Marine Lubrication

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Lubrication is rapidly developing from hit and miss, trial and error methods to a more exact science, although there is still a vast amount to be learned on the subject. The lubrication of machinery used aboard ship frequently poses special problems for the oil technologist. Typical of such problems is that of severe corrosion in geared turbines caused by salt water entering the oil system, or the increased cylinder wear in large Diesel engines operating on residual fuel oils. In this paper the author briefly covers the refining of lubricating oils, laboratory tests and their significance, the use of additives, the fundamental principles of lubrication, and lubrication problems associated with reciprocating steam engines, geared steam turbines and Diesel engines.

REFINERY PROCESSES

Crude petroleum occurs naturally in many parts of the earth and is one of the most valuable natural resources available to mankind. The greatest source of petroleum is still the U.S.A., but the Middle East oilfields are becoming increasingly important, and production in the not too distant future may well rival that of the earlier developed fields in America. In addition, the recently developed oilfields in Canada are believed to have a potential which will compare favourably with the other major oilfields of the world.

Although crude petroleum is found in many different localities, there are essentially three basic types of crude, namely paraffinic base, naphthenic base, and mixed base, which is a mixture of the two former. Unfortunately for the oil refiner, however, crudes are seldom found which correspond exactly with one of these classifications; in other words, essentially paraffinic crudes will include a proportion of naphthenes, and predominantly naphthenic crudes will contain paraffinic components.

All crude petroleum oils are most complex mixtures of different hydrocarbon compounds, ranging from simple structure low molecular weight gaseous compounds such as methane, propane, and butane, to complex high molecular weight heavy cylinder oils and asphalts. Advantage is taken of the different boiling range of all these various compounds to break down the crude into a series of commercially useful products, such as gasoline (or petrol), kerosine (or paraffin), gas oil, lubricating oil stocks, wax, bitumen and residual fuel oil, to mention only the main groups of compounds isolated, although there are many hundred different compounds which are seldom separated.

Refineries themselves vary considerably and have to be designed essentially for the type of crude oil which they are to handle. For example, a good quality paraffinic crude will contain as much as 30 per cent of lubricating oil stocks, whereas other crudes, essentially of naphthenic type, may contain as low as 5 per cent of lubricating oil fractions. Certain crudes contain 55 per cent of bitumen while others contain none. It is obvious, therefore, that a change in crude oil supply presents major difficulties to an oil refiner and would probably require extensive alterations to the refinery equipment.

Some idea of the complexity of a modern refinery can be gathered from Fig. 1, which shows a typical flow chart, and even here, only the major units are shown.

As we are mainly interested in lubricating oils in this paper, emphasis will be laid on the methods of producing the many different types required to meet the demands of modern industry.

It will be seen that crude oil brought in by tanker is pumped into storage tanks and thence into a pipe still furnace, which somewhat resembles a watertube boiler in which crude oil takes the place of water. Refinery gases and residual fuel oil are employed to raise the temperature of the crude oil to about 650 deg. F. before passing into the tall distillation, or fractionating, tower, in which the various compounds, or "fractions", are separated. Superheated steam is usually fed into the tower which assists distillation by carrying upwards the lighter fractions. By employing superheated steam the initial crude oil temperature can be reduced which lessens the risk of cracking. Inside the tower are a series of trays containing bubble caps through which vapours must pass as they rise in the tower. Some of the heavier vapours condense on the trays, while the lighter vapours rise to the top. Petroleum gases such as methane and propane pass out at the top of the tower, followed by gasolene, kerosene and the heavier fraction in succession as shown.

Distillation, of course, is only the first stage in refining, as the various fractions must be condensed and then subjected to extensive processing before the finished products are suitable for use.

In the diagram the lubricating oil stock is shown as a single stream whereas in an actual refinery it is split up into fractions ranging from light spindle stock to heavy steam cylinder oil. In the majority of refineries the lubricating oil fraction is fed into a second pipe still and a second fractionating tower, usually operated under a vacuum, to reduce the temperature at which the various fractions will distil off. Where the feed stock to the second tower has previously had gas oil and lighter fractions removed, it is known as "topped" crude.

The crude oil being handled in the refinery illustrated is of the asphaltic type, the residue taken from the bottom of the first fractionating tower being a mixture of cylinder stock, heavy residual fuel oil and asphalt, or bitumen, as it is more commonly termed in the petroleum industry. The bitumen is separated from the fuel oil and after further processing is

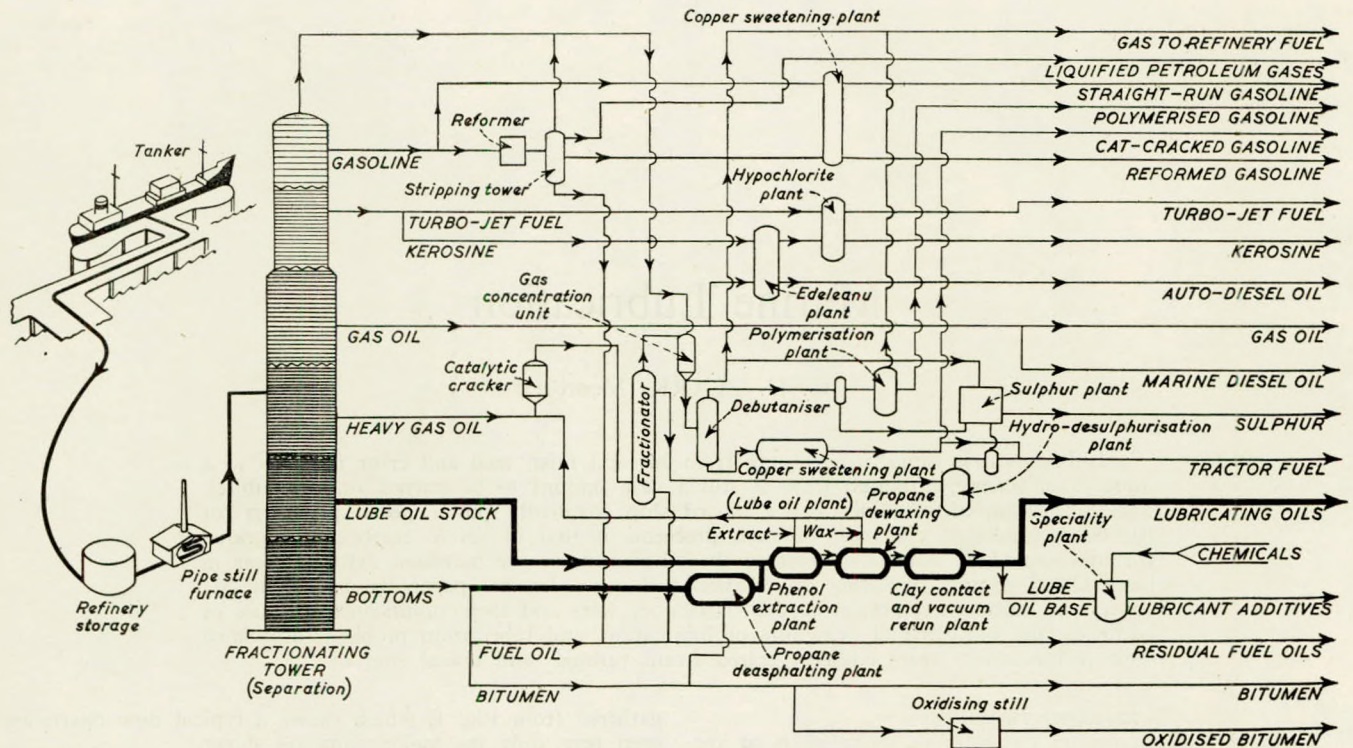*Technical adviser on fuels and lubricants, Esso Petroleum Co., Ltd.

FIG. 1—*Typical modern refinery flow diagram*

used for road making, waterproofing materials, insulating compounds and many other purposes. The fuel oil, which is later joined by residues from other processes in the refinery, is sold as residual fuel oil and is used as bunker fuel for steam-raising in industry and aboard ship, and of recent years, as heavy fuel for large marine Diesel engines. Marine Diesel

fuel, as can be seen, is a distillate taken off at an earlier stage and is, of course, of much lower viscosity. The cylinder oil fraction is separated from the fuel oil and bitumen and is further processed before use.

