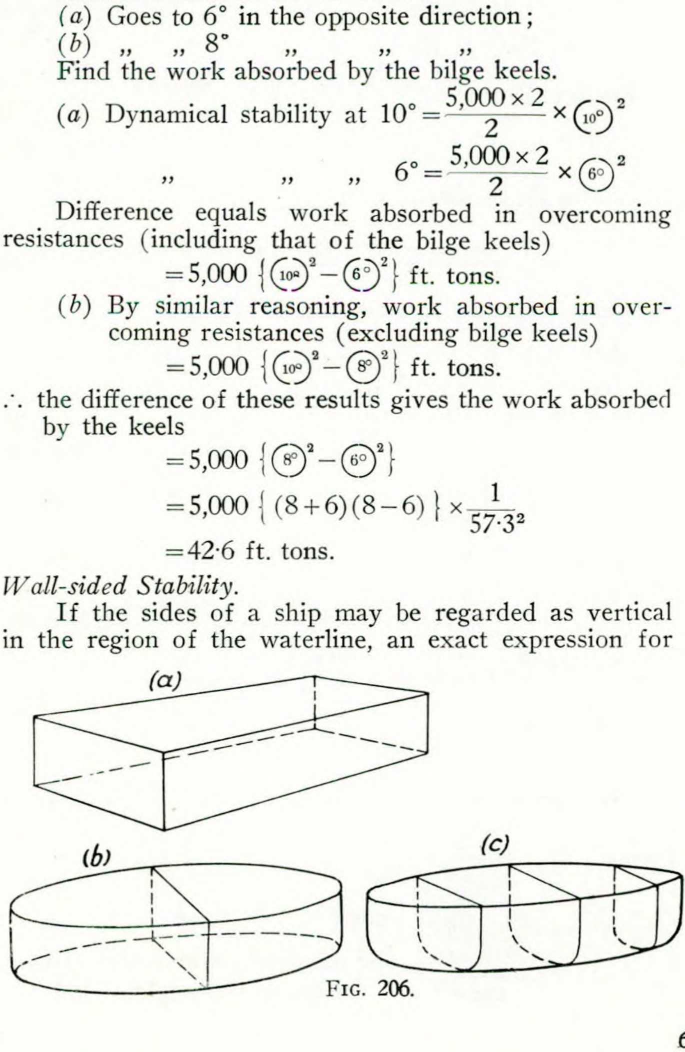


FIG. 206.

Fig. 206 (a), (b), (c) conform exactly to this requirement, from which it is evident that a full-lined cargo steamer with a block coefficient approaching 0·8 will not differ very much from the condition postulated.

However, the real value of the results to be deduced is to provide a method from which the angle of heel, arising from a shift of weight or otherwise, can be estimated in vessels with zero or negative metacentric height.

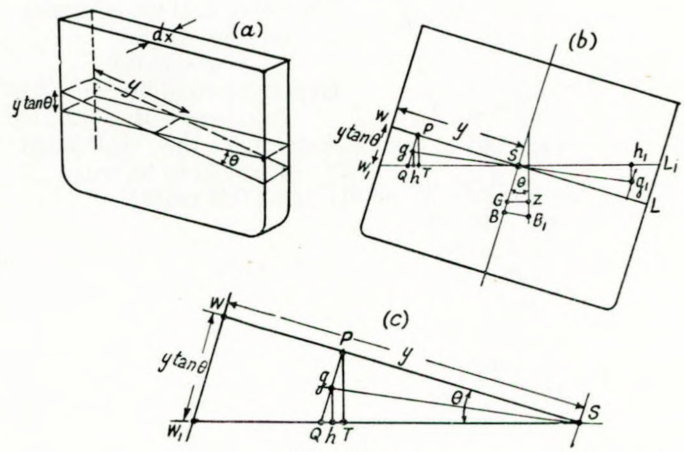


FIG. 207.

Fig. 207 (a) is a pictorial view of a length dx of the ship portraying what is meant by an elementary wedge.

Fig. 207 (b) is a section of ship at this position.

Fig. 207 (c) is an enlargement of the wedge.

(It is worth while to draw attention to the importance of an accurate figure in carrying out an investigation of this type).

The volume of the elementary wedge

$$= \frac{y \times y \tan\theta \times dx}{2} = \frac{y^2 \tan\theta\, dx}{2}$$

$Sh = ST + Th$

and $ST = SP\cos\theta$, $Th = Pg\sin\theta$

Now $SP = \frac{2}{3}WS = \frac{2}{3}y$ [from the properties of a triangle]

and $Pg = \frac{1}{3}y\tan\theta$ [„ „ „]

Whence $Sh = \frac{2}{3}y\cos\theta + \frac{1}{3}y\tan\theta\sin\theta$

$$hh_1 = 2Sh = \tfrac{2}{3}y[2\cos\theta + \tan\theta\sin\theta]$$

$\therefore$ Volume of elementary wedge $\times\, hh_1$

$$= \tfrac{1}{2}y^2\tan\theta\, dx \times \tfrac{2}{3}y(2\cos\theta + \tan\theta\sin\theta)$$

$$= \tfrac{2}{3}y^3dx(\sin\theta + \tfrac{1}{2}\tan^2\theta\sin\theta)$$

and the total moment of transference of the wedges

$$= v \times hh_1 = \int \tfrac{2}{3}y^3dx(\sin\theta + \tfrac{1}{2}\tan^2\theta\sin\theta)$$

between the appropriate limits.

But as previously proved

$\int \frac{2}{3}y^3dx = I_{CL}$ (moment of inertia of waterplane about C.L.)

$$\therefore\ v \times hh_1 = I_{CL}\ (\sin\theta + \tfrac{1}{2}\tan^2\theta\sin\theta)$$

$$\text{and } \frac{v \times hh_1}{V} = \frac{I_{CL}}{V}\sin\theta\left(1 + \frac{\tan^2\theta}{2}\right).$$

Since, however, $BM = \dfrac{I_{CL}}{V}$

$$\frac{v \times hh_1}{V} = BM\sin\theta\left(1 + \frac{\tan^2\theta}{2}\right)$$

From Atwood's formula $GZ = \left(\dfrac{v \times hh_1}{V} - BG\sin\theta\right)$

whence $GZ = \left[BM\sin\theta\left(1 + \dfrac{\tan^2\theta}{2}\right) - BG\sin\theta\right]$

$$=\left[(BM-BG)\sin\theta+\frac{BM\tan^2\theta\sin\theta}{2}\right]$$

$$=\sin\theta\left[GM+\frac{BM\tan^2\theta}{2}\right]$$

This is known as the *wall-sided formula* for GZ.

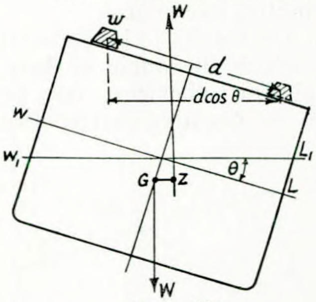


FIG. 208.

To find the angle of heel produced when a weight w is moved a distance d across the deck of a ship in which GM is zero.

The horizontal transference of the weight

$$=w\times d\cos\theta$$

Hence for equilibrium

$$w\times d\cos\theta=W\times GZ$$

Assuming the wall-sided formula to be true,

$$w\times d\cos\theta=W\sin\theta[GM+\tfrac{1}{2}BM\tan^2\theta]$$

put $GM=0$

$$\text{then}\quad\frac{w\times d\cos\theta}{W}=\sin\theta\times\tfrac{1}{2}BM\tan^2\theta$$

$$\text{or}\quad\frac{2wd}{W\times BM}=\tan^3\theta$$

$$\tan\theta=\sqrt[3]{\frac{2wd}{W\times BM}}$$

Example 83.

A vessel of 8,000 tons displacement has zero metacentric height. It is known that the distance between the centre of buoyancy and the transverse metacentre is 11ft. Estimate the angle of heel if 20 tons be moved 30ft. across the deck.

$$\tan\theta=\sqrt[3]{\frac{2\times w\times d}{W\times BM}}$$

$$=\sqrt[3]{\frac{2\times20\times30}{8{,}000\times11}}=\sqrt[3]{\frac{3}{220}}=0{\cdot}239$$

$$\theta=13\tfrac{1}{2}^\circ\text{ nearly.}$$

Negative Metacentric Height.

Assuming the wall-sided formula

$$GZ=\sin\theta\,(GM+\tfrac{1}{2}BM\tan^2\theta)$$

for equilibrium GZ must equal 0.

$\therefore\ \sin\theta=0$, $\theta=0$, and the ship is upright. This is the state of affairs when GM has a positive value. However the equation $0=\sin\theta(GM+\frac{1}{2}BM\tan^2\theta)$ can also be satisfied by $GM+\frac{1}{2}BM\tan^2\theta=0$.

$$\text{Whence }\tan\theta=\pm\sqrt{\frac{-2GM}{BM}}$$

Now this is imaginary unless the quantity under the root sign can be made positive. This in turn necessitates either GM or BM being negative. $BM=\frac{I}{V}$ and it is impossible for either I or V to be negative. Hence BM cannot be negative. GM may however have a negative value, and when this is so the expression can be written

$\tan\theta=\pm\sqrt{\frac{2GM}{BM}}$, which is real.

The value of θ obtained from this formula is known as the *angle of loll,* and it should be observed that the $\pm$ signs mean that a vessel with a negative metacentric height may "flop over" either to port or to starboard.

The expression also affirms the following truth :—

A vessel with a negative metacentric height cannot remain upright. It *must* fall over to its *angle of loll* which may be either port or starboard.

At the angle of loll the vessel has a positive metacentric height.

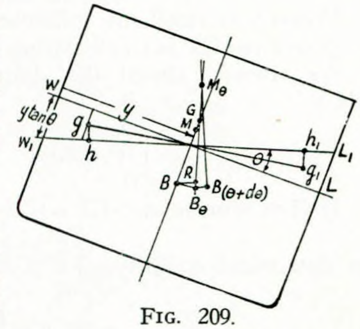


FIG. 209.

If y = any half ordinate of the plane when the ship is upright

y_θ = the corresponding half ordinate when the ship is inclined

$$\text{then }y_\theta=\frac{y}{\cos\theta}$$

The moment of inertia of an elementary strip about the centre line for the inclined

$$\text{position}=\tfrac{1}{3}\frac{y^3}{\cos^3\theta}\cdot dx$$

and for the whole plane MI_{CL} for inclined position

$$=\tfrac{2}{3}\int y^3dx\times\frac{1}{\cos^3\theta}$$

For the upright position $MI=\frac{2}{3}\int y^3dx=$ say I

also ,, ,, ,, $BM=\dfrac{I}{V}$

$$\text{Hence }I_\theta=\frac{I}{\cos^3\theta}$$

$$\text{and }B_\theta M_\theta=\frac{I_\theta}{V}=\frac{I}{V}\times\frac{1}{\cos^3\theta}=\frac{BM}{\cos^3\theta}$$

The required metacentric height in the inclined position is GM_θ

Where $GM_\theta=B_\theta M_\theta-B_\theta G$

and $B_\theta G=B_\theta R+RG.$

$$\text{Further }B_\theta R=\frac{v\{gh+g_1h_1\}}{V}=\frac{1}{V}\int\frac{y^2\tan\theta\,dx}{2}\times\{2\times\tfrac{1}{2}\times\tfrac{2}{3}y\tan\theta\cos\theta\}$$

$$=\frac{1}{3V}\int y^3dx.\tan\theta\sin\theta=\frac{\frac{2}{3}\int y^3dx}{V}\quad\frac{\tan\theta\sin\theta}{2}$$

$$=\frac{I}{V}\times\frac{\tan\theta\sin\theta}{2}=\frac{BM\tan\theta\sin\theta}{2}$$

Also $RG=BG\cos\theta=(BM+GM)\cos\theta$

$$\text{Whence }B_\theta G=\frac{BM\tan\theta\sin\theta}{2}+(BM+GM)\cos\theta$$

$$GM_\theta=B_\theta M_\theta-B_\theta G=\frac{BM}{\cos^3\theta}-\frac{BM\tan\theta\sin\theta}{2}-BM\cos\theta-GM\cos\theta$$

$$=BM\left(\frac{1}{\cos^3\theta}-\tfrac{1}{2}\tan\theta\sin\theta-\cos\theta\right)-GM\cos\theta$$

$$=\frac{BM}{\cos^3\theta}(1-\tfrac{1}{2}\sin^2\theta\cos^2\theta-\cos^4\theta)-GM\cos\theta$$

$$=\frac{BM}{\cos^3\theta}\left\{(1-\cos^2\theta)(1+\cos^2\theta)-\frac{\sin^2\theta\cos^2\theta}{2}\right\}-GM\cos\theta$$

$$=\frac{BM\sin^2\theta}{\cos^3\theta}\left\{1+\cos^2\theta-\frac{\cos^2\theta}{2}\right\}-GM\cos\theta$$

$$=\frac{BM\tan^2\theta}{2\cos\theta}\left\{2+\cos^2\theta\right\}-GM\cos\theta$$

But from formula for angle of loll $\tan^2\theta=\dfrac{2GM}{BM}$

$$\text{Hence }GM_\theta=\frac{BM}{\cos\theta}\times\frac{GM}{BM}\left\{2+\cos^2\theta\right\}-GM\cos\theta$$

$$=\frac{GM}{\cos\theta}\left\{2+\cos^2\theta\right\}-GM\cos\theta$$

$=\dfrac{2GM}{\cos\theta}$ the required positive metacentric height at the angle of loll.

If from angle of loll formula

$$\tan\theta=\frac{\sqrt{2GM}}{\sqrt{BM}}$$

then from Fig. 210

$$\frac{1}{\cos\theta}=\frac{\sqrt{BM+2GM}}{\sqrt{BM}}=\sqrt{1+\frac{2GM}{BM}}$$

and $\therefore\ GM_\theta=2GM\sqrt{1+\frac{2GM}{BM}}$

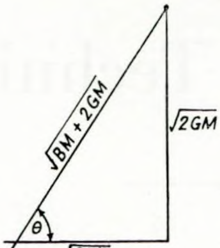


FIG. 210.

which is an alternative expression for the positive metacentric height at the angle of loll.