To return to lubricating oils, there are two main methods of refining after distillation to remove undesirable constituents,
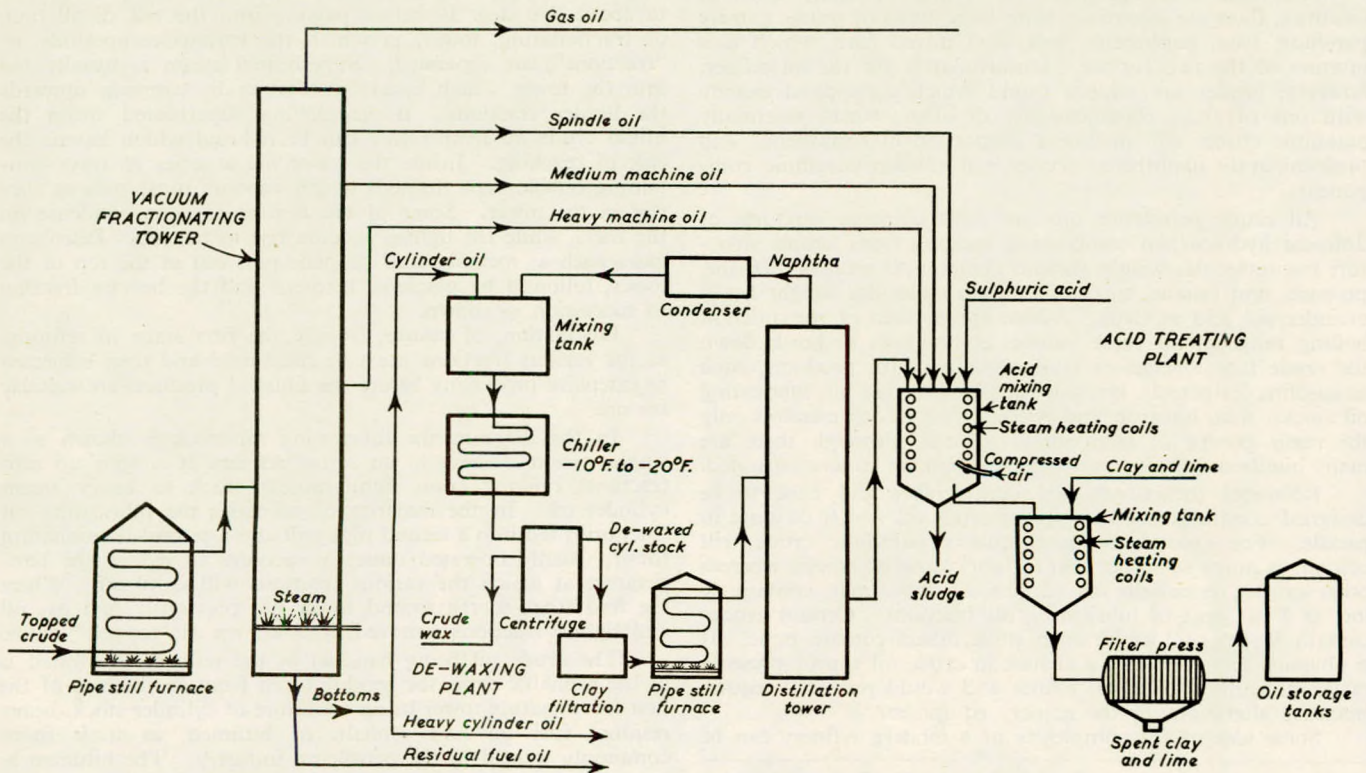

FIG. 2—*Refining of lubricating oil fractions: acid treatment*

namely acid refining and solvent refining. Certain of the compounds removed from the lubricating oil stocks, such as wax and bitumen, while undesirable in finished lubricating oils are extremely valuable and are widely used in industry.

ACID REFINING OF LUBRICATING OILS

Sulphuric acid refining, or conventional refining, as it is often termed, was until comparatively recently the main method of producing lubricating oils. A simplified layout of producing finished oils by this method is shown in Fig. 2. Topped crude is passed into a pipe still where it is heated to about 600 deg. F. before being passed into a vacuum fractionating tower into which superheated steam is also fed. As before, the lower boiling point fractions pass to the top of the tower, the lightest fraction, gas oil, being led to the main gas stream from the first fractionating tower. The boiling points correspond with the viscosity of the fractions, the lightest being spindle oil, followed by medium machine oil, heavy machine oil and finally cylinder oil. In this case, the stock being treated is of a paraffinic nature containing paraffin wax, the greater part of which must be removed in order to keep the pour point of the finished oils within acceptable commercial limits. A certain amount of paraffin wax is desirable, however, as the large molecules are good lubricants and give great stability to the finished oil.

Wax is difficult to remove from lubricating oil fractions as the boiling points are similar. The earliest method was to chill the mixture which crystallized the wax; the oil was then passed through a filter plant which retained the wax but allowed the oil to pass through. This method required large filters which had to be periodically cleaned to remove the wax. An improved method was developed in which a solvent is added to the oil, the mixture then being chilled well below the pour point required in the finished oil and filtered to remove the wax, usually by centrifuge. The dewaxed oil is then distilled to remove the solvent, which is condensed and re-used, the oil passing to an acid treating plant while the wax removed is taken away for further separation and processing. Wax removed from residual cylinder stocks is of the amorphous or micro-crystalline type, whereas when removed from distillate fractions, it is termed paraffin wax.

In certain cases the lighter lubricating oil distillates are also dewaxed, but in Fig. 1 they are led direct to the acid plant in which concentrated sulphuric acid is mixed with the oil in a lead-lined agitator tank. Although in the diagram all the oils to be treated are taken to the one tank, in practice each stream is batch-treated individually, the temperature during treatment, and the amount of acid treat, depending on the oil viscosity. Agitation in the tank is achieved by blowing the mixture with compressed air.

Treatment with sulphuric acid removes unstable aromatic and asphaltic constituents from oils, which improves their oxidation stability and reduces sludge and varnish-forming tendencies. The undesirable compounds react with the acid and form acid sludge which settles to the bottom of the tank and is periodically removed. At one time this sludge was an unwanted by-product but during recent years it has become extremely valuable as a source of petroleum sulphonates, widely used in cutting oils, rust preventives and as a base for synthetic detergents.

The treated oil still contains certain acidic soluble compounds and these are removed by mixing with activated alumina and an alkali, such as lime, in a steam-heated tank and then passed through a filter press. The alkali neutralizes the acid and other acidic, or polar, bodies are absorbed by the clay which is periodically removed from the filter. The finished oil is then led to storage tanks from which it may be used direct or blended with oils of different viscosities to give any required commercial grade.

Asphaltic or naphthenic feed stocks contain very little paraffin wax but may have an asphalt content as high as 55 per cent and, of course, a reduced yield of lubricating oil.

Furthermore, such crudes do not contain cylinder stocks as these are almost entirely paraffinic in nature. These topped feed stocks are led to a pipe still and vacuum distillation tower as previously described and the various distillates are taken off and sulphuric acid treated. The undistilled residue consisting of black oils and asphalt are treated to remove the asphalt, and the remaining oil, which is of relatively poor quality, is used as a cheap oil for rough machinery.

Solvent Refining of Lubricating Oils

During recent years, machinery of all descriptions has become more critical in its lubricant requirements. Bearing loads have greatly increased, in many cases operating temperatures are very much higher, as for example in modern high-pressure, high-temperature steam turbines; viscosity index has become increasingly important in such applications as aviation oils and certain hydraulic systems, but despite such requirements the lubricating oils have frequently to remain in service for much longer periods without deterioration and must maintain an oil film between moving surfaces and so reduce wear to a minimum.

Such conditions have shown the limitations of conventionally refined oils and it became necessary to develop new refining techniques to produce better lubricants from the same crude oils. A major step forward was the introduction of selective solvent refining, and today practically all premium quality oils are produced by this method. Even with the most careful sulphuric acid refining, it is impossible to remove entirely all the unsaturated aromatic compounds and asphaltic constituents except by very severe treatment which would also remove certain desirable constituents. Such treatment would involve considerable loss of oil with correspondingly large amounts of acid sludge. By treating the lubricating oil distillates with carefully selected solvents, however, the finished oils have greatly improved viscosity indexes and stability and sludge and varnish-forming tendencies are reduced to a minimum.

The principle of solvent refining is that desirable naphthenic and paraffinic lubricating oil fractions have little or no solubility in selected solvents, whereas the less stable unsaturated aromatic compounds are highly soluble. It is possible, therefore, to separate the different types of lubricating oil fractions on the basis of molecular type or structure. The solvent selected depends largely on the type of crude being handled and the plant used is most complex in nature. Typical solvents employed in this selective separation are liquid sulphur dioxide, furfural, phenol, chlorax and nitrobenzene.

In a typical batch solvent extraction plant the selected solvent is thoroughly agitated with the lubricating oil fraction being handled, to ensure that the undesirable constituents are dissolved in the solvent. Agitation is then stopped and the mixture settles into two layers, the upper layer consisting of the desired stable constituents plus a small amount of solvent, while the lower layer consists of the unwanted constituents and the bulk of the solvent. The layers are then drawn off separately and both streams are distilled to remove the solvent which is then condensed and re-used. The aromatic compounds and other impurities are further treated and made into useful products for certain industrial uses. The refined oil is then pumped into storage tanks or blending tanks, and may be treated with additives, depending on its eventual use.

Batch extraction suffers the disadvantage of not being a continuous operation and in many refineries it has been superseded by countercurrent extraction. In this system the oil fraction and the solvent are fed into the treating tank in opposite directions so that as the oil leaves the tank, it meets the incoming fresh solvent, while partially-spent solvent mixes with the incoming oil containing impurities. The whole system is thus continuous and allows a greater throughput than the batch extraction method.

Propane De-asphalting

This is a relatively new development in solvent refining

TABLE I—TYPICAL INSPECTIONS OF MARINE LUBRICANTS

Grade	Specific gravity at 60 deg. F.	P.M. closed flash, deg. F.	P.M. open flash, deg. F.	Pour point, deg. F.	Redwood No. 1 viscosity			Kinematic viscosity		V.I.	Remarks
					at 70 deg. F.	at 140 deg. F.	at 200 deg. F.	at 100 deg. F.	at 210 deg. F.		
						Seconds		Centistokes			
Light hydraulic oil	0·875	305	320	−50	120	42½	37	13·76	2·92	58	Suitable for telemotor systems
Refrigerator oil	0·893	385	400	−35	480	74	43	44·5	5·12	66	Suitable for ammonia, CO_2 and Freon systems
Medium-heavy turbine oil	0·872	425	445	15	690	111	52	68·4	8·62	107	Contains anti-oxidant and rust inhibitor
Diesel crankcase oil	0·898	425	445	−10	1,550	159	59	121·6	10·36	69	Contains anti-oxidant, also suitable for use as cylinder lubricant
Heavy gear oil	0·950	355	370	10	—	370	94	371·1	18·28	39	Medium-heavy extreme pressure gear oil
Heavy-steam cylinder oil	0·903	520	560	30	—	745	186	695·0	36·9	94	High-grade filtered steam cylinder oil

and can be said to be a reversed process to that previously described. Paraffinic, naphthenic, and the lighter aromatic hydrocarbon constituents of lubricating oil stocks, are readily soluble in liquid propane, whereas asphalts, which are heavy complex aromatic compounds, are completely insoluble. By mixing propane with asphaltic residual stocks which cannot be removed by distillation in the fractionating tower, the desirable lubricating oil fractions can be separated from the asphalt, which is removed for further treatment before sale. The lubricating oil extract is then further processed by normal solvent extraction methods. In this way asphaltic residues can be made to yield high-quality lubricating oils. Liquid propane may also be used to remove wax from lubricating oil stocks.

Fig. 1 illustrates typical refinery methods for extracting high quality lubricating oils from an asphaltic crude. In this simplified diagram the propane de-asphalting plant can be seen handling undistilled residium from the base of the fractionating tower. The treated lubricating oil fraction, of the cylinder oil type, then passes into the solvent phenol extraction plant for removal of lighter aromatic compounds. After solvent treatment the previously dark coloured cylinder oil becomes bright and clear and is termed solvent-refined bright stock. Dark cylinder oils are, of course, widely used for the internal lubrication of cylinders and valve chests in reciprocating steam engines, particularly where superheated steam is employed.

The distillate lubricating oil stocks are similarly treated in the phenol extraction plant, then dewaxed, filtered, and finally pumped to storage tanks. In practice, of course, the several distillate fractions and the treated residium are treated separately, in order to give a variety of finished oils or blending stocks.

It will be noted that mention has been made of only a very limited number of lubricating oil fractions being taken from the fractionating towers. Although in practice additional cuts are taken, the number is still limited, the required viscosity of any commercial grade being obtained by carefully blending the various finished oils. Light turbine oils, for example, are usually a straight distillate but a heavy Diesel oil will be a blend of distillate and bright stock. Steam refined cylinder oils are widely used as heavy gear oils where the "heavy ends" as they are termed, are valuable load carrying components.

LABORATORY TESTS OF LUBRICATING OILS

Inspection figures for typical oils widely used as marine lubricants are given in Table I; these include a light hydraulic oil, a refrigerator oil, a medium heavy turbine oil, a Diesel engine crankcase oil, an extreme-pressure gear oil and a steam cylinder oil. These oils vary widely in viscosity, which is, of course, one of the most important physical properties of a lubricating oil.

Viscosity

The viscosity of a liquid is a measure of its internal friction, or its resistance to flow. A light distillate, such as kerosine, flows very freely and is said to have a low viscosity. On the other hand, a heavy cylinder oil which flows sluggishly at ambient temperature, has a high viscosity. In more technical language viscosity is defined as "the force required to move a plane surface of one square centimetre area over another plane surface at the rate of one centimetre per second when the two surfaces are separated by a layer of liquid one centimetre thick". The unit of this force is known as the "poise", which is the unit of absolute viscosity.

Kinematic Viscosity

Kinematic viscosity is the ratio of the absolute viscosity of the oil to its specific gravity at the temperature at which the viscosity is measured. The unit of kinematic viscosity is the "stoke" although for convenience it is more usual to express it in "centistokes" (i.e. 1/100th of a stoke). The viscosity in stokes multiplied by the specific gravity at the test temperature gives the absolute viscosity in poises at that temperature.

Direct determination of absolute viscosity is difficult and for practical purposes in the laboratory it is much more convenient to measure viscosity by noting the time taken for a given quantity of oil to flow through a standard capillary tube at certain fixed temperatures.

There are two main methods of measuring viscosity. Absolute, or kinematic, viscosities are obtained from a glass instrument of which there are several types, the two most widely used being the Ubbelöhde viscometer and the U-tube viscometer. In both types a given volume of the oil sample is drawn up by vacuum or blown up a capillary tube and is then allowed to fall by gravity. A typical Ubbelöhde viscometer is illustrated in Fig. 3, which shows the capillary tube from which the oil flows and the time is taken for the oil level to pass between the upper and lower etched marks. The kinematic or absolute viscosity of the oil being tested is then calculated from a simple equation.
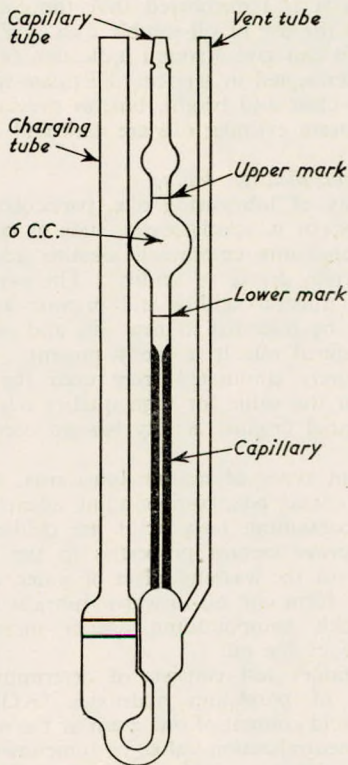

FIG. 3—*Ubbelöhde kinematic viscometer*

The viscometers are suspended in a water or oil bath with the upper ends of the tubes just above the liquid level. The bath is accurately maintained at the temperature at which the viscosity is required.

Redwood Viscosity

The kinematic viscometer is much more accurate than the older Redwood viscometer which has been widely used in the United Kingdom and it is now customary to obtain the kinematic viscosity and to convert it to Redwood viscosity by means of suitable tables. Industry is much more familiar with viscosities measured in Redwood seconds, but it is likely that kinematic viscosities will be universally adopted in the future.

There are two forms of the Redwood instrument, similar in principle, known as the No. 1 and the No. 2 viscometer. The No. 1 instrument is used for all oils, the viscosity of which does not exceed 2,000 seconds at the test temperature. The No. 2 instrument is designed to have a rate of flow ten times as fast as the No. 1 and is therefore generally used for viscous oils with viscosities exceeding 1,000 seconds Redwood No. 1.
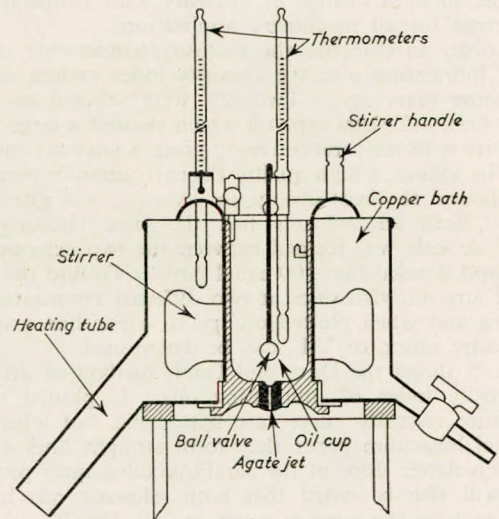

FIG. 4—*Redwood No. 1 viscometer*

A typical Redwood No. 1 instrument is illustrated in Fig. 4 and consists essentially of an oil cup, into the base of which is fitted a carefully calibrated agate jet through which the oil flows when released by removal of a spherical plug. The oil is surrounded by a water bath which is maintained slightly above the required test temperature. The viscosity of the oil sample is obtained by taking the time in seconds for an exact quantity of oil to flow through the jet into a calibrated flask placed below.