Example 84.

A vessel of 8,000 tons displacement has a negative metacentric height of 3in. The distance between the centre of buoyancy and the transverse metacentre is known to be 11ft. Approximate to her *angle of loll* and determine the positive metacentric height at that angle.

Let θ = angle of loll.

$$\tan\theta=\pm\sqrt{\frac{2GM}{BM}}=\pm\sqrt{\frac{2\times\frac{1}{4}}{11}}=0{\cdot}2132$$

$$\theta=12^\circ \text{ approx.}$$

$$GM_\theta=\frac{2\times 3''}{\cos 12^\circ}=\frac{2\times 3''}{0{\cdot}978}=6{\cdot}13\text{in. positive.}$$

Simpson's Rules.

The truth of these rules is based on the assumption that the bounding curve is parabolic.

1st Rule.

The equation of the curve is taken as $y=a_0+a_1x+a_2x^2$. 3 ordinates are chosen, *viz.*:— y_1, y_2 *and* y_3, and the common interval is h.

Then area $ABCD=$

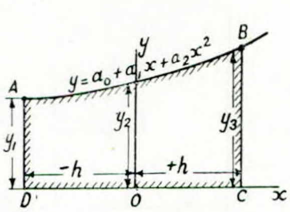


FIG. 211.

$$\int_{h}^{h} ydx=\int_{h}^{h}(a_0+a_1x+a_2x^2)dx$$

$$=\left[a_0x+a_1\frac{x^2}{2}+a_2\frac{x^3}{3}\right]_h^h=2a_0h+\tfrac{2}{3}a_2h^3$$

The constants a_0 and a_2 may be eliminated, and the result reduces to the form area $ABCD=\frac{1}{3}h(y_1+4y_2+y_3)$.

The completion of this case is left as an exercise for the reader, who may use the method set out in detail for the 2nd rule as a pattern.

2nd Rule.

This assumes the curve to be a parabola of the 3rd order whose equation is

$y=a_0+a_1x+a_2x^2+a_3x^3$.

4 ordinates, *viz.*, y_1, y_2, y_3 and y_4 must be used.

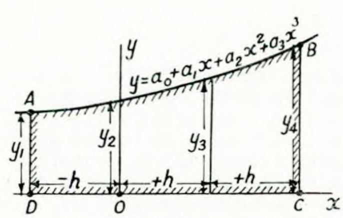


FIG. 212.

$$\text{Then area } ABCD=\int_{-h}^{2h} ydx$$

$$=\int_{-h}^{2h}(a_0+a_1x+a_2x^2+a_3x^3)dx$$

$$=\left[a_0x+a_1\frac{x^2}{2}+a_2\frac{x^3}{3}+a_3\frac{x^4}{4}\right]_{-h}^{2h}$$

$$=2a_0h+2a_1h^2+\tfrac{8}{3}a_2h^3+4a_3h^4+a_0h-a_1\frac{h^2}{2}+a_2\frac{h^3}{3}-a_3\frac{h^4}{4}$$

$$=3a_0h+\tfrac{3}{2}a_1h^2+3a_2h^3+\tfrac{15}{4}a_3h^4$$

The elimination of a_0 a_1 a_2 a_3:—

If $y=a_0+a_1x+a_2x^2+a_3x^3$

$$\begin{aligned}\text{when } x&=-h,\ y=y_1=a_0-a_1h+a_2h^2-a_3h^3 \quad\ldots\quad\ldots\ (1)\\ &=0,\ y=y_2=a_0 \quad\ldots\quad\ldots\quad\ldots\quad\ldots\quad\ldots\ (2)\\ &=+h,\ y=y_3=a_0+a_1h+a_2h^2+a_3h^3 \quad\ldots\quad\ldots\ (3)\\ &=+2h,\ y=y_4=a_0+2a_1h+4a_2h^2+8a_3h^3 \quad\ldots\ (4)\end{aligned}$$

Put y_2 for a_0 and add (1) and (3)

then $y_1+y_3=2y_2+2a_2h^2$

$$\therefore\ a_2=\frac{y_1+y_3-2y_2}{2h^2}\ \ldots\quad\ldots\quad\ldots\quad\ldots\quad\ldots\ (5)$$

Multiply (3) by 2 and then subtract (3) from (4)

$$y_4-2y_3=-a_0+2a_2h^2+6a_3h^3$$

$$=-y_2+y_1+y_3-2y_2+6a_3h^3$$

$$\therefore\ a_3=\frac{y_4-2y_3-y_1-y_3+3y_2}{6h^3}=\frac{y_4-3y_3-y_1+3y_2}{6h^3}\ \ldots\ (6)$$

Substitute these values of a_0, a_2, a_3 in (1)

$$y_1=y_2-a_1h+\frac{y_1+y_3-2y_2}{2}-\frac{y_4-3y_3-y_1+3y_2}{6}$$

$$\therefore\ a_1=\frac{1}{h}\left(y_2+\frac{y_1+y_3-2y_2}{2}-\frac{y_4-3y_3-y_1+3y_2}{6}-y_1\right)$$

$$=\frac{1}{6h}(6y_3-2y_1-3y_2-y_4)\quad\ldots\quad\ldots\quad\ldots\ (7)$$

$\therefore$ Area $ABCD$

$$=3hy_2+\frac{3h^2}{2}\times\frac{1}{6h}(6y_3-2y_1-3y_2-y_4)+\frac{3h^3}{2h^2}(y_1+y_3-2y_2)$$

$$+\frac{15h^4}{4}\times\frac{1}{6h^3}(y_4-3y_3-y_1+3y_2)$$

$$=h[3y_2+\tfrac{1}{4}(6y_3-2y_1-3y_2-y_4)+\tfrac{3}{2}(y_1+y_3-2y_2)$$
$$+\tfrac{5}{8}(y_4-3y_3-y_1+2y_2)]$$

$$=\frac{h}{8}[3y_1+9y_2+9y_3+3y_4]$$

$$=\tfrac{3}{8}h[y_1+3y_2+3y_3+y_4]$$

The Mean of Means.

As already stated in the text, if the speed of a ship making a series of runs over the measured mile with and against the tide be V_1 V_2 V_3 V_4, the mean is obtained as follows:—

$$\begin{array}{lllll} V_1 & & & & \\ & \frac{V_1+V_2}{2} & & & \\ V_2 & & \frac{V_1+2V_2+V_3}{4} & & \\ & \frac{V_2+V_3}{2} & & \frac{V_1+3V_2+3V_3+V_4}{8} & \\ V_3 & & \frac{V_2+2V_3+V_4}{4} & & \\ & \frac{V_3+V_4}{2} & & & \\ V_4 & & & & \end{array}$$

If $V_1=V+v_1$ where v_1 is speed of tide during initial run

$V_2=V-v_2$ where v_2 is speed of tide during 2nd run (after time t)

$V_3=V+v_3$ where v_3 is speed of tide during 3rd run (after time $2t$)

$V_4=V-v_4$ where v_4 is speed of tide during 4th run (after time $3t$)

Then V is the true speed independent of tidal effect

$$\text{and } V=\frac{V+v_1+3(V-v_2)+3(V+v_3)+V-v_4}{8}$$

$$=\frac{8V+v_1+3v_3-3v_2-v_4}{8}=V+\frac{v_1+3v_3-3v_2-v_4}{8}$$

It is necessary therefore to show that $v_1+3v_3-3v_2-v_4=0$

Suppose the speed of the tide to follow a parabolic law, *i.e.*, if v = speed after time t, $v=a_0+a_1t+a_2t^2$

$$\begin{aligned}\text{then when } t&=o & v&=v_1=a_0\\ t&=t & v&=v_2=a_0+a_1t+a_2t^2\\ t&=2t & v&=v_3=a_0+2a_1t+4a_2t^2\\ t&=3t & v&=v_4=a_0+3a_1t+9a_2t^2\end{aligned}$$

$\therefore\ (v_1+3v_3-3v_2-v_4)$

$$=a_0+3(a_0+2a_1t+4a_2t^2)-3(a_0+a_1t+a_2t^2)-(a_0+3a_1t+9a_2t^2)$$

$$=a_0+3a_0+6a_1t+12a_2t^2-3a_0-3a_1t-3a_2t^2-a_0-3a_1t-9a_2t^2$$

$$=0$$

Hence required mean speed is V obtained as above.

Abstracts of the Technical Press

Some Recent German Patents.

The present trend of German technical development in marine engineering is indicated by the nature of the patents which have been published in Germany during the past few months. The following newly patented inventions were described in recent issues of the technical periodical *Schiffbau*:—"Means of preventing oil leakage from shaft bearings on board ship". This invention is claimed to provide for the oil-tightness of propeller-shaft bearings in the manner shown in Fig. 1. The bore (*b, b*) of the ends of the housing (*a*) of the bearing are fitted with flanged rings (c, c^1). The width of these rings is such that the free edges engage either in an annular groove in the bearing shell (*e*), or in the bearing (*f*), or in a groove (d^1) in the wall (*i*) of an oil bath; sufficient clearance being provided for the edges of the rings to ensure that the self-adjusting movement of the bearing is unhindered. The rings (c, c^1) are also provided with grooves to fit packing of the normal type. A groove in the bearing shell (*e*) and a drain hole (*g*) connecting with it enable the oil to return to the sump before it can reach the spaces enclosed by the rings (c, c^1). These rings are made in halves to facilitate fitting and renewal of the packing material. "Improvements in cylindrical boilers with two or more furnaces". A boiler embodying these improvements is illustrated in the accompanying diagram (Fig. 2), which shows the disposition of a number of longitudinal drums in a multi-furnace boiler of the cylindrical type. At least two such drums (d, d^1) are provided in the upper part of the boiler for each lower drum (*e*), and are connected with these by water-tube walls (*f, f*) which form combustion chambers behind each furnace; the effect being the equivalent of a watertube boiler behind a cylindrical boiler, somewhat after the manner of the Prudhon-Capus design. Superheater sections of tubes (*g, g, g*) are arranged between the water walls of the combustion chambers, the bottom ends of the superheater elements being located just above the level of the arch (*i*) in each combustion chamber. The purpose of these arches is to deflect the combustion gases to the superheaters, the actual height of which may be varied to suit the superheat temperature required. The upper drums (d, d^1) are not connected to the water space of the cylindrical boiler; the steam generated in them passes into the steam space of the latter. By spacing the upper drums (d, d^1) far enough apart, ample room is available for the withdrawal of the superheater elements for dismantling, if necessary, and as these upper drums are not in the high-temperature region, there is an adequate and

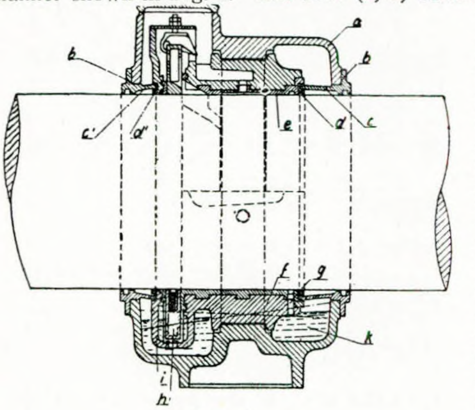


FIG. 1.

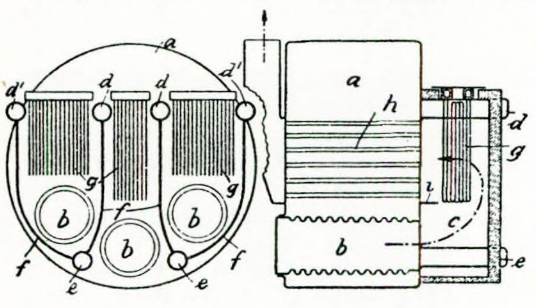


FIG. 2.

unhampered circulation of the water in the watertube portion of the boiler. "Safety devices for marine turbines, especially for those using high-pressure superheated steam". This patent, which is owned by Brown, Boveri and Co., covers an invention for overcoming the trouble that is sometimes experienced with marine propulsion turbines due to overheating of the astern blades when running ahead and of the ahead blades when running astern. This may be the result of an insufficient vacuum in the condenser, and in some installations there are automatic devices which shut steam off the turbine if the absolute pressure in the condenser exceeds a predetermined figure. Overheating of the turbine blades may, however, also be caused by steam leaking through the ahead or astern manœuvring valves when these are closed, in which case this steam impinges directly on blades set for rotation in the opposite direction, thereby increasing the windage loss. Under such circumstances the temperature of the blades may become much higher than that of the steam leaking through the valve, and where the working steam conditions are already near the safe limit, overheating is very liable to occur. The new invention provides for the fitting of automatic shut-off valves to both the ahead and astern turbines, so arranged that the astern shut-off valve is operative during ahead running and the ahead shut-off valve during astern running. The lay-out is shown diagrammatically in Fig. 3. For ahead running the H.P. and L.P.