The Redwood viscosity of lubricating oils is usually expressed as Redwood No. 1 seconds at three temperatures, namely 70 deg. F., 140 deg. F., and 200 deg. F., while for fuels it is more customary to take the viscosity at 100 deg. F.

In the U.S.A. the Saybolt viscometer, which is similar to the Redwood instrument, is widely used, while on the Continent the Engler instrument is favoured.

It is usual to quote kinematic viscosity in centistokes at 100 deg. F. and 210 deg. F. and the same temperatures are generally used for expressing Saybolt viscosities in S.S.U. (Saybolt Seconds, Universal).

Viscosity Index

The viscosity of all mineral oils decreases as the temperature increases, the reduction in viscosity, or "thinning out", being greater with naphthenic oils than with paraffinic oils. This phenomenon is of major importance with all types of lubricating oils, as, everything else being equal, the oil which
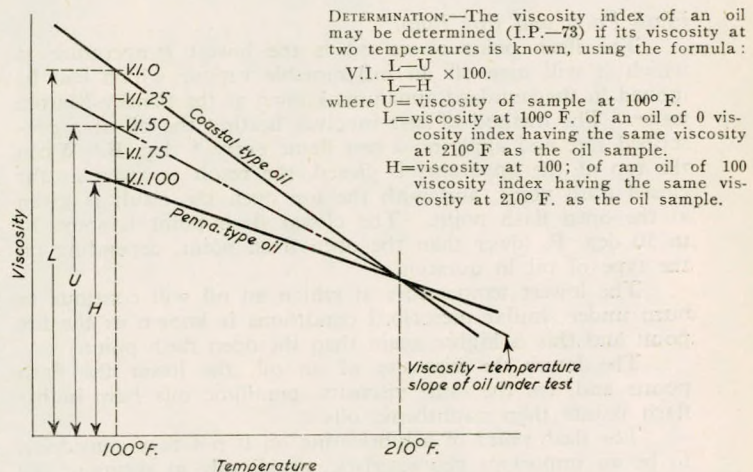

DETERMINATION.—The viscosity index of an oil may be determined (I.P.—73) if its viscosity at two temperatures is known, using the formula:

$$V.I. \frac{L-U}{L-H} \times 100.$$

where U = viscosity of sample at 100° F.
L = viscosity at 100° F. of an oil of 0 viscosity index having the same viscosity at 210° F as the oil sample.
H = viscosity at 100°; of an oil of 100 viscosity index having the same viscosity at 210° F. as the oil sample.

FIG. 5—*Dean and Davis method of determining viscosity index*

shows the smallest change of viscosity with temperature is to be preferred for all machinery applications.

In order to compare the viscosity/temperature characteristics of lubricating oils, the viscosity index system was introduced some years ago. Two oils were selected as reference oils, the first, a Coastal type oil which showed a large variation of viscosity with temperature, being given a viscosity index value of 0. The second, a high quality Pennsylvanian or paraffinic oil which showed the least change in viscosity, was given a value of 100. Both selected oils had the same viscosity at 210 deg. F. A scale was formed between the two extremes at 100 deg. F. and divided into 100 equal parts. To find the viscosity index of any oil, viscosities at two different temperatures must be known and when plotted on special logarithm graph paper the viscosity index, or V.I., can be determined.

Fig. 5 shows the Dean and Davis method of determining the viscosity index of lubricating oils. If plotted normally, temperature viscosity curves are hyperbolic, but when plotted on special logarithm paper they form straight lines as shown. The much flatter slope of the paraffinic oil is most pronounced and it will also be noted that both reference oils have been selected to have the same viscosity at 210 deg. F.

The viscosity index may be calculated if its viscosity at two temperatures is known, from the equation $V.I. = \dfrac{L-U}{L-H} \times 100$, where U = viscosity of sample at 100 deg. F., L = viscosity at 100 deg. F. of an oil of 0 V.I. having the same viscosity at 210 deg. F. as the sample, and H = viscosity at 100 deg. F. of an oil of 100 V.I. having the same viscosity at 210 deg. F. as the oil sample.

It is usually more convenient, knowing the oil sample viscosity at two temperatures, to read off its viscosity index from specially prepared charts.

Pour Point

The pour point of an oil is important in certain applications, a particular case being refrigerator oil where it is essential that the oil should flow at temperatures below the lowest found anywhere in the system. Lubricating oils, having complex chemical structures, do not solidify at definite temperatures and any determination of this characteristic must, therefore, be arbitrary. In a paraffinic oil, it is the solidifying of wax crystals which finally prevents the oil flowing as its temperature is lowered, but in the case of naphthenic oils, the viscosity increases so rapidly at low temperatures that they finally become too viscous to flow. It will be noted, however, that low viscosity index naphthenic oils have lower pour points than paraffinic oils and are, therefore, widely used as refrigerator oils. Solvent dewaxing of paraffinic oils lowers the pour point considerably and high viscosity index turbine oils, for example, frequently have pour points as low as 10 deg. F.

Flash Point and Fire Point

The flash point of an oil is the lowest temperature at which it will give off an inflammable vapour which can be ignited in the standard apparatus known as the Pensky-Marten tester. The method of test involves heating the oil at a prescribed rate and applying a test flame every 5 deg. F. When the top of the apparatus is closed, the result is given as the closed flash point, and with the top open the result is given as the open flash point. The closed flash point is some 10 to 50 deg. F. lower than the open flash point, depending on the type of oil in question.

The lowest temperature at which an oil will continue to burn under similar prescribed conditions is known as the fire point and this is higher again than the open flash point.

The lower the viscosity of an oil, the lower the flash points and, for the same viscosity, paraffinic oils have higher flash points than naphthenic oils.

The flash point of a lubricating oil is not now considered to be an important characteristic, as all oils in common use have flash points well above the 150 deg. F. minimum stipulated by the Board of Trade. Even when considered as a fire risk, flash point is usually less important than the spontaneous ignition temperature of an oil, as the majority of oil fires have been caused by spontaneous combustion rather than by the oil being brought into contact with a naked flame.

Specific Gravity

Specific gravity is of little value in evaluating the properties of a lubricating oil. With acid refined oils it gives some idea of the origin of the oil as paraffinic oils have the lowest specific gravity, but with solvent refined oils coming into greater use this can be somewhat misleading. With liquid fuels, however, gravity is important when calculating thermal values where the fuel is bought by volume. The higher the gravity the greater the weight per unit volume and, therefore, the higher the calorific value.

Colour

The colour of a lubricating oil has little significance, particularly when it is remembered that the colour can readily be changed by the use of oil-soluble dyes. With used turbine oils, however, it can give a rough indication of whether or not the oil has deteriorated in service. Distillate-type high quality oils should be clear and bright, but, as previously mentioned, residual-type steam cylinder oils are dark and opaque.

Acidity or Neutralization Value

The acidity of lubricating oils, particularly used oils, is important. Except in special cases, such as certain detergent-type Diesel lubricants employing alkaline additives, all new oils have a certain degree of acidity. The two possible forms of acidity are mineral acidity and organic acidity. Mineral acidity cannot be tolerated in new oils and except with poor quality acid-refined oils it is rarely present. Organic acidity cannot be entirely eliminated from even the most carefully refined oils but the value for high quality oils is usually very low. Such initial organic acidity has no corrosive effects on machinery.

For certain types of marine lubricants, as, for example, steam engine bearing oils, compounding agents, such as blown rapeseed oil, containing fatty acids are deliberately added to impart or improve certain properties in the base oil. Such an oil must resist the washing effect of water, and in eccentric pans it must form an oil-in-water lubricating and cooling emulsion. Such compounding agents increase the initial organic acidity of the oil.

The laboratory test consists of determining the number of milligrams of potassium hydroxide (KOH) required to neutralize the acid content of one gram of the oil. The amount is termed the neutralization value, or sometimes the neutralization number, and is expressed as "X" mgm. KOH/gm.

All mineral oils oxidize in service to a greater or lesser degree depending on the type of base oil used, the method and degree of refining and whether or not an anti-oxidant additive is incorporated. As oxidation proceeds, organic acids and other oxidation products are formed. It is possible, therefore, by taking the neutralization value of a used oil to determine in conjunction with other tests the degree to which oxidation has progressed. It is not possible, unfortunately, to set a limit to the neutralization value of any used oil, as this varies with the service in which it is employed, the type of oil in use and whether or not additives have been used. In modern turbine oils, for example, a value of 0.3 mgm. KOH/gm. for a used oil would be considered high and the figure would not be reached until it had been in service many years, whereas a Diesel-engine crankcase oil could reach this figure within a few weeks.