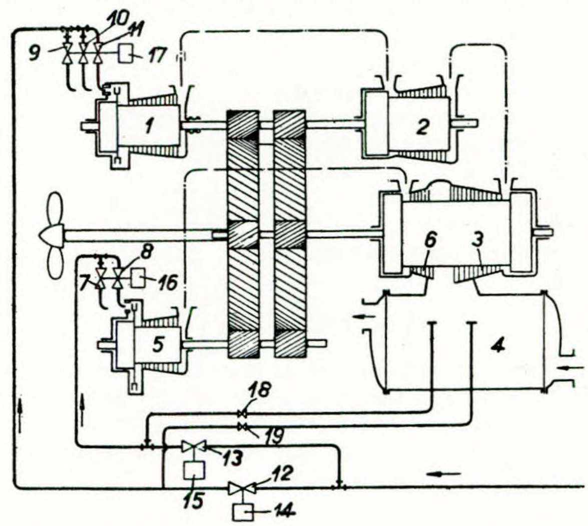


FIG. 3.

astern turbines (*5, 6*) idle with the astern nozzle valves (*7, 8*) closed and the ahead nozzle valves (*9, 10, 11*) partially or fully open. During astern running the H.P., M.P., and L.P. ahead turbines (*1, 2, 3*) are idling with the ahead nozzle valves (*9, 10, 11*) closed and the astern valves (*7, 8*) partially or fully open. Both the ahead and astern turbines are fitted with quick-shut-off valves (*12, 13*) which are operated by controls (*14, 15*) connected by gearing (*16, 17*) to the nozzle valves in such a manner that the astern shut-off valve (*13*) is closed during ahead running and the ahead shut-off valve (*14*) is closed when the turbines are running astern. Release valves (*18, 19*) are provided to allow any steam which may have leaked into the pipes between the shut-off valves and the nozzle valves to escape to the condenser. These release valves may, if desired, be arranged to operate in conjunction with the shut-off valves, so that the release valve (*18*) opens when the shut-off valve (*13*) closes, *i.e.*, during ahead running, and the valve (*19*) opens when the valve (*12*) is closed, *i.e.*, for astern running. It is claimed that with this arrangement a leaky shut-off valve will not build up a steam pressure in the inlet of a turbine which is idling. "Improvements in watertube boilers with two or more drums". The object of this invention is to improve superheater accessibility in high-powered installations where there is little room between adjacent boilers and between the boilers and bulkheads. Referring to the accompanying diagrams (Fig. 4), the riser tubes (*1*) and the downcomer tubes (*2*) are arranged arc-wise, whilst the superheater (*3*) is divided in the centre, and special collector boxes (*4, 4'*) are provided at the outer ends of each section. As the thickness of the nests of both the riser and downcomer tubes is the same, the superheater elements are uniformly protected from the effects of the hot gases throughout their length. "Improvements in geared Diesel propelling machinery". In geared marine installations with oil-pressure-operated couplings, the oil is supplied from a pump driven by the main engine. This pump also provides the oil pressure for the hydraulic brake mechanism which prevents the driven shaft from turning when the engine is stopped, for which reason it is necessary to provide a separate and alternative drive (or a separately-driven pump) to maintain the oil pressure while the main engine is stopped. The invention, the patent for which is owned by the Klöckner-Humboldt-Deutz A.G., avoids this need by employing engine starting air instead of oil pressure for operating the hydraulic brake mechanism. In the diagram (Fig. 5) the main engine (*1*) drives the shaft (*2*) through a reverse gear, and when the vessel is going ahead the coupling (*3*) is disconnected and the power is transmitted through the couplings (*4* and *5*). When running astern the coupling (*5*) is disconnected and the power to the shaft (*2*) is then transmitted through the couplings (*4* and *3*). The brake (*6*) holds the driven shaft stationary if desired, when the couplings (*5* and *3*) are disconnected and the engine is either running or stopped. The shaft (*7*) drives the oil pump (*11*) through bevel gears (*8, 9, 10*), and this pump draws oil from the tank (*12*) and discharges it through the pipe (*13*) to the couplings (*5* and *3*) and to the brake (*6*). The oil returns to the tank (*12*) through the pipes (*14, 15* and *16*). The valves (*17, 18* and *19*) regulate the oil supply to the couplings (*5* and *3*) and the brake (*6*) and also by-pass the oil through the pipes (*20, 21* and *22*) when the couplings and brake are not under pressure. The installation also comprises a pressure-regulating valve (*23*), a non-return valve (*24*), a pipe (*27*) connecting the air receiver (*26*) to the pipe (*25*) coming from the brake, a pressure-reducing valve (*29*) and an oil reservoir (*28*). As may be seen from the diagram, when the engine is stopped and the pump (*11*) is inoperative, the air pressure in the oil reservoir (*28*) produces the oil pressure required to hold the brake.—*"The Marine Engineer", Vol. 65, No. 776, March, 1942, pp. 59-61.*

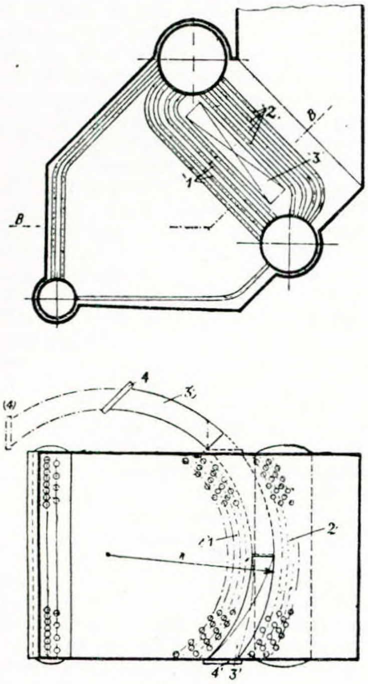


FIG. 4.

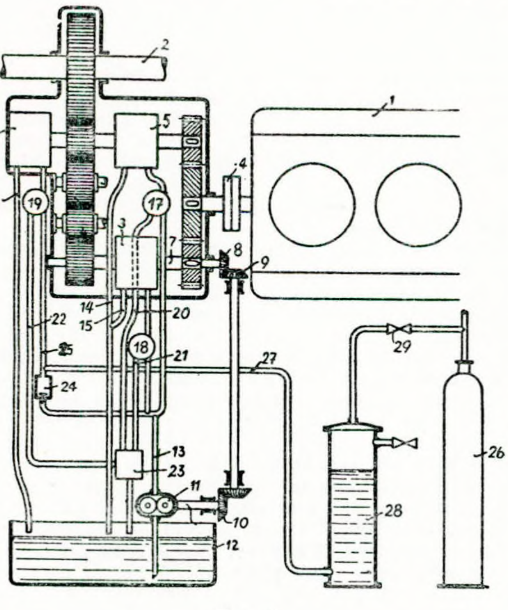


FIG. 5.

Hand-operated Oil-fuel Pumps in American Warships.

U.S. naval vessels are now equipped with a new type of hand-operated positive-displacement rotary pump, known as the Geroto pump. It may be either geared or directly driven, and the pumping element is unusual in that it consists of two internal lobar-toothed rotors which do not require the usual crescent of an internal-gear type pump. The inner element (or Gerotor) has one less tooth than the outer and its tooth form is generated from that of the outer rotor as both Gerotors turn in the same direction, providing continuous fluid-tight engagement and enabling the pump to maintain a high suction vacuum with a high discharge pressure. The output of the smallest Gerotor pump (size NHP-1·7) varies according to the viscosity of the oil being handled, but is normally about 1 gall./min., with a suction vacuum of 19½in. of mercury and a discharge pressure of 300lb./in.^{2}.—*"Marine Engineering and Shipping Review", Vol. XVII, No. 1, January, 1942, p. 111.*

U.S. Tanker Construction.

Two single-screw tank steamers, the "Esso Albany" and "Esso Trenton", were recently completed by the Sun Shipbuilding and Dry Dock Company for the Standard Oil Company of New Jersey. They are vessels of 16,400 tons d.w. capacity, constructed on the longitudinal-framing system, with flared bows and cruiser sterns. The propelling machinery consists of D.R. geared turbines normally developing 9,000 s.h.p. when supplied with steam at 435lb./in.^{2} pressure and 130° F. total temperature by two 2-drum express-type watertube boilers, each of 5,250ft.^{2} heating surface and 1,347ft.^{2} economiser heating surface, fitted with superheaters and desuperheaters. Electric current for driving the pumps and other auxiliaries is furnished by two 240-volt d.c. 400-kW. turbo-generators and one 150-kW. port service generator. The 123 single-screw turbo-electric tankers of the so-called T2-SE-A1 type building or on order for the U.S. Maritime Commission, are vessels of 10,750 gross tons and 18,000 tons d.w. capacity. The designed normal output of the propelling machinery is 6,000 s.h.p., and the propulsion motors are connected directly to the propeller shaft. The ships are due for completion towards the end of this year and in the early part of 1943, and will be turned over to various tanker companies for operation immediately they are ready for delivery.—*"Lloyd's List and Shipping Gazette", No. 39,719, 4th March, 1942, pp. 4 and 11.*

British Merchant Shipbuilding During the War.

Press representatives were recently permitted to inspect a large shipbuilding and marine engineering establishment on the North-East Coast, where a considerable number of naval and mercantile vessels are in various stages of construction. The ships visited included a completed single-screw steam tanker, a single-screw cargo steamer, and a large twin-screw passenger and refrigerated-cargo motorship. The oil-carrying steamship is of 8,129 gross tons and has a carrying capacity of about 12,000 tons. The propelling machinery, constructed by the firm responsible for the hull, consists of a set of triple-expansion engines having cylinders 26½, 44 and 73in. in diameter, with a 48-in. stroke. Steam at 220lb./in.^{2} pressure, without superheat, is supplied by three cylindrical boilers. The cargo steamship is of the so-called "Y" class, and has a gross tonnage of about 7,000 and a d.w. carrying capacity of about 10,000 tons. The hull is largely of welded construction. There are five cargo holds and the cargo-handling equipment comprises a 30-ton, two 10-ton and eight 5-ton derricks served by ten steam winches. The twin-screw passenger and refrigerated-cargo motorship is of 12,440 gross tons and is propelled by two sets of 5-cylr. Neptune-Doxford standard-type Diesel engines with a total output of 10,000 i.h.p. The refrigerating and deck machinery, as well as the E.R. auxiliaries, are electrically driven, current at 220 volts being furnished by four 8-cylr. Diesel-driven generators, each with a capacity of 245 kW. at 600 r.p.m. There is also an emergency Diesel-driven generator on the top deck. An oil-fired Cochran donkey boiler is installed for heating and domestic purposes, and for warming the main engines, in addition to which there is a Clarkson waste-heat boiler for use at sea. Accommodation is provided for 22 passengers, while the ship's officers and crew number 85. The decks of this ship are of welded construction, but they are riveted to the beams. Some of the butts of the shell plating are also welded, as are all the bulkheads and most of the superstructure.—*"The Shipbuilder", Vol. XLIX, No. 392, March, 1942, p. 65.*

Bearings for Centrifugal Pumps.

When possible, it is customary to employ the horizontal type of centrifugal pump in preference to the vertical. The vertical shaft type, while at first sight appearing to require a minimum of floor space, etc., may give trouble after installation owing to the difficulty of lubricating vertical bearings. If the pump is submerged, the problem of lubrication is further complicated and it usually results that the pump bearings run with water lubrication which often causes rapid wear. In some types of vertical pumps, the bearings are fitted with stuffing boxes with a view to excluding water. As these boxes may be under water, it is doubtful if they are ever packed after the pump has been first installed, with the result that water finds its way inside the bearing and floats out whatever lubricant was present. There are some automatic centrifugal oilers for vertical guide bearings which have a certain merit. They are, however, likely to clog with foreign matter, in which event the oil supply fails and the bearing runs hot. Then there are the thrust bearings to be considered. In addition to the thrust of the pump, the weight of the impellers and shafting must be carried by the thrust bearing. Here, again, the question of lubrication comes in, and it has been found quite difficult to design a thrust bearing that will operate without trouble at the high speed necessary for centrifugal pumps. Marine type multi-collar, roller, and ball bearings have all been used; the self-aligning ball type works out well under these conditions as it can be packed in grease which provides lubricant for a considerable period of time without further attention. Where there is a length of line shaft connecting the vertical centrifugal pump and its motor, it will be necessary to place several thrust bearings between the motor and the pump, installing a flexible coupling above each. This is to provide for a certain flexibility to take care of the misalignment that is sure to take place in an installation of this sort, all of which still further complicates matters. End thrust in centrifugal pumps is caused by two elements, the first of which is the effect of a static pressure acting upon unequal surfaces of the impellers and the second the sudden changes in direction of flow of the hydraulic currents entering and leaving the impeller. One or both of these causes may produce a tendency for the impeller and shaft to move in a lateral direction. In order to consider this more in detail, it is necessary to classify pumps roughly as those having single-inlet impellers and those fitted with impellers having two inlets. It will be understood that all the impellers are of the enclosed type as end thrust does not amount to very much in impellers of the open type, as in this type there are no enclosing walls and consequently the areas, upon which the pressure differences can take effect, are composed merely of the thickness of the vanes multiplied by the length. The single suction impeller is usually balanced by two devices. In the first system the impeller is fitted with wearing surfaces in front and rear and these are usually equipped with renewable wearing rings. The diameter of these surfaces is the same, so that the area of each wall of the impeller outside of these surfaces is equal. An equal pressure is supposed to exist upon each side of the impeller, so that, acting upon equal surfaces, there is no resultant movement either one way or the other. Leakage water across the forward sealing surfaces enters the suction opening of the impeller again and water leaking by the rear sealing surfaces collects in an annular space surrounding the hub of the impeller. If this space were entirely shut off from the entrance of the impeller, the discharge pressure would build up therein, creating a heavy end thrust towards the suction opening of the pump. To reduce this pressure, a number of holes are cored or drilled through the hub of the impeller so that this chamber in the rear communicates with the entrance to the impeller. By this means a pressure or vacuum exists in the balancing chamber equal to the pressure or the vacuum at the entrance of the impeller and a theoretical balance is obtained. As a matter of fact the change in the direction of the entering column of water creates an end thrust which is often of considerable extent. It cannot be determined just how great this will be or in what direction it will act and, therefore, a heavy-duty mechanical thrust bearing of the double-acting type is necessary in order to care for this unbalanced force. In the automatic balancing system the impellers are fitted with a single sealing surface around the inlet opening of the impeller. There is, therefore, a greater area on the rear wall of the impeller than upon the front wall which is exposed to the pressure generated by the impeller. This sets up a heavy end thrust toward the suction opening, and, to counteract this, water from the discharge side is allowed to flow through a comparatively long restricted passage finally to collect in what is termed a balancing chamber, which is formed on one side by a partition in the pump casing and on the other side by a disc which is attached to the shaft and rotates with it. The outside edge of this disc approaches contact with the stationary partition as it revolves, and thus a pressure is built up in the balancing chamber approximately equal to the unbalanced thrust produced by the impellers. The area of the balancing piston is so proportioned that when the discharge pressure of the pump acts upon it it will cause the shaft and impeller to move in the opposite direction from the end thrust that it is desired to balance. When this movement occurs, the balancing piston moves away from the stationary partition and allows part of the water in the balancing chamber to escape. Thus the pressure in the balancing chamber decreases and then the thrust of the pump draws the balancing piston back until it again closes the balancing chamber, and in turn the pressure again builds up in the balancing chamber and so on. This action is automatic. The small amount of water escaping through the balancing chamber may be discharged into the suction side of the pump, or into a stage having a lower pressure. Although the face of the balancing piston is not supposed to come in contact with the stationary partition, it is very likely to touch it momentarily at times, and on account of the fact that more or less grit is contained in most waters wear eventually takes place. In order to compensate for this wear, the piston should be faced off, or a new one installed. A renewable plate or ring is usually provided, attached to the partition in the casing against which the piston works. The balancing piston is therefore an internal automatic thrust bearing which cannot run hot because it is constantly cooled by the water flowing over it. It is not limited in speed, and can be operated at any speed at which it is safe to run the impellers. Although subject to wear, it is doubtful if the rate is any greater than that of the surfaces of the impellers, and the wearing rings. In double-suction impellers, the liquid is supposed to enter in equal volumes from both sides, and as the inlet openings are equal an equal vacuum or pressure exists on both inlet openings, and there is therefore no end thrust from this part of the impeller. The double impeller is also equipped with two sealing or wearing surfaces, which are equal in diameter and, as equal pressures act upon equal surfaces, there is no resultant end thrust so far as that portion of the impeller outside the sealing surfaces is concerned. Theoretically, this type of impeller is perfectly balanced. As a matter of fact, however, the inlet openings of the impeller are seldom exactly equal, and unequal wear upon the wearing surfaces may result in a higher pressure existing upon one side than upon the other. One or both of these causes usually sets up an end thrust in the double-suction impeller, and it is therefore necessary to equip this type with some type of mechanical thrust bearing to take care of thrust. As this is seldom very great, ordinary spacing collars are usually sufficient for this purpose, and this is the design adopted by most manufacturers at the present time. In large double-suction centrifugal pumps it is advisable to use a double-acting ball thrust bearing on account of the fact that it is impossible to determine in which direction the thrust will act.—*"The Journal of Commerce" (Shipbuilding and Engineering Edition), No. 35,604, 19th March, 1942, pp. 1-2.*