Some manufacturers of high quality turbine oils give an upper limit for their turbine oils of 0.2 mgm. KOH/gm., after which they advise that the oil should be taken out of service, but such oils may be expected to give some twenty years' service before this figure is reached.

It will thus be seen that quoting the neutralization value of a used oil is in itself of little value. It is the rate of increase

of neutralization value which is of more significance, but here again the value obtained should not by itself determine whether or not an oil is fit for further service. In oil-circulatory systems containing large volumes of oil, such as steam turbines, Diesel engines, large gearboxes and similar applications, the neutralization value should be determined at intervals agreed between the oil supplier and the operator, as it is usually possible, after considering all the factors concerned, to remove the oil from service before oxidation has progressed to the stage when troublesome sludge, varnish, and other objectionable oxidation products have formed.

Demulsibility or Steam Emulsion Number

Straight new oil will not mix with water under normal conditions. If, however, oil and water are finely sub-divided or dispersed in one another an emulsion is formed. With a few exceptions, depending on which predominates, this emulsion can be water-in-oil or oil-in-water. On standing, however, separation takes place, with the oil rising to the top, the rate of separation depending on the presence of materials such as acids, alkalis, soaps or compounding agents purposely added or present as contaminants. Dirt and other foreign matter also affects the rate of separation.

It is highly important that steam turbine oils should separate readily from water as it is impossible to entirely exclude all water from the lubricating oil systems. Stable emulsions could lead to bearing failures or to sludge formation resulting in choked oil pipes and oil cooler tubes. Modern turbine oils, however, usually incorporate an anti-oxidant to improve stability and a rust inhibitor to prevent corrosion of oil-bathed surfaces. Certain of these additives have a marked effect on demulsibility as determined by laboratory methods, but under service conditions no detrimental effect has been observed. Under such conditions the value of the test method employed must be suspect.

With reciprocating steam engines it is essential that the internal working parts should be adequately lubricated, but at the same time any oil carried over with the exhaust steam must separate readily from the condensate, otherwise serious trouble with overheating may occur in the boilers.

On the other hand, for certain industrial applications, an emulsifying agent is deliberately incorporated in the oil so that a stable emulsion will be formed. Good examples of this are soluble cutting or grinding oils, and soluble roll oils used in the cold rolling of steel strip or sheets.

As oxidation of an oil progresses, polar bodies are formed which act to some degree as emusifying agents and in extreme cases a stable emulsion can be formed which will not break under normal operating conditions. Such emulsions are dangerous, and an oil should be removed from service long before this could happen.

Standard test methods of determining demulsibility, such as the steam emulsion number, are of little value with used oils, particularly Diesel engine lubricants, due to the presence of contaminants, and are rarely taken. Even with new oils, steam emulsion test results can be misleading because of the extreme sensitivity of the test.

Sediment and Water

Both sediment and water can be classed as contaminants and the amount which can be tolerated depends largely on the service to which the oil is applied. For example, even very small quantities of water cannot be tolerated in electrical insulating oils or in refrigerator oils. In the presence of certain refrigerants, moisture will form highly corrosive mineral acids which can seriously damage working parts. One specification for a refrigerating oil demands that the moisture content must not exceed four parts per million.

On the other hand, Diesel engine lubricants in large marine engines have shown water contents as high as 8 per cent without apparent damage to the working parts. It is obviously desirable, however, that the water content in all lubricating oils should be kept to an absolute minimum.

Sediment is always undesirable in any lubricating oil. Everything possible is done by the oil supplier to deliver oil completely free from moisture or sediment so as to avoid damage to working parts, but in certain services trouble is frequently experienced with foreign matter finding its way into the oil and causing abrasive wear. Diesel engines are, of course, particularly liable to trouble through foreign matter entering with the induction air. Air compressors, too, frequently give trouble for the same reason.

Rust particles formed from corrosion in lubricating oil systems frequently cause damage. Everything possible must be done to prevent corrosion, but if it cannot be entirely prevented the oil should be kept as clean as possible by efficient filtration.

A standard sediment and water test, frequently termed B.S. and W. (Bottom Sediment and Water), is often carried out in the laboratory. It is particularly important in the case of dark, heavy fuel oils and residual steam cylinder oils, where, because of their opaqueness, it is difficult to detect the presence of suspended solids.

OXIDATION TESTS

In service, except perhaps in completely oil-filled hydraulic systems, lubricating oils are always exposed to air in some part of the system. All petroleum oils consist essentially of a complex mixture of hydrocarbons, and when exposed to the atmosphere the oxygen present combines chemically with the hydrocarbon molecules to form new compounds. This chemical reaction is termed oxidation.

High viscosity index paraffinic hydrocarbons are much more stable than aromatic or unsaturated hydrocarbons; that is, they are more strongly resistant to attack by oxygen. The nature and type of oxidation products formed are very varied, ranging from oil-soluble organic acids to insoluble sludge, varnish, and solid carbonaceous compounds. In general, paraffinic oils tend to form more weak organic acids but less insoluble compounds than naphthenic or aromatic oils.

Being a chemical reaction, temperature has a great influence on oil oxidation. The rate of oxidation of lubricating oil on the walls of a Diesel engine cylinder, for example, is many times greater than the rate of oxidation of a similar oil in a Michell thrust bearing, where operating temperatures are relatively low.

It is true to say that all oil-oxidation products are undesirable. Insoluble solids or semi-solids may choke oil cooler tubes, filter screens or small oil pipes, eventually starving bearings of oil, with subsequent high wear and eventual failure. Soluble materials circulating with the oil can cause corrosion to bearing metals and finely-finished machine parts, although organic acids are very weak as compared to mineral acids, which can only be present as contaminants in a well-refined oil.

Although it is impossible to prevent oil oxidation, it can be considerably retarded. Carefully selected crude oils, refined by modern methods with the addition of powerful anti-oxidants, have resulted in lubricating oils of a degree of stability far superior to oils produced a few years ago.

It is obviously desirable to obtain accurate knowledge regarding the oxidation resistance of new oils before selection for a particular application is made. This is particularly important where the oil is subjected to difficult conditions for long periods. Many laboratory oxidation tests have been developed to try and assess the oxidation resistance of lubricating oils but, unfortunately, as oxidation is normally fairly slow, it is necessary to accelerate the process by deliberately creating much more difficult conditions than the oil would meet in service. For this reason no laboratory test correlates closely to field conditions, so that oxidation test results can only be taken as an approximate guide to the behaviour of an oil under actual operating conditions.

ADDITIONAL TESTS

Many other tests may be carried out on petroleum pro-

ducts, depending on their nature and eventual use. For example, with distillate fuel oils it is usual to determine the distillation range, cetane number and sulphur content. Tests are carried out on special gear machines to determine the load-carrying properties of extreme pressure gear oils. Steel corrosion tests are carried out with new turbine oils to ascertain their ability to prevent corrosion in lubricating oil systems. Dilution tests are carried out on used Diesel crankcase oils to see to what degree the oil has been contaminated with unburnt fuel oil. With electrical oils used for insulating purposes dielectric strength tests are carried out to determine the insulating properties of the oil. The ash content of used detergent-type Diesel lubricants is frequently taken to determine to what degree the additive has been consumed in service.

All these tests require highly skilled chemists and, in many cases, elaborate and costly apparatus, so that a full analysis of, say, a used Diesel lubricant may require several days' work by a number of men. This might well be borne in mind by operators of plant who demand "all the answers" a day after a test sample is received in a laboratory!

ADDITIVES

Although mention will be made later of additives used in lubricating oils for special applications, it may not be out of place to devote a few lines to a brief overall survey of this subject. By definition, additives are substances which are added to petroleum oils to modify or enhance their natural properties. Additives in lubricating oils can be compared to elements added to iron to form alloy steels, possessing special properties.

It cannot be overemphasized, however, that an additive will not change a poorly refined, poor quality base oil into a high quality lubricant. The oil itself must be of a suitable type, of the correct viscosity and properly refined before it should be considered for any specific application.

The earliest additives used in mineral oils were fatty oils, or fatty acids, such as lard, tallow, rapeseed oil, oleic acid, stearic acid and several others. These were found to give superior metal-wetting properties and to improve load carrying in applications where speeds were too low or loads too high to permit fluid film lubrication, i.e. where working parts are completely separated by a film of oil. In other cases, such as in steam cylinder oils used in engines operating with wet steam, fatty materials like lard or tallow were found to resist the washing effect of water in the steam and thus provide better lubrication of the internal rubbing surfaces.

In modern practice it is usual to term such materials compounding agents to distinguish them from chemical compounds developed for specific purposes.

Some of the more common additives employed for modern high quality mineral oils are: —

Anti-oxidants to slow down the rate of oxidation.
Rust-inhibitors to prevent corrosion of oil-bathed surfaces in circulating oil systems.
Bearing corrosion inhibitors mainly used in high-speed internal combustion engine oils to prevent corrosion of copper lead and similar hard bearing alloys.
Detergent-dispersant additives used in many Diesel engine lubricants to keep pistons, piston rings and liners clean for long periods under severe operating conditions.
Pour-point depressants used in lubricants exposed to low temperature conditions as, for example, in aircraft engines and refrigerators.
Extreme-pressure (E.P.) agents used in gear oils subjected to heavy intermittent or continuous loading.
Adhesive agents which, when added to oils or greases, improve their anti-spattering properties and greatly improve their water-resistant properties.

It must not be supposed that all such additives are used in any one oil but the complex demands of modern machinery in many cases require a lubricant virtually tailored for the job. The blending of additives into a lubricating oil is a highly specialized job which must be left to the oil supplier. All the characteristics of the base oil must be known before a suitable additive can be selected, as the response of an oil to an additive can vary considerably. In some cases certain oils have been found to be completely incompatible with specific additives, and serious trouble could occur if some of the high molecular weight additives came out of solution, as these would accumulate in the lowest points in the oil system, possibly causing choking of oil pump suction strainers.

LUBRICATION—FUNDAMENTAL PRINCIPLES

During recent years there have been noteworthy advances in our knowledge of the science of lubrication and in particular the work done by Bowden and his associates needs no introduction. Several papers presented to the Institute during the last few years,[1, 2, 3] have included valuable information on this subject. These comments are, therefore, confined to a broad outline of the fundamental principles of lubrication, and will be referred to later in the paper when specific lubrication problems associated with marine prime movers are being dealt with.

The main purpose of lubrication is to interpose a film of oil or grease between fixed and moving metallic surfaces, or between two moving surfaces, in order to prevent metal-to-metal contact, thereby reducing friction to a minimum and preventing mechanical wear of the surfaces.

Lubrication of moving bodies can be divided into two main categories: —

(1) *Fluid film lubrication,* or hydrodynamic lubrication, is said to exist when the moving surfaces are completely separated by a continuous unbroken fluid film of lubricant.

(2) *Boundary lubrication* (or thin film lubrication as it is termed by some authorities) exists when conditions are such that it is not possible to form a full fluid film. Under these conditions some degree of metal-to-metal contact usually occurs between the moving and fixed surfaces.

Some authorities include a third category, namely partial fluid film lubrication. This may be said to be an intermediate stage between boundary and full fluid film lubrication and can be regarded as a halfway stage between the two.

Fluid Film Lubrication

It is obvious that the ideal condition to aim at in any piece of moving machinery is when friction, and thereby loss of power, is at a minimum and where mechanical wear of the moving parts is eliminated. This can be achieved in practice by providing conditions suitable for the formation of a fluid film of lubricant, so that the moving parts "float" on a film of oil.

The factors to be considered before fluid film lubrication can be achieved are: (*a*) the load on the bearing, as obviously the higher the load the more difficult it is to establish the oil film; (*b*) the speed of rotation of the journal, as the higher the speed, the easier it is to form an oil film; (*c*) the viscosity of the lubricant; (*d*) the radius of the journal and (*e*) the clearance between journal and bearing. A further essential requirement is that there must be an ample continuous supply of oil to the bearing.

Viscosity of the oil is a most important feature, because the higher the viscosity the greater the load it will carry. On the other hand, there is continual fluid friction between the molecules of oil forming the oil film, and with a very viscous oil, fluid friction or "drag" can be considerable, involving an unnecessary waste of power. With a very low viscosity oil, even at high speeds bearing end leakage can be great enough to prevent the formation of a fluid film. A compromise must, therefore, always be made between an oil with a viscosity just great enough to carry the load and a more viscous oil which provides a greater margin of safety.

As a simple illustration of fluid film lubrication, take the case of a journal rotating in a bearing under constant load conditions, with an ample supply of oil fed to the bearing.
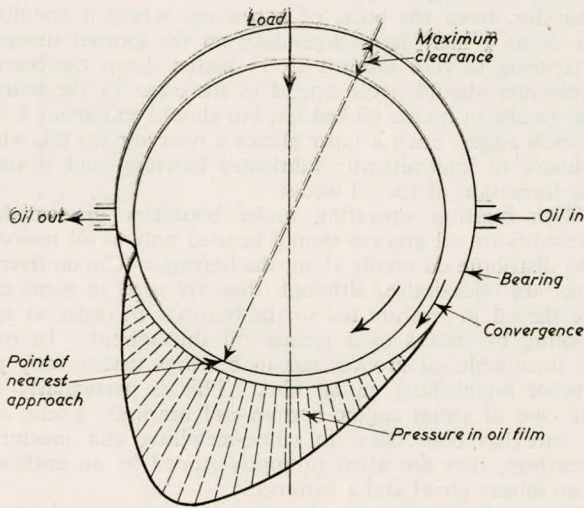
FIG. 6—*Pressure distribution in a journal bearing*

As shown in Fig. 6, the load on the bearing tends to squeeze out the oil film. This is resisted by the viscosity of the oil and the speed of the journal, which exert an equal and upward pressure on the journal. It will be seen that the pressure within the oil film rises from zero at the point of oil entry, builds up to a maximum just past the vertical centre line, then drops rapidly and may actually create a slight negative pressure as the oil leaves the bearing.

Due to the speed of rotation, the point of nearest approach between journal and bearing and the greatest pressure exerted by the oil film is not vertically downward as might be expected but is offset in the direction of rotation, as shown in the illustration.

As the journal starts from rest, particularly if the machine has been out of use for some time, there is only a thin adsorbed layer of oil between journal and bearing, therefore metal-to-metal contact occurs, but as speed increases oil is dragged by the journal into the converging space and the pressure gradually builds up until the shaft is lifted and carried on the oil film. Equilibrium is reached when the pressure in the oil is equal in magnitude, but opposite in direction, to the applied load.

As has already been mentioned, under hydrodynamic conditions, the sole source of friction in the bearing is fluid friction within the oil film and this is proportional to the viscosity of the oil employed. Fluid friction, however, is much lower than solid or mechanical friction, being usually in the order of $0\cdot002$-$0\cdot01$ unless extremely viscous oils are used. The energy consumed by fluid friction in a bearing is converted into heat and the temperature of the oil rises, with a corresponding fall in viscosity. A state of equilibrium is reached when heat generation within the oil film equals heat dissipation from the bearing.

Obviously the most suitable lubricant from the fluid friction point of view is one with a viscosity just high enough to carry the load at the operating speed of the bearing. Unfortunately, however, all machines have to be stopped and started and, furthermore, operating conditions, particularly load, are rarely constant. It is desirable, therefore, that the oil viscosity should be considerably greater than the minimum required to support the load, so as to give a reasonable factor of safety to take care of changes in operating conditions. In addition, the increased viscosity allows a more rapid establishment of a fluid film and so reduces wear.

In any bearing operating under fluid film conditions there is a definite relationship between the coefficient of friction (μ), the absolute viscosity of the lubricant (Z), the speed of rotation (N), the radius of the journal (r), the load per unit area (P), and the clearance (c). In bearing calculations the Sommerfield

number (S), is frequently used, and this is obtained from the equation $S = \left(\dfrac{r}{c}\right)^2 \dfrac{ZN}{P}$.

It can be more simply stated, however, that the coefficient of friction is equal to a function of the absolute viscosity of the oil, multiplied by the speed of the journal in r.p.m., divided by the load per unit area in lb. per sq. in. The dimensionless expression used is $\mu = f\dfrac{ZN}{P}$ where f is a variable function for different bearing designs and conditions. For the purposes of the equation it is assumed that there is no end leakage of oil from the bearing, that there is no rise in temperature due to fluid friction which would reduce the oil viscosity, and that there is no rise in viscosity due to bearing pressure. Such conditions are not, of course, met with in practice, but a typical $\dfrac{ZN}{P}$ curve as illustrated in Fig. 7 gives a fairly clear picture of fluid film lubrication. Assuming for simplicity that
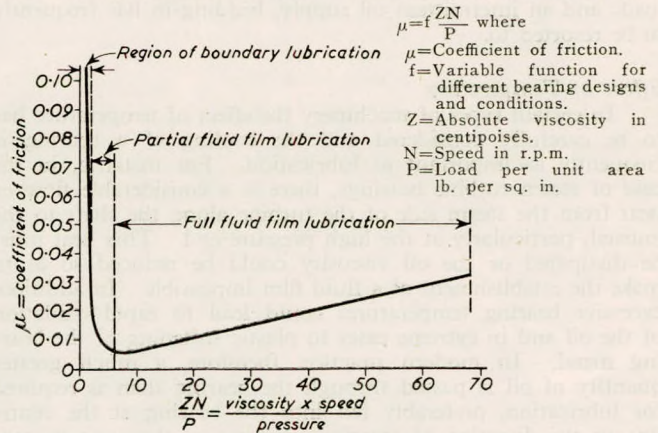

$\mu = f\dfrac{ZN}{P}$ where
μ = Coefficient of friction.
f = Variable function for different bearing designs and conditions.
Z = Absolute viscosity in centipoises.
N = Speed in r.p.m.
P = Load per unit area lb. per sq. in.