Cetane Number for Diesel Fuels.

A paper entitled "A Cetane Number Study of Diesel Fuel", was recently read at a meeting of the American Society of Automotive Engineers by Messrs. Ainsley, Young and Hamilton, of the American Sinclair Refining Company. The authors pointed out that some of the problems confronting the refiner are outside his control, and that his ability to meet in an economical manner a wide variety of Diesel fuel specifications, is therefore limited. If the Diesel engine user is to continue to enjoy low-cost fuel, and if the difficulties traceable to the wide variation in the fuels now supplied by different refiners are to be eliminated, it will be necessary to have a fuel which will satisfy the refiner as well as the consumer and which will meet the requirements of the Diesel engines in use at the present time. It is suggested that the "2D" Diesel fuel of the American Society for Testing Materials should be adopted as a universal fuel for high-speed Diesel engines. It has a minimum flash point of 140° F., a maximum pour point of 20° F., a maximum carbon residue figure of 0·20 per cent. with 1 per cent. of sulphur, and its viscosity (Saybolt seconds, universal) should be between 32·6 and 45·5. The minimum cetane number is 45.—*"The Motor Ship", Vol. XXII, No. 266, March, 1942, p. 392.*

Future Supply of Fuel Oil.

Disquieting statements have appeared in regard to the effect of the loss of the Netherlands East Indies oilfields on the supply of liquid fuel to this country, but it is, perhaps, not realised that the total petroleum production in the N.E.I. in 1941 was only 7,800,000 tons out of a total world output of nearly 300 million tons, so that the East Indies output was under 3 per cent. of the total. The oilfields of the U.S.A. were responsible for some 185 million tons, and in 1941 America produced about 9 million tons more oil than in 1940, a corresponding increase being anticipated during the present year. The normal annual increase of production in the U.S.A. is, therefore, greater than the total output of oil in the N.E.I., and the amount of oil fuel available for the Allies should be greater this year than in 1941, despite the loss of the East Indies oilfields. Iran produces more oil than the N.E.I.—rather more than 10 million tons per annum—and the U.S.S.R. has an output of nearly 30 million tons. Venezuela also produces about 27 million tons of petroleum per annum. It can be said, therefore, that according to recent figures, the Allied and Allied-controlled countries still have a daily oil production of about 800,000 tons, whilst the Axis and Axis-controlled countries, including synthetic production in Germany, have a daily production of little more than 41,000 tons, and the N.E.I. oilfields, before they fell into Japanese hands, had a daily output of 27,000 tons.—*"The Oil Engine", Vol. IX, No. 107, March, 1942, p. 273.*

Ventilation of Refrigerated Holds.

Store ventilation systems for refrigerated-cargo ships are designed somewhat in the manner shown in Fig. 1, from which it may

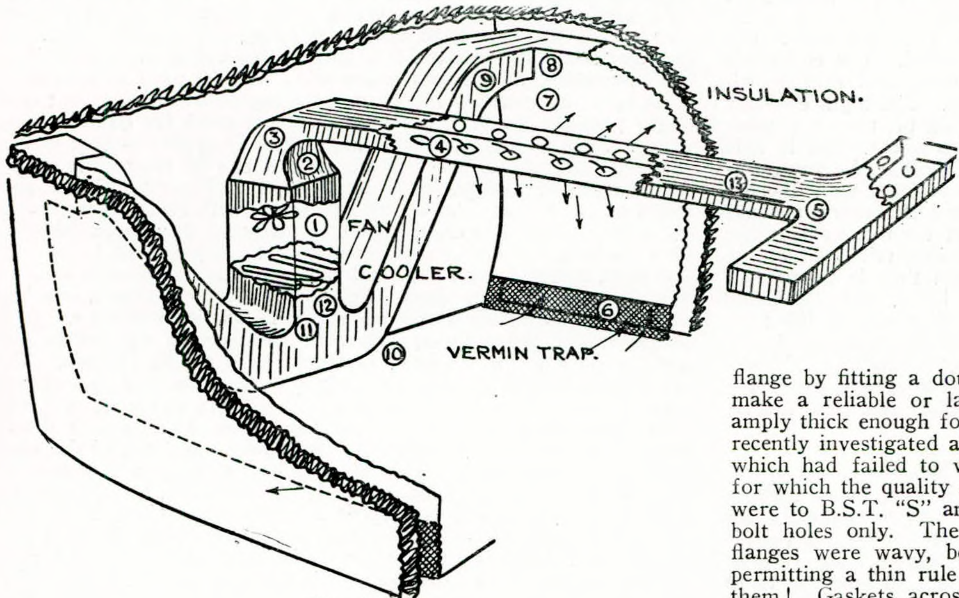


FIG. 1.

be seen that the cold air is admitted to the chamber through a rectangular duct near the ceiling of the compartment, and, after circulating around the free storage space, is discharged from positions very near the floor. The object of this arrangement is to force the air to flow in a direction against the normal convection currents which uncontrolled heating would produce and thereby to destroy any secondary circulating effects. In order to prevent the addition of heat by leakage the admission of warm air through cracks or by diffusion, it is necessary to maintain a sufficiently low temperature within the storage compartment to allow a small amount of cold air to be diffused outwards. This is achieved by creating a potential difference between the inside and outside of the store, thereby eliminating heat-addition by the infiltration of air. The air-supply fans for the compartment must be capable of producing a pressure-head within it, for which reason it is usual to employ fans of the centrifugal type. The propeller type of fan is used for "open" storage spaces which have no duct system, but rely upon the cooling effect of brine-circulating grids disposed round the inside walls, the function of the fan being limited to agitation of the cooled air within the compartment. But propeller type fans can only operate against low pressure-heads, and are unsuitable for the cold storage ventilation of large spaces. While it is essential that the ventilating fan should be capable of impressing a static pressure-head upon the air, it is equally important that the circulated air should encounter the least possible resistance to flow. The resistances to the flow of air through a cold storage space are made up of frictional resistances between the air and the duct walls, and by dynamic losses due to the various obstructions encountered in the path of the flowing air. Referring to the circuit shown in Fig. 1, the fan *(1)* drives the air into an overhead duct through an open bell-mouth *(2)*, a change of shape occurring at the right-angled bend *(3)*. Air is distributed in the compartment through the staggered holes *(4)* in the duct. The velocity of the air is high in that part of the duct which is nearest to the fan, for which reason the holes there are made small. Although the duct is tapered so as to give the air a reasonably uniform velocity, it is necessary to increase the size of the discharge holes as the distance from the fan increases. When the air enters the compartment its velocity falls to almost zero so that some energy is lost in discharging through the holes; this is generally allowed for in the discharge coefficient. At the point *(5)* at the end of the duct there is another right-angled bend and the air flow is divided, the discharge holes being arranged for the distribution of the air through this end piece. The air flow in the exhaust circuit of the compartment has also to overcome several points of resistance. At the base *(6)* of the exhaust ducts the air is contracted into the small area of the exhaust trunking where its velocity is increased, the dynamic loss incurred being proportional to the velocity-increase squared. At the point *(8)* where the air is collected from the compartment walls *(7)* and passed to the cooler space, there is a vortex effect as the air is collected from a wide shallow space and constrained to move into a short tunnel, in addition to which there is a loss of head due to bends. More bends increase the pressure drop at *(9)* and *(10)*, while at *(11)* the two flows are mixed before entering the cooler *(12)*. The air—and gas—in the exhaust trunk contains a larger percentage of CO_2 due to the respiration of the foodstuffs in the cold storage space, but the changes in density, specific heat and specific volume are too small to be of any practical significance.—*R. A. Collacott, B.Sc., "The Shipping World", Vol. CVI, No. 2,545, 25th March, 1942, pp. 231-233.*

Joints with Asbestos Gaskets.

No jointing material, however, good, can make an effective joint if the flange faces are not flat and in proper alignment. A lazy fitter may try to remedy an uneven flange by fitting a double thickness of jointing, but this will never make a reliable or lasting joint. Jointing $\frac{1}{32}$-in. thick should be amply thick enough for any flanges in proper condition. The writer recently investigated a case of defective joints in a new installation, which had failed to withstand a 30lb./in.^{2} water-pressure test and for which the quality of the jointing had been blamed. The flanges were to B.S.T. "S" and $\frac{1}{16}$-in. joint rings had been used inside the bolt holes only. The bolts had been tightened so much that the flanges were wavy, being almost in contact around the bolts and permitting a thin rule to be pushed into the pipe between some of them! Gaskets across the whole face of the flange should have been used in this case, and an ordinary standard-size spanner should have been employed to tighten the bolts. As regards the maximum temperature which compressed asbestos jointing will withstand, there are plenty of installations in this country in which good quality jointing is withstanding temperatures of 900° to 1,000° F., whilst in I.C. engines still higher local temperatures are met with. The process of calendering compressed-asbestos jointing sheets produces a definite "grain" in the material due to the tendency of the asbestos fibres to wrap themselves round the callender roller. A test piece cut with the grain therefore has a greater tensile strength than one cut across the grain, but by cementing two thin sheets together with their grains running at right angles to each other, a sheet can be produced from which test pieces cut in any direction will have the same tensile strength. However, it is doubtful whether this presents any advantage in practice, since a narrow ring cut from such a sheet will still break down at the weakest point—*i.e.,* against the grain—the only difference from an unlaminated sheet being that the break will extend across half the cross-sectional area of the ring instead of across the whole of it. This is especially the case at high temperatures, when the effectiveness of the cement joining the two sheets together quickly vanishes. Statements that a gasket may be "too thin to withstand the bolt pressure" are contrary to laboratory tests and actual practice, as good quality asbestos jointing containing a minimum of rubber binding will stand extremely high compressive loading, and narrow rings cut from thin material will stand a higher loading than those cut from thicker material. Modern practice for high-pressure joints favours the use of thin jointing only $\frac{1}{64}$in. or 0·008in. thick, and this is sometimes combined with the use of "gramophone finish" flanges, *i.e.,* flanges machined with a fine spiral groove so that their finish is similar in appearance to that of a gramophone record. The exact pitch and depth of the grooves is relatively unimportant, and it may be assumed that for all general purposes flanges require an ordinary machined finish only, the really important factors being that they should be heavy enough, flat, and properly lined up.—*W. E. Hoes, "The Marine Engineer", Vol. 65, No. 776, March, 1942, p. 57.*

Large Flying Boats.

The flying boat "Berwick" in which the Prime Minister recently crossed the Atlantic, is one of three Boeing type 314A aircraft purchased by the British Government for nearly a million pounds, and now operated by British Airways. These flying boats have each a loaded weight of 40 tons and normally carry 64 passengers and a crew of 11, in addition to 4,500 gallons of 100-octane petrol. There are four 1,600-h.p. Wright Cyclone engines each driving a Hamilton 3-blade variable-pitch hydraulic air screw. The Boeing 314A has a cruising speed of 140 m.p.h. for a range of over 4,000 miles. The body of this type of flying boat is somewhat larger than that of the Curtiss Wright C.W.20 or any other Empire flying boat. There are nine cabins with bunks for night use arranged in a manner similar to that in a railway sleeping car. The first cabin is used for the stowage of mails and cargo and contains sleeping accommodation for the crew. The second cabin contains the pantry and the men's dressing room, whilst the remainder are used by the passengers. The W/T installation includes a radio compass and an automatic bearing recorder. The main W/T transmitter is on the flight deck (above the cabin deck) and a fixed aerial is fitted. The engineer's control station is located aft of the W/T room and contains all the controls and instruments for the machinery. The flight engineer is responsible for starting the engines, regulating the air-screw pitch mechanism, the engine throttles and other equipment. He is able to inspect the engines, when necessary, through bulkhead doors opening into either wing. The 314A flying boats are fitted with de-icing equipment. They fly via Foynes or Lisbon and Baltimore, and are used solely for the conveyance of members of the Government and diplomatic mails. All the maintenance work is carried out at the American station. The tendency at the present time is to increase the size and weight of Transatlantic flying boats, and the Glenn Martin Company are reported to be constructing one of 100 tons. The Junkers Company are stated to be introducing a modified type of JU 52-3m 'plane in which the rear part of the fuselage is hinged and folds upwards, so that tanks and guns can be stowed inside for transport.—*L. C. Grant, "Shipping", Vol. XXX, No. 356, March, 1942, p. 12.*

High-altitude Power Plant.