FIG. 7—*Typical $\dfrac{ZN}{P}$ curve*

the viscosity of the oil and the load on the bearing are constant, then the curve represents the coefficient of friction plotted against the speed of the journal. Starting from rest, the initial friction (due to metal-to-metal contact), is very high but this falls rapidly as speed increases until it reaches a minimum point, after which there is a gradual rise again. To the left of the minimum point, conditions of boundary lubrication with high mechanical friction exist, while to the right, there is a full fluid film lubrication with much lower fluid friction. The region of partial fluid film under which conditions are unstable is also indicated.

Bedding-in of Bearings

Fluid film conditions can only exist in full clearance bearings, i.e. where the radius of the bearing is machined greater than that of the journal so as to provide the necessary convergence and sufficient clearance to lift the journal on to the oil film. If, however, bearings are completely bedded-in, the radii become equal and, although the bearing may still have sufficient vertical clearance, this destroys convergence between
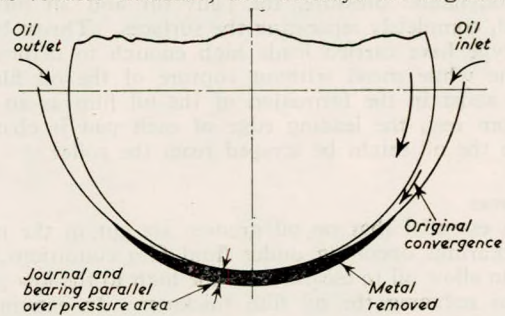

FIG. 8—*Effect of bedding-in a journal bearing*

journal and bearing on the oil inlet side and it is practically impossible to create a fluid film. As shown in Fig. 8, bearings are usually bedded-in over the arc of contact where loading is greatest, so that journal and bearing are parallel over only a relatively short distance. Even so, such conditions make it more difficult to establish a full fluid film. It will thus be seen that even if a bearing has been correctly designed and machined to give the necessary clearance and convergence for the lubricant, this can be largely nullified by bedding-in. Generally speaking, therefore, bedding-in of bearings is considered to be a bad practice; it should only be resorted to in applications where fluid film conditions are unlikely to exist and where the bearing metal shows definite high spots, indicating metal-to-metal contact.

Bedding-in of turbine bearings cannot be recommended, as, due to the high journal speeds and relatively light loads, conditions are ideal for the establishment of fluid film lubrication. On the other hand, with slow-speed reciprocating steam engine bottom end bearings, with fairly heavy fluctuating loads and an intermittent oil supply, bedding-in has frequently to be resorted to.

Effect of Temperature

In certain types of machinery the effect of temperature has to be carefully considered and the cooling of a bearing is frequently as important as lubrication. For instance, in the case of steam turbine bearings, there is a considerable flow of heat from the steam side of the turbine along the shaft to the journal, particularly at the high pressure end. This heat must be dissipated or the oil viscosity could be reduced so as to make the establishment of a fluid film impossible. In addition, excessive bearing temperatures could lead to rapid oxidation of the oil and in extreme cases to plastic softening of the bearing metal. In modern practice, therefore, a much greater quantity of oil is passed through the bearing than is required for lubrication, preferably fed into the bearing at the centre line in the direction of rotation, that is, on the opposite side to that shown in Fig. 6, so that the greater part passes over the top of the journal, thus acting as a coolant, while only a relatively small amount is carried round with the journal into the pressure area.

Michell Thrust Bearings

The journal bearing does not, of course, represent the only example of fluid film lubrication. An equally important example is the Michell thrust bearing which is widely used in marine equipment. This bearing has almost entirely replaced the older type of multi-collar main thrust block in modern vessels. In the horseshoe multi-collar thrust, the white metalled bearing faces were parallel to the collars on the shaft, and as speeds were slow and there was no convergence between bearing and collar, fluid film conditions could not be established, therefore designed maximum bearing loads were usually in the order of 50lb. per sq. in.

In the Michell thrust block a single collar is used. A number of white metal-faced pads are fitted which are free to tilt or pivot in the direction of rotation. The revolving collar drags oil into the clearance between collar and pad, building up hydrodynamic pressure, the pads tilt and an oil wedge is formed, completely separating the surfaces. Thrust bearings of this type have carried loads high enough to deform plastically the white metal without rupture of the oil film. In order to assist in the formation of the oil film as an engine starts from rest, the leading edge of each pad is chamfered, otherwise the oil might be scraped from the collar.

Oil Grooves

It is essential that no oil grooves are cut in the pressure area of bearings operating under fluid film conditions, as the effect is to allow oil to escape from the high to the low pressure area, thus reducing the oil film thickness. In extreme cases the leakage can be so great as to prevent the formation of a fluid film. It is good practice, however, to machine a taper, or chamfer, from the butts of a bearing, where it should be about $\frac{1}{16}$ to $\frac{1}{8}$ inch deep, depending on the journal diameter, and tapering to zero about 1 to 1½ inches down the bearing. The chamfer should never extend to the edges of the bearing, as this results in undue oil leakage, but should end about $\frac{1}{4}$ inch from each edge. Such a taper allows a reservoir for oil, which is valuable in intermittently lubricated bearings, and it assists in the formation of the oil wedge.

With bearings operating under boundary or semi-fluid film conditions, oil grooves should be used only as oil reservoirs and to distribute oil evenly along the bearings. Circumferential grooves are undesirable, although they are used in some cases where the oil is pressure fed to the bearings in order to assist in cooling by means of a greater oil throughput. In many cases, unsuitable oil grooves cut in bearings destroy any possibility of establishing an oil film. This is particularly true in the case of steam engine bottom-end bearings, where, after great care has been taken in white-metalling and machining the bearings, they are often promptly ruined by an enthusiast with an oilway chisel and a hammer!

Where oil grooves must be provided, it is important that both leading and trailing edges should be chamfered, otherwise oil will be scraped off the journal, instead of the groove supplying oil to the journal as was intended.

Boundary Lubrication

If all machinery could be designed to operate under fluid film lubrication, wear would be confined to stopping and starting and bearings would last indefinitely. Unfortunately, there are many cases where bearing loads are too high, speeds too low, bearing design unsuitable or method of application of the lubricant unsatisfactory and a fluid film cannot be established or maintained. Under such conditions the journal, slider, or gear is said to operate under boundary conditions of lubrication.

Worm gears provide a good example of machinery operating under boundary lubrication conditions, as the greater part of the motion between worm and worm-wheel is sliding; in addition, loads are frequently high and speeds low. Another example is a guide shoe working on a guide bar, or slab, in a slow-speed reciprocating engine. At the end of the stroke the shoe comes to rest and then reverses direction, so that it is impossible to establish a fluid film.

In boundary lubrication, asperities of metal on the moving surface protrude through the oil film, coming in contact with similar asperities on the fixed surface, causing high sliding friction with a marked local rise in temperature. In some cases temperatures are high enough to melt the metal, and with two hard metals in contact, cause local seizures. With a combination of a steel journal and a white-metalled bearing under such conditions, the softer metal asperities melt and run into the adjacent hollows or valleys. It must be remembered that with finely finished parts the asperities are extremely minute and may not exceed one or two microns in height.

Even under severe loading it is doubtful whether an oil film is ever completely squeezed out. All fluids increase in viscosity as pressure is applied[4] but the effect under normal loads is negligible. Where pressures reach 2,000 atmospheres (about 28,400lb. per sq. in.), however, the relative increase in viscosity of mineral oils is about fifteen times that at atmospheric pressure, while at 4,000 atmospheres (56,800lb. per sq. in.) the relative viscosity increase is 112. Under such conditions the viscosity of the oil is so high that it becomes a solid. In the majority of applications, however, as high pressures are applied, side leakage allows most of the lubricant to escape before the solid state is reached but some of the oil remains trapped in surface hollows or remains adsorbed on the surface.

As evidence of this phenomenon, in the case of cold rolling of steel, pressures exerted by the rolls exceed the yield point of the steel, which is plastically deformed, and the yield point of alloy steels can exceed 80,000lb. per sq. in. Even

under such conditions the surface of the steel is still thinly coated with the roll oil on the outgoing side of the rolls. An adsorbed film of roll oil must, therefore, have resisted pressures high enough to deform the steel as it passes through the roll gap.

The local melting of surface asperities referred to must not be confused with surface welding of areas which can result in seizure of a bearing. In the latter case a much greater area is involved and parts of the surface of one body are torn out by the other. This surface welding could be described as an extreme case of the local welding of asperities.

As the surface asperities are smoothed over, or melted, and the peaks flow into the valleys, the whole surface acquires a smoother finish. This is in effect what takes place when two surfaces are "run-in".