The power developed by an aircraft engine drops with increasing height because of the lower air density, and to make up for that the engine is equipped with a blower which compresses the air and is usually driven direct by the engine. The power absorbed by the blower increases considerably with altitude, and amounts to as much as 30 per cent. of the useful engine power at 33,000ft. if the blower pressure is to be maintained at 1·3 atm. (20lb./in.^{2}) up to that height. The mechanical drive of a supercharger is therefore frequently uneconomical at great heights, and it is preferable to make use of the drop in pressure between the engine exhaust and the surrounding air, which increases with height, in order to maintain the power of the engine. The exhaust gases are passed through a gas turbine, and the latter, in turn, drives a blower. Air enters a three-stage supercharger at the speed of the aircraft, passes through an air cooler in which it gives up some of the heat generated during compression, and finally goes into the engine cylinders. From the engine cylinders the exhaust gases pass through an exhaust collector into the exhaust-gas turbine, in which they give up some of their energy; the remainder is utilised in the form of jet propulsion (ejector exhaust). The flow of air and gases through the entire installation may be regulated by varying the pressure of the air behind the blower, whence it is passed to a regulator which controls a relief valve in the exhaust-collector pipe. In this way only sufficient exhaust gas is passed to the turbine for the blower pressure desired. While data for the design of blowers were available from rotary compressor and pump practice, the construction of exhaust-gas turbines presented special difficulties. The steam turbine could, to a certain extent, be used as a model, but the temperatures in exhaust-gas turbines are about twice as great as those encountered in steam turbine practice, the exhaust gases sometimes leaving the engine cylinders at temperatures of 1,800° F. or more. It therefore became necessary to control these high temperatures by cooling the exhaust gases before they enter the turbine in order to ensure that no overheating of the materials employed in the construction of the latter can take place. This course was adopted in America 10 years ago by exposing the exhaust pipes on their way to the turbine to the cooling effect of an air stream, and by the provision of similar cooling arrangements for the turbine casing and even the turbine rotor itself. However, the reduction in temperature of the exhaust gases in front of the turbine reduces the temperature drop which it is desired to use in the turbine, and in order to produce the required blower pressure it becomes necessary to make up for the falling gas temperature in front of the turbine by an increase in the gas pressure. Such a rise of the pressure in the exhaust pipe has an adverse effect on the expansion of the burnt gases in the engine cylinders and reduces the amount of fresh air entering the cylinders before each power stroke. It is therefore clear that for a given engine there exists, in determining altitude and blower pressure over the required turbine output, a relationship between exhaust-gas temperature and filling of the engine, which can be estimated by calculation. In practice, it has been found that exhaust-gas temperatures below about 1,300° F. result in a great deterioration of engine output, while temperatures above 1,650° F. produce no material advantages in power output. It is therefore desirable to work with exhaust-gas temperatures between 1,300° and 1,650° F.—*G. Bock, "Flight", Vol. XLI, No. 1,735, 26th March, 1942, pp. 292-293.*

Modern Standards of Ships' Accommodation.

Substantial improvements in the extent and equipment of the accommodation for the officers and men in British merchant ships have been effected in recent years, more especially in oil tankers. In some of the new ships now completing, the engineer's office is directly accessible from the machinery space and is provided with a leather-covered, upholstered settee which can be used as a sleeping berth by an engineer off watch when for any reason he does not wish to (or possibly cannot, due to bad weather) reach his quarters elsewhere. Such an arrangement is not ordinarily applicable to a tanker, where the engineers' quarters are aft and can be reached from the engine room without going on deck. Another innovation is the provision of an engineers' auxiliary messroom, with an adjoining washroom, to enable engineers to take emergency meals without changing their clothing. There is now a tendency to provide one large well-appointed saloon for all deck and engineer officers. This is a sound scheme, said to have been originated by Dutch shipowners and adopted by them for large passenger liners, in particular. It is probable that a similar arrangement will become general in this country after the war when passenger liners are again built. The practice in British vessels carrying a large E.R. staff has always been to have separate saloons for deck officers and engineers, and it is sometimes found that a not too subtle distinction is drawn between the standard of furnishing (and, indeed perhaps, of size also) in the two apartments. This is also indicated by a difference in title, the deck officers' dining-room being labelled a saloon, whereas that of the engineers is known as a messroom. There is no need for such distinctions and both rooms should be equally large in proportion to the number of officers using each apartment, and most certainly equally well furnished. The quarters for the seamen, firemen and greasers in the new ships are far more comfortable than hitherto, and the forecastle is no longer used for housing the crew. Whereas the Merchant Shipping Act, 1906, required 12ft.^{2} per man in the sleeping cabins, the space allotted in the new tonnage is at least 32ft.^{2}, in addition to which it is becoming a general rule to put not more than two bunks in one cabin. The crew's messrooms are better furnished than formerly, while the furnishing and equipment of the sleeping cabins leaves nothing to be desired.—*"The Motor Ship", Vol. XXII, No. 266, March, 1942, p. 414.*

Auxiliary Oil Engines for Ships After the War.

Our two greatest needs after the war will presumably be the re-building of partially destroyed towns and cities, and the production of new ships to replace the tonnage which we have lost. It has been stated that in order to maintain her standard of life, Britain must have 20 million gross tons of shipping, and it is therefore probable that British shipyards and marine engineering works will be fully occupied for a long time to come. Although quite a considerable portion of our marine engineers build both propelling and stationary engines, it is suggested that they may then find that it will prove to be more profitable even for those who can produce small Diesel machinery on a reasonably economical basis, to purchase their auxiliary equipment from stationary oil-engine manufacturers who are able to build suitable engines on a much larger scale. Many of these firms produce standard ranges of engines which are eminently suitable for marine service of various kinds. Shipowners and shipbuilders should be more willing, after the war, to accept the standard high-speed relatively cheap Diesel engines of from 100 to 400 b.h.p., which are now available from many British stationary engine factories, despite their former prejudice against engines running at more than 700 r.p.m.—*"The Oil Engine", Vol. IX, No. 107, March, 1942, p. 273.*

The 15-knot Tramp.

It is reported that a well-known British shipbuilding concern has preparea plans for the construction of 14-knot to 15-knot tramps after the war, despite the fact that it is often taken for granted that a tramp, even in future years, will be designed for no higher speed than 11 knots. After the vessel has been in service for some time, such a speed, of course, represents only 8 to 9 knots at sea in average weather. It is not at all certain that shippers throughout the world will remain content with such speeds even for tramps. With modern Diesel machinery, especially if a multi-engine system be employed, so that one or two engines may be shut down when lower speeds are needed, the actual efficiency of the machinery is as high when operating at half power, say 11 knots, as when running full speed at 15 knots. This is a factor which somewhat changes the situation in regard to the speed of tramps, since before the advent of the oil engine a ship designed for 15 knots would have had a hopelessly low efficiency at 11 knots. A 9,000-ton 15-knot tramp would require machinery of about 5,000 b.h.p., and if at sea about 200 days per annum, the total expenditure on fuel, at £4 per ton, would be £17,000. An 11-knot tramp requiring machinery of 1,800 b.h.p. would have a fuel bill of £6,000, the difference thus being £11,000. This may seem to be a large difference, but as the slow ship would travel only about 52,000 miles and the faster vessel 72,000 miles, the income derived from the 15-knot tramp, at any rate in fairly good shipping years, would probably much more than counter-balance the additional fuel expenditure. Admittedly, somewhat higher interest and depreciation charges would be involved, but it is to be hoped that some British shipowners, at any rate, will not hesitate to take the risk of experimenting with a sufficient number of high-speed tramps, after the war, in order to ascertain in normal service whether they are better profit-earners than the slower ships. Although the fast tramp does not appeal to a large proportion of British shipowners at the present time, there is every reason to believe that circumstances are changing so rapidly, that their ideas on this subject will be greatly modified after the war.—*"The Motor Ship", Vol. XXII, No. 266, March, 1942, p. 390.*

Water-ballast Arrangements in New Tramps.

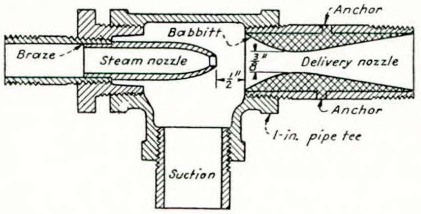
A good immersion for a cargo vessel is important because it is essential for the screw to be immersed to the greatest extent possible in order to obtain adequate propeller efficiency, since the latter falls off very rapidly as the blade tips break the surface of the water. At the same time the seaworthiness of the ship is improved if the freeboard is reduced in order to ensure that the area presented to the effect of the wind and waves is not excessive. Even in good weather a well-ballasted ship has an advantage over one riding high in the water, as may be judged by the case of two sister ships on a ballast voyage from the U.S.A. to the West Indies. One of these vessels had her deep tank filled and maintained a speed of about 10 knots, whereas the sister ship, in which this precaution had not been taken, averaged only just over 9 knots. In the North Atlantic the difference would be even more marked, and for this reason it was the practice, during the last war, to arrange that all ships should be loaded down to about 60 per cent. of their full-load draught before starting on a ballast voyage. A similar procedure is being followed at the present time, and it is common knowledge that the authorities' insistence on a close adherence to the letter of the instructions has sometimes given rise to a certain amount of dissatisfaction. In some of the newer ships a deep tank and large peak tanks for water ballast are usually available, so that it is rarely necessary to take in additional ballast in the form of rock or sand, but there is no universal rule in the matter, as some shipowners found that the additional complication resulting from the fitting of a deep tank was not justified. The Ministry of War Transport are now taking steps to provide extra ballast capacity in all new tramp steamers in the form of additional tanks at the tunnel sides in the after hold, abreast of the machinery space, and also in the forward hold. Not only does this disposition of water ballast give a good trim to the ship, but it also provides a system of ballasting which keeps the stresses on the main structure of the vessel as low as possible.—*"Fairplay", Vol. CLVIII, No. 3,069, 5th March, 1942, pp. 302 and 304.*

Engines for Ships' Lifeboats.

The Austin Motor Company state that in addition to the 600 petrol engines for ship's boats first ordered from them, a further contract for 1,000 similar engines has now been placed with the company. The first batch of engines were designed for direct drive to the propeller, but the later ones will employ geared drive. Apart from the slightly higher speed that will be obtained with this form of drive on account of improved propeller efficiency, its main advantage lies in the fact that it allows the lifeboat to get away from the ship's side without difficulty, whereas with direct drive there is a risk of the engine stalling at the critical moment. The engine is a four-cylinder water-cooled model of 900 c.c. capacity, rated at 10 b.h.p. at 1,450 r.p.m., but capable of developing up to about 17 b.h.p. at 2,500 r.p.m. A standard 26-ft. lifeboat carrying its full complement of 37 persons and fitted with one of the new geared-drive engines, recently completed highly satisfactory trials, a mean speed of 5½ knots being attained at 1,680 r.p.m. of the engine. The gear is incorporated with the reverse gear and adds very little to the length of the engine, the ratio of reduction being 1·8 to 1 with 20 teeth on the driving wheel and 36 on the driven wheel. The gearing is of the double helical type and the distance between the centres of the driving and driven shafts is only 2·8in. This particular lifeboat was built at Molesey in ten days and was one of 30 completed by the boatbuilders to the order of various shipping companies.—*"The Motor Boat", Vol. LXXV, No. 1,893, March, 1942, pp. 56-59.*

Steam Ejector from Pipe Fittings.

The accompanying illustration shows the construction of a steam ejector described by C. O. Hagen in *"Power"*. He states that it operates very satisfactorily on either steam or water. The central component may be a 1-in. pipe T-piece, into which the suction pipe, steam nozzle and delivery nozzle are screwed as shown. The steam nozzle, in this instance is made by heating and drawing down one end of a piece of ⅜-in. pipe which is then drilled axially to $\frac{3}{16}$-in. diameter, cut to length and brazed into a ½-in. nipple.



The delivery nozzle is made by pouring Babbitt metal between a 3-in. length of 1-in. pipe (or a 1-in.×3-in. nipple) and a suitable wooden core. Four holes drilled in this pipe or nipple, and wrapped outside with paper during pouring, provide keys for the cast nozzle. The wooden core should be turned in a lathe to the correct Venturi form, but if no lathe is available, good results can be obtained by hand-whittling. The core should project 2in. or so beyond one end of the pipe or nipple to provide a grip for removing the core by twisting and breaking it at the throat after the Babbitt has cooled. A notch cut along the side of the projecting core forms a channel for pouring the Babbitt. A coat of white lead on the core prevents it from smoking and forming blow-holes in the Babbitt. As cast, the throat should be slightly less than ⅜in. in diameter to allow for finishing by careful scraping. Smooth finish and accurate lining up are important and not difficult to secure.—*"The Power and Works Engineer", Vol. XXXVII, No. 430, April, 1942, p. 127.*

Boiler Scale and Fuel Wastage.

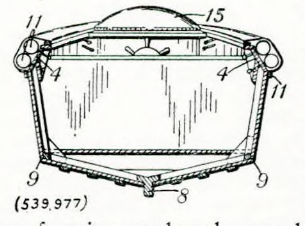
An article in *"Coal Heat"* gives the following data concerning loss in boiler efficiency due to scale or incrustation:—

Thickness of scale.	*Loss of efficiency.*	*Coal wasted per ton.*
$\frac{1}{32}$in.	7·2 per cent.	144lb.
$\frac{1}{20}$ „	11·1 „ „	222 „
$\frac{1}{16}$ „	12·4 „ „	248 „
$\frac{1}{8}$ „	15·9 „ „	318 „

—*"The Iron and Coal Trades Review", Vol. CXLIV, No. 3,865, 27th March, 1942, p. 288.*

Launch for Service with Flying Boats.

The British Overseas Airways Corporation have developed a new design for a launch intended for the conveyance of personnel and loads to and from flying boats. The design is claimed to give improved manœuvrability and good stability, so that the risk of damage to the hull of the flying boat is minimised, even in a choppy sea. The launch, the construction of which forms the subject of a British patent, has a bow and stern rounded in plan and connected by long flat curves that form the top edges of the gunwales. The maximum beam of the launch is at about 6 to 8in. from the



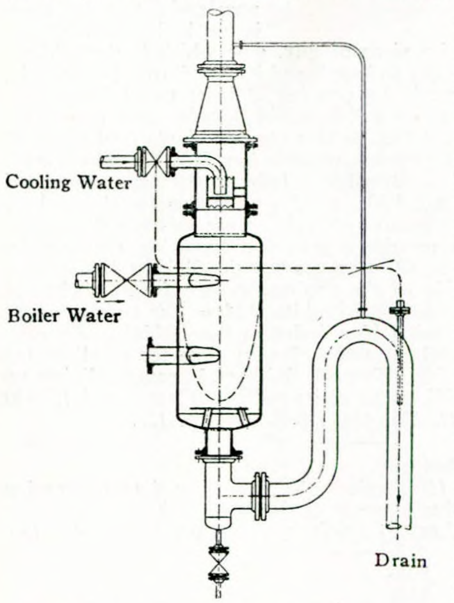
top, forming cambered gunwales (*4*) set at an angle of about 30° to the vertical amidships and throughout almost the entire length of the sides. Below the cambered gunwales the sides are set at a small angle of 7½° or 10° to the vertical. The bottom of the boat on each side of the keel (*8*) is flat amidships and inclined at an angle of approximately 20° to the horizontal, giving a chine (*9*) which extends from the lower edge of the gunwales at the bow to the lower edge at the stern, and which has an obtuse angle that increases as it approaches the gunwales at each end. The entire length of each gunwale is fitted with a longitudinal fender composed of two superimposed air tubes in a common casing attached to the top and bottom edges of the gunwales. The air tubes are in short sections, each with its own valve for inflation, and additional fenders (*15*) are secured to the top of the bow and stern decks. It is claimed that the form of the launch is such that it is practically impossible for any part other than the fenders to come into contact with the fragile hull of a flying boat.—*"Engineering", Vol. 153, No. 3,978, 10th April, March, 1942, p. 62.*

British Foster Wheeler Watertube Boilers.

It is known that a number of fast cargo vessels with geared turbines and watertube boilers are now under construction in various British yards. While some of these ships have Yarrow boilers and others Babcock and Wilcox sectional-header type boilers, it is interesting to note that in some cases a new British version of the Foster Wheeler watertube boiler has been adopted. This boiler is generally similar to the American-built "D" type Foster Wheeler boiler which is installed in many of the U.S. Maritime Commission's new cargo steamers.—*"The Marine Engineer", Vol. 65, No. 766, March, 1942, p. 62..*

Silent Blow-off Expansion Apparatus.

The accompanying illustration from *"Combustion"* shows the essential features of a boiler blow-down device which, according to *Archiv für Wärmewirtschaft und Dampfkesselwesen,* is silent in operation. The water blow down is directed tangentially into a cylindrical drum about 20in. in diameter and 40in. high, resulting in a vortex motion which facilitates the release of steam from the water. As it rises, the released steam comes into contact with cooling water in the upper part of the device, where it is thus condensed. The cooling water is admitted to a basin with notched edges to promote uniform overflow. This water and the condensed steam mix with the water in the lower part of the drum, forming a mixture at about 140° F., which is not injurious to the drain pipes. The cooling water admission is controlled by means of a thermostat depending on the temperature of the water running to the drain. It is stated that on trial a blow-down device of this type has dealt with 17,650lb./hr. of boiler water at a pressure of 356lb./in.² and temperature of 410° F.—*"The Power and Works Engineer", Vol. XXXVII, No. 430, April, 1942, p. 127.*



Speed Control of Electric Fans.

Magnetic means of varying the speed of induced-draught fans driven by constant-speed motors are being utilised in America. According to *"Power"*, a well-known electrical manufacturing concern has made for one customer 21 sets of control gear for speed ranges up to 10 to 1, in sizes of from 75 to 250 h.p. Each set is self-contained on its own pedestal bearings and base, flexibly coupled to the driving motor and driven fan. Electro-magnetic flux linkage takes place between two elements. The outer member is a "stator ring" consisting of a circular steel backplate to which is clamped a laminated iron core, slotted for a squirrel-cage winding, which is rotated on roller bearings by the driving motor. The inner member, which drives the fan, consists of a steel spider carrying six magnetic poles without damper windings in their faces, the pole windings being connected to collector rings through which d.c. is conveyed for excitation. The driven speed relative to that of the driver is determined by the magnetic slip between the two elements, and is controlled by the amount of current permitted to flow in the magnet coils by an adjustable rheostat, which thus regulates the torque transmitted from the motor magnetically through the coupling to the fan. The d.c. excitation is provided by copper-oxide rectifiers that are fed with a.c. through variable-voltage adjustors actuated either manually or by Hagan automatic combustion-control equipment.—*"Electrical Review", Vol. CXXX, No. 3,354, 6th March, 1942, p. 314.*

Improved Air Preheaters for Boilers.

The latest design of air preheater for boilers developed by Jas. Howden & Co., Ltd., is described in a recently published British patent. The rotor of this air preheater comprises two stages each traversed by air and by gases, one of the stages permitting recovery of heat from the hot gases moving in contra-flow to the air flow through this stage, while the other stage permits the recovery of heat from the partially cooled gases moving in parallel flow to the air flow through the first stage. Figs. 1 and 3 are a vertical section, and a diagrammatic side elevation, respectively, of an air preheater of the Ljungström type embodying the above principles. Referring to Fig. 1, the vertical rotary shaft (*20*) carries a rotor which runs inside a casing (*21*), and which is made up of two stages of heating elements (*22* and *23*), with a central horizontal division plate (*24*), radial partition plates and vertical intermediate partition plates (*26*) fitted between the latter as shown. Radial passages (*27*, *28*) are afforded between the plate (*24*) and the two stages (*22* and *23*). The hot gases enter the gas side of the preheater by way of the gas inlet (*29*) and flow, as indicated by the dotted flow lines (*30*), upwards through the gas pass of the stage (*22*), radially outwards along the passages (*27*), radially inwards along the passages (*28*), and upwards through the gas pass of the stage (*23*) to the gas outlet (*31*). Referring to Fig. 3, cool air enters the air side of the preheater by way of a

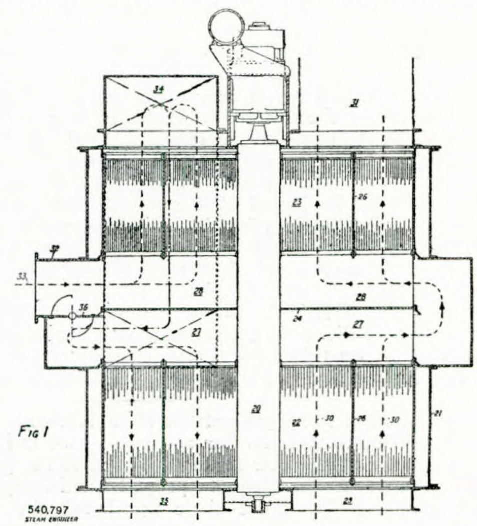


Fig. 1

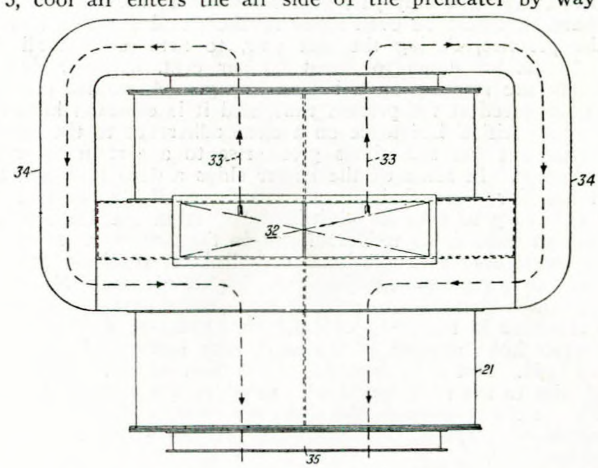


Fig. 3

lateral air inlet (*32*) and flows, as indicated by the dotted flow lines (*33*), radially inwards along the passages (*28*), upwards through the air pass of the stage (*23*), through external ducting (*34*) into the passages (*27*), radially inwards along these, and downwards through the air pass of the stage (*22*) to the hot air outlet (*33*). Thus, the hot gases traverse the stages (*22* and *23*) in the same axial direction, whilst the air first axially traverses the cool stage (*23*) in parallel with the gas flow through the latter, and passes axially through the hot stage (*22*) in contraflow to the gases. Axial, circumferential and radial seals, and stationary sector plates of suitable design, are provided to prevent any leakage between the air and gas spaces, or any by-passing of the heating surface.—*"The Steam Engineer", Vol. XI, No. 127, April, 1942, p. 194.*

Leaky Superheaters.

It would appear that trouble caused by leaks in the superheaters of high-pressure boilers has been experienced in some of the latest American steamships. Such leakage usually occurs at the tube ends, where these are expanded into the superheater headers. In service, the steam side of the superheaters is maintained at a temperature of 750° F. or more, and whilst the tubes are relatively thin, the headers are much thicker, for which reason the expansion and contraction of the former takes place more rapidly. Therefore, rapid cooling and quicker contraction of the tubes causes the tube metal to draw away from the header seat, thus producing leakage. The expansion and contraction of the metal corresponding to a change of temperature of something like 700° F. is obviously sufficient to cause leaky tube ends when the changes in temperature are fairly rapid, and for this reason special care is needed to guard against this. These parts of the boilers must, therefore, always be heated up or cooled down as slowly as possible when lighting up or shutting down, by suitable operation of the oil-fuel burners and air dampers.—*"Marine Engineering and Shipping Review", Vol. XLVII, No. 3, March, 1942, p. 146.*

New Type Feed Pump for Forced-circulation Boilers.

The sensitivity of drum-less, once-through, forced-flow boilers to load fluctuations makes it essential to provide accurate and sensitive control gear for the regulation of the supply of feed water and fuel. The task is further complicated by the fact that the quality of the control action must be maintained over the entire load range, *i.e.*, from zero load to maximum load. Infinitely-variable-delivery control of a reciprocating pump driven at constant speed can be achieved by dividing the pump into two cylinder groups with separate crankshafts, the relative angular positions of the latter being adjustable by mechanical means during operation. By connecting corresponding cylinders of the two groups to joint valve chests the output of the pump at constant speed is made a function of the relative angular positions of the respective cranks. An infinitely-variable output over the entire load range is thus obtainable with very little loss of efficiency. The general outline of a pump of this type is given in Fig. 1. The first application of this pump

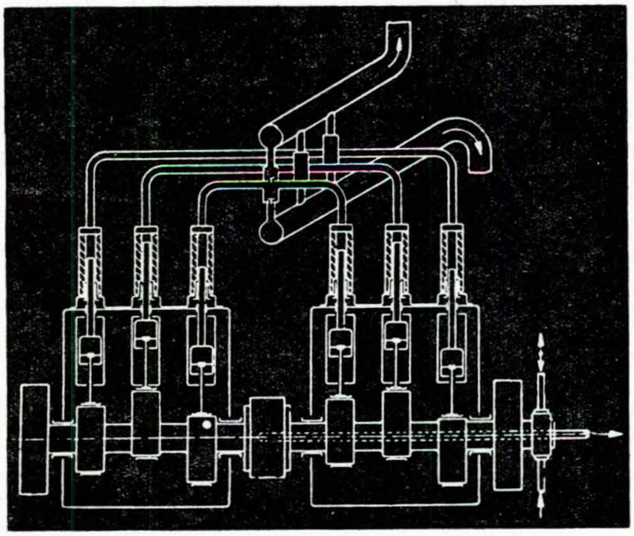


FIG. 1.

principle to once-through forced-flow boiler practice seems to have taken place in 1936, when the oil-fuel pumps of a large Ramzin boiler installed at the Moscow Thermotechnical Institute were designed to operate in this manner, but since that time very little has been heard of any further applications of this pump principle until quite recently. The construction of a Sulzer type feed pump designed to operate in a similar manner has now been announced. In the Russian installation, the adjustment of the relative crank angularity was effected by a planetary gear device, but this has now been superseded by a special type of oil-pressure coupling, an outline of which is shown in Fig. 2. The movable wall serving as a "piston" and two division walls forming an integral part of the casting, divide the latter into four chambers, two of which serve as pressure chambers, and the other two as baffle chambers. Maximum output is obtained when the two crankshafts are in the same position and the piston surfaces of the coupling contact with the division walls, thereby making the torque transmission from the division walls to the piston purely mechanical. However, when a change in angularity is required in order to reduce the pump discharge, an adequate oil pressure is built up in the pressure chamber by the action of the actuating control mechanism, and the piston surfaces move away from the compartment walls. At the same time, a portion of the oil contained in the buffer chambers is rejected into the pressure lubrication system of the bearings of the pump. The back pressure in the buffer system is maintained at a pressure high enough to ensure a balanced piston at all positions of the latter. Conversely, the pump delivery can be increased by lowering the oil pressure in the control system, thus enabling the torsional moment, in combination with the buffer oil pressure, to move the piston towards the compartment walls. The control gear for the actuating oil pressure is constructed on the principle of an oil-operated pilot valve.—*"Boiler House Review", Vol. 55, No. 10, April, 1942, p. 326.*

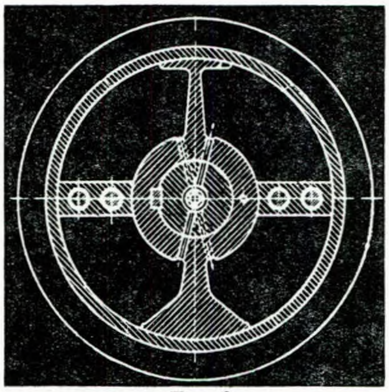


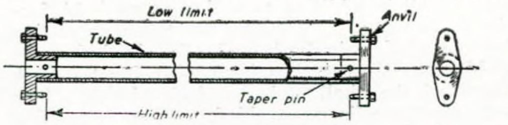
FIG. 2.

New 3,000-ton Railway Dry Dock at Halifax.

A new railway dry dock or slip, with a lifting capacity of 3,000 tons, has recently been constructed at Halifax, N.S. It consists of a steel cradle travelling over a four-way track on a system of free rollers. The overall length of this cradle is 370ft., the length over the keel blocks being 350ft. The width over the transverse beams is 60ft., with a clear width of 62ft., whilst the depth of water over the keel blocks at mean high water is 13ft. forward and 18ft. aft. Except for the deck, docking platforms and blocks, the cradle is built of structural steel and consists of two heavy central girders or runners and two lateral ones, to the lower sides of which steel rail plates are riveted, while the upper surfaces support the heavy transverse beams. Docking platforms, supported by steel uprights attached to the ends of these transverse beams, are arranged at each side of the cradle. The latter is decked over for its full width and length to provide a working platform. On each side of the cradle are 19 sliding bilge blocks operated by hand winches and chains. The railway tracks are of the four-way type, 700ft. long, constructed on a uniform gradient. The above-water portion is of reinforced concrete, whilst the submerged portion is of wood, protected against marine borers, the entire structure being supported on concrete foundations and wood piles, according to the varying nature of the sub-soil. The cradle moves over this track on a system of free rollers of alloy iron, operating between the rail plates attached to the lower sides of the cradle runners and the upper surfaces of the rails. Four endless hauling chains of weldless heat-treated manganese steel, operated by an electric hauling machine, are used to move the cradle, and are capable of dry-docking a vessel weighing 3,000 tons in 30 minutes. The cradle is constructed in two sections to facilitate maintenance. When necessary, the upper section can be floated off and the lower one hauled to the upper end of the track for examination, cleaning and painting.—*"Marine Engineering and Shipping Review", Vol. XLVII, No. 3, March, 1942, p. 134.*

Caliper Limit Gauge.

The accompanying sketch shows the construction of an improvised gauge which can be used for the accurate diametrical measurement of large discs, etc., *e.g.*, the end plates of marine main gear wheels of the built-up type. The gauge consists of a solid-drawn mild steel tube of the lightest gauge possible consistent with rigidity, with mild steel or light alloy ends, machined and fitted, pinned into each end. These ends each carry two hardened steel



An improved gauge.

anvils or pins, which are screwed into them and locked by nuts. Actual examples of such a gauge are made of solid-drawn tubing of 1⅜in. outside diameter and 1⅝in. bore, with a length of from 6ft. to 10ft. The anvil faces may be finished either flat or spherical and can be set by means of standard or ordinary length gauges to the required tolerance. The ends can be utilised for various lengths of tubes. Holes should be drilled in the ends to lighten them.—*"Practical Engineering", Vol. 5, No. 116, 9th April, 1942, p. 307.*

Method of Coal Storage.

According to *"Blast Furnace and Steel Plant"*, a U.S. patent has been granted to W. T. Brown, formerly research engineer of the Bethlehem Steel Company, on a method of coal storage by which oxidation of the coal is prevented. As is well known to those who store coal in large quantities, there is an upward passage of air through the pile and the greater the flow of air the greater the oxidation of the coal. The upward flow of air in a pile of coal is shown graphically in Fig. 1. The method for checking the oxidation of the piled coal covered by the patent referred to requires the coal to be spread on the ground to a height of rather less than 10ft. as it is received into storage. This layer of coal is level and preferably compacted by a roller or other means in order to decrease the spaces through which air can travel. The top and sides of this layer are then covered with Sisalkraft paper. Further layers of coal are piled on this layer and similarly covered with Sisalkraft, as shown in Fig. 2, until the pile is completed. The paper on the sides and ends of the pile is held firmly in position by covering it thinly with coal. When thus protected there can be no passage of air through the pile.—*"The Iron and Coal Trades Review", Vol. CXLIV, No. 3,867, 10th April, 1942, p. 331.*

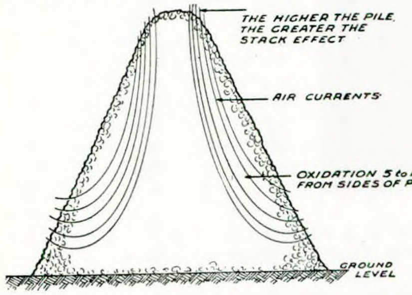


FIG. 1.

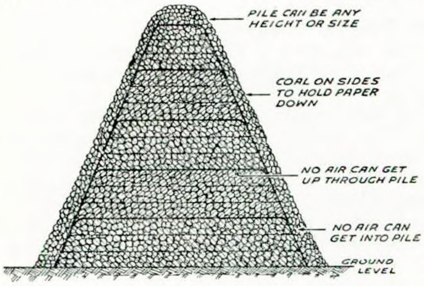


FIG. 2.

American Tanker "Sinclair H-C".

The completion at the Fore River Yard of the Bethlehem Steel Company of the steam tanker "Sinclair H-C" is claimed to mark the achievement of a world record in the building of a large ship. The vessel was launched 76 days after the laying of the keel plate—*i.e.,* after 63 working days on the slip—and delivered to her owners, the Sinclair Refining Company, 24 days later, making her a "100-day ship". The tanker has a gross tonnage of 7,874 tons and a full-load displacement of 16,182 tons. The hull is constructed on the longitudinal type of framing with the Frear-Bethlehem fluted bulkhead system. There are two continuous longitudinal bulkheads 17ft. 6in. at either side of the centre line of the ship extending from the forward pump room to the E.R. bulkhead. Longitudinal framing is employed for the shell and decks except at the ends, where transverse framing is used in the peak tanks, forward deep tanks and shell in the D.B. space under the E.R. compartment, at the after end of the machinery spaces and in way of the poop deck. Transverse bulkheads divide the hull into seven triple cargo tanks, making 21 main cargo oil tanks in all in addition to a dry cargo hold forward. Welding has been used to a great extent for the construction of the hull. The bottom plating is flush and all-welded for the entire length of the ship, but riveting has been used for the deck longitudinals, the seams of the shell and the deck plating. The remainder of the hull structure is arc-welded. The propelling machinery, which is located aft, consists of a set of De Laval cross-compound turbines driving a single propeller shaft through D.R. helical gearing, the output being 4,400 s.h.p. Steam at a pressure of 440lb./in.^{2} and total temperature of 740° F. is supplied to the turbines by two single-pass, sectional-header Babcock and Wilcox boilers installed on a flat above and abaft the turbines. They are each fired by three mechanical-pressure atomising oil-fuel burners and operate on the induced-draught system. Electric current is furnished by two 200-kW. 240-volt d.c. turbo-generators, supplemented by a 25-kW. 240/120 volt d.c. Diesel generator for emergency use. There are six steam-driven cargo oil pumps of the Warren vertical-compound duplex type, arranged in two pump rooms, each with a capacity of 140 tons/hr. against a pressure of 125lb./in.^{2}. The total capacity of these pumps is sufficient to discharge the whole of the oil cargo in 16 hours. Special attention has been paid to the design of the living quarters for the officers and crew, the men being berthed in double or single cabins and all the officers having separate rooms. The normal complement of the ship is 38 officers and men, but there is accommodation for more, in addition to messrooms, smoking rooms, a hospital, laundry, cold storage and provision rooms, etc. Hot and cold running water is provided in all cabins and mechanical ventilation is installed throughout the ship.—*"Marine Engineering and Shipping Review", Vol. XLVII, No. 3, March, 1942, pp. 106-113.*

Strength of Welded T-Joints for Ships' Bulkheads.

The author states that he had the privilege of collaborating with the late Professor B. P. Haigh, whose intention it had been to present this paper. The object of the work with which it is concerned was the investigation, under conditions of both static and alternating stresses, of the behaviour of different welded types of T-joints for the attachment of ships' bulkhead plates to the shell and tank plates. The author points out that the results are of equal interest for similar joints on other classes of engineering structures. The results are presented in the form of annotated graphs and photographs, and the conclusions arrived at are: *(1)* The simple fillet-welding of two plates T fashion gives a joint almost as strong as the parent plate under steadily imposed stress. *(2)* Such joints are unexpectedly weak under repeated equal compressive and tensile loads, and surprisingly so if pulsating loads are applied. *(3)* Preparing the edges of the plates so as to decrease the unwelded gap increases the fatigue limit by 65 per cent. *(4)* Decreasing the gap by 60 per cent. and grinding the toes of the welds raises the fatigue value by 90 per cent. If there is no gap and the toes of the welds are ground the fatigue limit is trebled, but is still 40 per cent. less than a butt-welded joint. *(5)* The simple fillet-welding of the plates with full gap gives a joint at least four times as strong in fatigue and five times as strong under steady tensile stress as plates attached by fillet-welded angles. *(6)* The best method of attaching these plates is to scarf the edges of the abutting plates so as to enable weld penetration to be made right through, without gap, and to do away with the fillets except for a very small corner radius. For this type of joint the limiting fatigue stress, both under conditions of equal compressive and tensile loads and of pulsating loads with steady imposed component, is four times that of plain filleted welds having the same tensile strengths as the plate, thus approaching within 80 per cent. of the fatigue strength of Lloyd's Class I X-rayed, welded butt joints.—*Paper (No. 11), prepared by H. B. Ferguson for publication in the "Transactions of the Institution of Naval Architects" for 1942.*

"Experiments with Low-pitch-ratio Screws Behind a Single-screw Hull".

The paper bearing the above title deals with the propulsion of cargo vessel with high-revolution screws of small diameter and pitch ratio, and describes the results obtained from tests made with a number of propellers of 0·5 face-pitch ratio behind a 24-ft. model of a single-screw ship. The first series of tests was made with all the propellers running in the deep load condition, whilst the second series comprised runs made with six propellers at draughts varying from deep load to a condition in which half an upright propeller blade was out of water. The results showed: *(1)* a variation of propulsive efficiency in the deep condition from 0·66 to 0·55 with normal design, and down to 0·50 with some abnormal screws; *(2)* that an increase in the blade area causes a loss of efficiency and an increase in revolutions—this latter change being contrary to the effect obtained by Froude; *(3)* that the best type of blade differs slightly from that which shows the best results at high-pitch ratios, having no washback of the face at the trailing edge; and *(4)* that although none of the screws broke down at thrusts below the self-propulsion point, the very narrow tip screw did break down at a slip somewhat above this, and a similar but worse breakdown occurred with a screw with symmetrical sections to the blades. Appendix I contains a summarised description of a novel method adopted by the authors to determine wake in light condition, whilst Appendix II explains how to find the approximate change in wake due to the stern wave for varying immersion of hull.—*Paper (No. 1)* *prepared by G. S. Baker, O.B.E., D.Sc., and L. T. G. Clarke, B.Sc., for publication in the "Transactions of the Institution of Naval Architects", for 1942.*

Variable-pitch Propellers for High Powers.

The results achieved with the new types of variable-pitch propellers fitted to Swedish-built coasters of fairly large size are stated to be promising. Plans for the utilisation of similar propellers for high-powered motorships have been prepared, and but for the war it is possible that particulars of the performance of a 16,000-ton, 14-knot tanker equipped with a variable-pitch propeller, would have been available at the present time. Experiments are likewise being conducted in regard to the use of variable-pitch propellers for fast motorboats driven by four high-speed engines with a total output of 4,000 h.p. The blades of these propellers are carried on roller bearings with suitable gearing, and the control shaft is driven by an electric motor actuated from the bridge. The propeller blades remain fixed in any direction, and can be feathered and reversed.—*"The Motor Ship", Vol. XXIII, No. 261, April, 1942, p. 3.*

Castable Refractories.

According to a statement published in the *"Iron and Steel Engineer"*, a new lightweight castable refractory recommended for service with temperatures as high as 2,200° F., has been developed by a Chicago firm. Called Plicast L-W-1, this material is intended for use where refractoriness is required with insulating efficiency. It is stated to be specially suitable for backing certain types of water walls in steam boilers, when tube spacing and operating temperature permit. Plicast L-W-1 weighs 57lb./cu. ft., and sets in 12 to 24 hours, with a compressive strength of 1,485lb./in.^{2}. The thermal conductivity, in B.Th.U./ft.^{2} per inch of thickness per hour per deg. F., is 1·80, 2·15 and 2·35 at mean temperatures of 400°, 800° and 1,200° F. respectively. The heat storage capacity of the new material is said to be 60 per cent. lower than that of firebrick.—*"Foundry Trade Journal", Vol. 66, No. 1,338, 9th April, 1942, p. 228.*

S.L.M. Starting and Relief Valve.

The Swiss Locomotive and Machine Works, Winterthur, have recently patented a combined air starting and relief valve for Diesel engines, two forms of which are illustrated in Fig. 2. Between the relief piston and the starting valve is an air admission pipe, and a detachable connection allows relative movement between the valve stem and the piston. The valve operates against the action of a spring, and the device is automatically put out of action as soon as the engine begins to fire. The valve stem has a head (*6*) which is interlocked with the relief piston (*8*) by means of a T-head bolt-and-slot connection. The piston serves as a push-rod, and a spring (*22*) engages its upper shoulder (*9*). The tappet lever (*10*) is mounted on an adjustable eccentric (*13*) and is actuated by a cam (*16*). A control arm (*17*) is provided and the diagrams show it in the extreme positions, one for allowing the push-rod (*14*) to be moved by the cam and the other for moving the rod out of range of the cam. Compressed air is supplied to the valve through a connection (*25*). At first the valve remains closed by the effect of the pressure acting on the piston (*8*) together with the force of the spring (*22*). When the piston moves down, due to the action of the tappet lever (*10*), the valve (*4*) opens and admits air to the cylinder. If, during the opening period, the contents of the cylinder ignite, the valve (*4*) closes independently of the piston (*8*). As soon as the engine starts firing, the control arm (*17*) is set to the right-hand position and the air supply to the connection (*25*) is cut off. The detail diagrams show the valve-stem connections to the piston. The head (*6*) is provided with flats (*18*) and a portion (*19*) of square cross-section. In assembling, the piston is pushed into the collar (*20*) against the pressure of the spring (*22*), while the head (*6*) is pushed through the slot (*21*) in the bottom (*7*) of the piston, and is then turned through an angle of 90°, whereupon the spring is released. The square portion (*19*) is of such a length (*L*) that it cannot separate from the slot (*21*) and thus lose its guidance. In the right-hand diagram the head of the bolt-and-slot connection is formed by the relief piston, and the valve stem has a hollow guide plunger (*26*). Pressure equalisation is effected by the provision of a passage (*27*).—*"The Oil Engine", Vol. IX, No. 108, April, 1942, p. 323.*

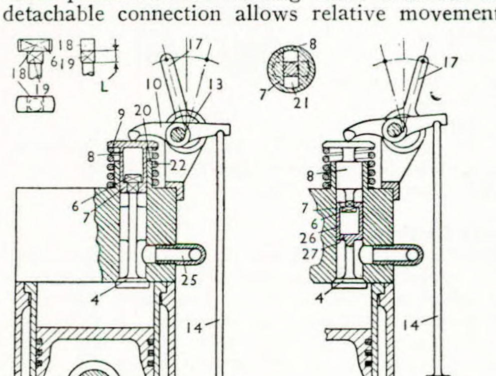


FIG. 2.

Atlas Diesel Fuel Timing Variation System.

An improved method of varying the fuel timing of a two-cycle Diesel engine has been developed and patented by the A.B. Atlas Diesel, of Stockholm. With this fuel timing system the moment of injection is dependent on the scavenging air pressure. When the engine is running at a constant speed of rotation the timing is made to vary with the torque. Earlier injection occurs with an increasing torque, giving improved combustion, while at lower torques and when the engine is idling, the injection occurs later. A diagrammatic representation of the engine (*M*), together with an engine-driven scavenging-air pump (*P*), is shown in Fig. 1. The fuel pump (*1*) has a camshaft (*2*) driven from the engine by a shaft (*3*). In these shafts are straight and oblique splines (*4, 5*) respectively, fitted in a sleeve coupling (*6*). The coupling is moved through a ring (*7*) having pins (*8*) engaging a forked lever (*9*), a rod (*11*) attached to this lever being connected to the piston (*13*) of a servo-motor. One end (*14*) of the latter is connected by a pipe (*15*) with the scavenging-air receiver of the engine, while the opposite end (*16*) is in communication with the exhaust pipe in front of a throttling device. The servo-motor piston (*13*) has a spring (*18*) which counteracts the pressure in the chamber (*14*). When the torque or revolutions alter, the scavenging-air pressure varies accordingly and the servo-motor piston (*13*) is moved in one direction or the other, thereby displacing the coupling sleeve (*6*) and causing it to rotate relatively to the shaft (*3*). This rotary motion is transmitted to the camshaft of the injection pump, so that the time of injection varies with the scavenging-air pressure. In engines which operate with a certain degree of throttling in the exhaust pipe, the space (*16*) of the servo-motor can, as already indicated, assist in the control by communicating with the exhaust pipe, making use of the pressure of the exhaust gas.—*"The Motor Ship", Vol. XXIII, No. 267, April, 1942, p. 34.*

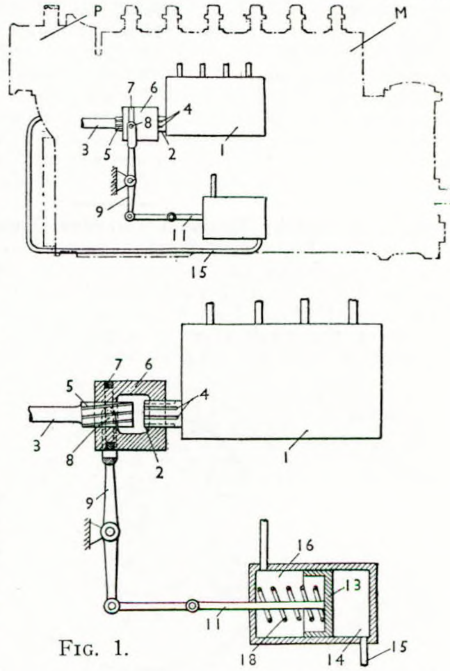


FIG. 1.

Economy Necessary in Tin for Bearings.

In order to effect economy in the use of tin for plain bearings and bearing metals, the Non-Ferrous Metals Control has adopted a number of recommendations by technical experts. It is admitted that some unavoidable risk is involved, but the guiding principle is to place the risk where the consequences of failure are likely to be the least serious. As regards alloy compositions, it is recommended that the use of white-metal bearing alloys containing between 12 and 68 per cent. of tin should be discontinued. White-metals are divided into four groups, according to their tin content, as follows: No. 1, 80 to 92 per cent.; No. 2, 68 to 75 per cent.; No. 3, 7 to 12 per cent.; and No. 4, 4 to 5 per cent. Backed bearings having thin linings are to supersede solid-cast unbacked white-metal bearings, but where this is not possible immediately, solid-cast bearings in groups Nos. 3 and 4 may be used. As to bearing bronzes, lined bearings in which no bearing properties are required of the backing should be of steel, brass or other tin-free backing material. Lined bearings in which bearing properties are required of the backing, as, *e.g.*, bearings with thrust faces or bearings which must continue in use after failure of lubrication or severe wear, may be made from tin-bronzes, but the use of 10 per cent. tin phosphor-bronze and of Admiralty gunmetal should be discontinued, alloys of lower tin content being employed instead. Where circumstances permit, the thickness of white-metal in a bearing should be reduced, and the statement is made that pre-finished and ready-to-fit bearings of less than 2½in. bore with linings of 0·03in. thick, of from 2½in. to 4in. bore with linings 0·04in. thick, and from 4 to 6in. bore and 0·06in. thick are being found satisfactory. Again, the thickness of white-metal in unflanged, thin-walled bearings should be reduced to between 0·01in. to 0·015in. Dovetailing should be eliminated where possible, or, if not, the volume of the dovetails reduced. As regards unlined bronze bearings and bushes, designers should be asked to consider their use instead of steel bushes lined with bronze.—*"The Machinist", Vol. 85, No. 53, 28th March, 1942, p. 385 E.*

Detecting Cracks in Crankshafts.

Magnetic testing of crankshafts as carried out at the Schenectady works of the General Electric Company includes the magnetising of the shaft, making the direction of the flux longitudinal so that it will intercept any possible cracks at right angles. While magnetised, the shaft is sprayed with kerosene carrying in suspension finely divided particles of magnetic iron oxide. The test is made as a matter of routine each time a shaft is removed from a machine which is normally subjected to very heavy strains. One large firm recently tested 43 crankshafts in this manner and found that 14 of them were cracked and required replacement.—*"Foundry Trade Journal", Vol. 66, No. 1,336, 26th March, 1942, p. 200.*

Ruston and Hornsby Torsional Vibration Damper.

A torsional vibration damper constructed so that the elastic portions of the device can be replaced and adjustments carried out, obviating the necessity for a frictional grip or bonding of the rubber to the assembly, has been developed by Ruston and Hornsby, Ltd., and forms the subject of a recent British patent. The general arrangement of the device is shown in Fig. 2. The boss (*3*) is keyed on the crankshaft (*4*) at the point of maximum amplitude of torsional vibration, usually at the end remote from the flywheel; an external flange (*5*) is provided. The damping mass (*6*) has an internal flange (*7*) and the faces of the flanges are clear of each other. Aligned holes (*8*) through each flange accommodate rubber bushes (*9*) and the coupling bolts (*10*) are reduced to form shoulders (*11*). The distance between the faces of the bolt heads (*12*) and the shoulders of the bolts is greater than the thickness of the flanges and less than the length of the rubbber bushes. Washers (*19*) are fitted as shown in the diagram. When the nuts (*13*) are screwed up, relative rotational movement between the shaft and the damping mass takes place freely, the bushes being deformed by compression and sheer strain. A modification of the arrangement is shown in the lower detail diagram. This makes use of a number of rubber washers (*15*) which fit into holes (*8*) in the damping mass (*6*). Studs (*10*) are employed, together with nuts (*13, 16*) and stud washers (*17*). With this arrangement the torsional vibration is damped by the compression of the rubber washers, the stiffness in torsion and consequently in the damping capacity being varied by altering the number of washers employed.—*"The Motor Ship", Vol. XXIII, No. 267, April, 1942, p. 34.*

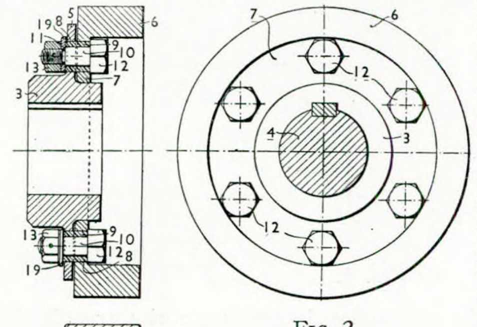
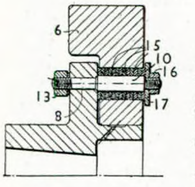


FIG. 2.



Bolinder Diesel Engine Crankhead Bearings.

An improved design of crankhead bearing for oil engines has recently been patented in this country by the Swedish makers of the Bolinder Diesel engine. Tilting bearing blocks are employed, their size varying according to the position in the housing. White metal is not used, the blocks being preferably made of steel and coated with a lead-bronze alloy. Forced lubrication is provided, and the weight of the bearing is reduced owing to the saving in the dimensions of the housing at those points where the blocks are not located. Referring to the accompanying sectional drawings (Fig. 3), the two parts of the bearing (*1, 2*) carry the blocks (*3, 4, 5*), the bottom block (*4*) being smaller than the upper block (*3*), and both being unsymmetrical with reference to the centre line (*6*). That portion of the block (*3*) which is located behind the centre line, viewed in the direction of rotation (*7*), is larger than the forward part and the block is eccentrically supported along a line (*8*). The surface bearing against the base of the recess (*9*) has a smaller radius of curvature (r_1) for its rear portion than the radius (r_2) for the forward part. The smaller blocks (*5, 5*) take up the lateral thrust and the combined peripheral extension of all the blocks is less than half the bearing circumference. Packing rings (*10*) are fitted between the connecting-rod end and the crank-web faces (*11*). Adjustment of the rings is provided by fitting an artificial-rubber ring (*12*), while a spiral spring (*13*) ensures that each packing ring (*10*) will be pressed against the web face (*11*). The retention of the oil pressure by this device facilitates the discharge of oil up the connecting rod to the gudgeon pin.—*"The Motor Ship", Vol. XXXIII, No. 267, April, 1942, p. 34.*

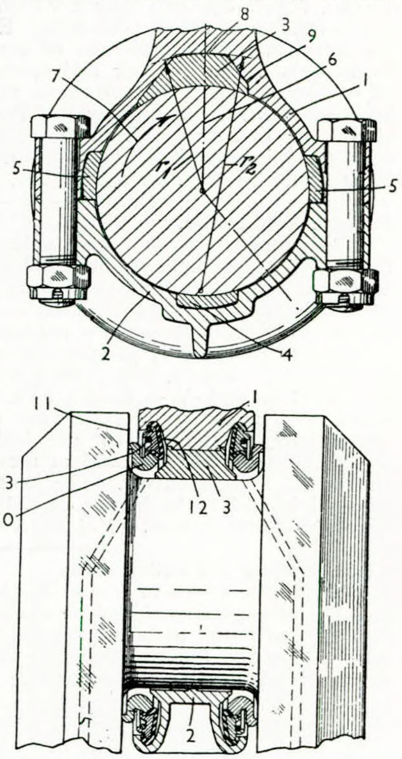


FIG. 3.

Tin Substitute in Bronze Castings.

An alloy metal known as P-M-G Hardener, has been developed in the United States to replace tin in the production of bronze castings. The new metal, which is being smelted in increasing quantities, is claimed to produce good castings at no greater cost. It is already being used by the U.S. Navy and private shipbuilders, and is reported to be exceptionally valuable in the manufacture of valves, pump bodies, impellers, pipe flanges and numerous other castings, including parts of ammunition hoists and gun mountings.—*"Foundry Trade Journal", Vol. 66, No. 1,336, 26th March, 1942, p. 200.*

Tin Solders.

The present need for economy in the use of tin is making it necessary to modify the composition of tin solders, and with this object in view the British Non-Ferrous Metals Research Association is issuing a second edition of its monograph, *"Tin Solders"*. The first edition, by S. J. Nightingale, has been revised by Dr. O. F. Hudson and brought up to date in the light of investigations carried out during the last 10 years in the Association's laboratories and elsewhere, whilst additions to the second edition include new sections dealing with the creep properties of solders and soldered joints. The book is priced at 10s. 6d. post free, and is obtainable from the B.N.F.M.R.A., Euston Street, London, N.W.1, or through any bookseller.—*Press notice issued by the British Non-Ferrous Metals Research Association on the 14th April, 1942.*