In the case of journal bearings, good surface finish of both journal and bearing is extremely important, whether the bearing operates under fluid film or boundary lubrication conditions. With smooth surfaces it is obvious that the surfaces can be separated by a thinner oil film than when they are in a rough condition.

With Diesel cylinder liners, however, it has been found that a superfine finish can cause excessive wear, and in some cases, seizure. This apparent anomaly can be explained by the fact that particularly towards the ends of the stroke, it is unlikely that fluid film conditions can exist and as there are no surface hollows to trap the oil it is squeezed out. This is particularly true at the top end of the cylinder where the high surface temperatures greatly reduce the viscosity of the oil. In the almost complete absence of lubricant, metal-to-metal contact becomes excessive, with correspondingly high friction and wear.

Improved Boundary Lubricants

One of the most obvious ways of reducing metal-to-metal contact under boundary conditions is to employ a high viscosity oil such as a heavy cylinder oil, and these oils are frequently used for applications such as the lubrication of large, heavily loaded, slow-speed gears. In the majority of cases, however, there is a practical limit to the viscosity of the oil which can be used. High viscosity oils are difficult to pump, particularly in cold weather, and they cannot be handled satisfactorily in many simple lubrication devices, such as drip feed or siphon-type lubricators.

Lubricants used under boundary conditions should possess the property of adhering tenaciously to metal surfaces, this property being frequently termed "oiliness". Animal and vegetable fatty oils possess this property to a much greater degree than do mineral oils. Typical examples of such fatty materials used alone as lubricants, or blended in mineral oils, are lard, tallow, rapeseed oil, castor oil and palm oil, and fatty acids such as oleic acid, stearic acid and palmitic acid.

Fatty materials, particularly fatty acids, consist largely of long chain unsymmetrical molecules with non-hydrocarbon end groups. These molecules have a large electric moment and are referred to as "polar molecules", while the non-hydrocarbon end group is termed the "polar end group". These polar end groups, of which carboxyl groups are the most important, are strongly attracted to metal surfaces to which they adhere tenaciously and form adsorbed oil films. The polar molecules orientate themselves rather like the pile on a carpet, with the back cloth representing the metal surface, with one end secured and the other end unattached. A layer of several molecules can build up nose to tail in this manner, but the attraction, or bond, between molecules is apparently less than the bond of the bottom layer to the metal. An oriented adsorbed film of this type offers appreciable resistance to compression but little resistance to motion, as the pile structure is very flexible and the molecules bend over but do not easily break. Sliding friction between metal surfaces is thus greatly reduced.

While the initial attraction of the polar molecules to metals is electrical, it is generally considered that a chemical action then takes place between the adsorbed film and the metal, particularly at high temperatures. Fatty acids react with the majority of metals to form metallic soaps which have high melting points and which are themselves active polar bodies. The bond of polar molecules to metals can therefore be both physical and chemical.

If an adsorbed film is ruptured by metal peaks during sliding, this rupture should only be local and the film should re-establish itself immediately the peaks have disengaged.

It would appear from this that fatty materials would be ideal boundary lubricants but, unfortunately, they oxidize or decompose fairly rapidly, forming objectionable sludge and deposits. They are not suitable, therefore, for use in circulating systems where the lubricant has to remain in service for extended periods. It has been found, however, that by adding fairly small percentages of fatty oils or fatty acids to good quality mineral oils, that these compounded oils are excellent boundary lubricants, while having the added advantage of being much more stable. Typical examples of such oils are compounded steam cylinder oils, used as heavy gear oils, in which about 5 to 10 per cent of a fatty oil such as lard is added to good quality base stocks. Steam engine bearing oils are usually compounded with a fatty oil such as rapeseed and this will withstand quite heavy loading.

Even compounded oils are not so stable as would be liked for circulating oil systems, and a considerable amount of work has been done to develop materials possessing the desired polar molecules without the undesirable poor resistance to oxidation. Esters of fatty acids have proved very promising, being much more stable than fatty acids, and they give excellent results blended in small percentages in mineral oils under quite severe loading.

It is interesting to note that unused mineral oils contain very few polar molecules but as oxidation progresses, polar bodies are formed, so that in effect a used oil possesses superior boundary lubrication properties to a new oil.

Extreme Pressure Lubricants

For extreme cases of boundary lubrication it has been found that even efficient oiliness agents cannot prevent metal-to-metal contact and incipient seizure, so that other forms of boundary lubricants usually termed extreme-pressure (E.P.) lubricants have had to be evolved. These extreme-pressure lubricants form boundary layers under high temperature, high pressure conditions, by chemically combining with the metal surfaces to form thin, solid isotropic non-polar layers. Such films offer little resistance to shear but a high resistance to compression, and as they have considerably higher melting points than polar compounds, they are more suitable for high temperature, high load, slow-speed conditions. Many extreme-pressure agents are chemically inert except at high temperatures and are sometimes described as anti-weld agents. They do not appear to affect materially the oxidation characteristics of the base oil with which they are blended. Typical examples of commonly used extreme-pressure agents are sulphides, chlorides and phosphides which react with metals to form metallic sulphides, chlorides or phosphides. Typical applications for such boundary lubricants are straight cutting oils, wire drawing compounds and hypoid gear oils.

The solid chemical extreme-pressure boundary film, not having the flexible pile-like structure of polar compounds, tends to shear as surface asperities pass over each other, and before the layer can be reformed or healed over, a further chemical action must take place, which is a function of time and the chemical activity of the agent employed. It has also been noted by research workers[5] that in many instances polar type boundary lubricants give lower values of friction than true extreme-pressure agents. This is only true, of course, where conditions are not so severe as to preclude the use of the former type.

It is also worth noting that certain alloys are unreactive with some extreme-pressure agents. A typical example of this is 18/8 stainless steel, which does not appear to react with sulphides or phosphides but is highly responsive to chlorides. This may be due to a protective layer of chromium oxide,

which is inert to sulphur and phosphorus but not to chlorine. The addition of small quantities of copper to this alloy apparently protects the chromium from attack, even by chlorides and hydrochloric acid. The inertness of chromium may well be one of the main reasons why chromium-plated liners greatly reduce Diesel engine cylinder wear, as the metal is highly resistant to attack by sulphur compounds such as sulphur trioxide or sulphuric acid, which are formed during combustion.

On the other hand, certain nickel alloy steels which do not contain chromium do not appear to form boundary lubricating films with chlorinated compounds. The question of selecting a suitable extreme-pressure lubricant for any given set of conditions is, therefore, most complex, and metallurgical properties have to be taken fully into account.

It would appear, therefore, that the ideal lubricant for severe conditions of boundary lubrication should combine the long chain polar compound, with the higher load capacity true extreme-pressure agent. In practice this is usually accomplished by adding separate oiliness and extreme-pressure agents to a suitable mineral base oil and several such lubricants are commercially available.

There are not a great number of applications encountered with marine main and auxiliary machinery which require extreme-pressure agents but there are many instances where lubricants incorporating polar compounds greatly improve lubrication and reduce wear. Some of these applications will be dealt with later in this paper.

Bearing Materials

The subject of bearing metals and their effect on friction, particularly in bearings operating under boundary lubrication conditions, is a most important one which can only be touched on here. A good bearing metal must possess certain characteristics but, unfortunately, some of the requirements are contradictory and a satisfactory compromise must be arrived at. For example, a metal with the lowest frictional characteristics is highly desirable but such a metal, necessarily a soft one, would not possess the requisite load-carrying properties. It is also important that the bearing metal should have good anti-weld properties when used with steel journals. White metal is, of course, the most widely used bearing metal in marine practice, and it is interesting that the alloy used in modern Diesel engine bearings has practically the same composition as that proposed by Babbitt and patented in 1839. Babbitts, or white metals, are divided into two main types, tin base and lead base, the former consisting of a matrix of pure tin with compounds of copper and antimony interspersed, while, in the latter, lead forms the matrix although a smaller amount of tin is also used. A typical good quality tin base white metal has a composition of 90·65 per cent tin, 4·5 per cent copper, 4·5 per cent antimony, and 0·35 per cent lead. Lead is added mainly to improve machineability. A typical lead base alloy consists of 83·5 per cent lead, 15 per cent antimony, 0·5 per cent copper, and 1 per cent tin.

It is quite possible, where unlike metals are used in a bearing combination, that the interposing of an oil film between them which acts as an electrolyte, results in weak electric currents being set up, increasing the polar activities of the carboxyl or similar groups in the oil, thereby forming a tenacious adsorbed film and improved lubrication. If the oil could be maintained completely moisture-free, it would act as an insulating medium, but this cannot occur in practice.

It is also true, of course, that where a relatively soft white metal is used, its melting point is much lower than that of the steel journal, so that under boundary conditions the minute asperities on the bearing surfaces would tend to melt and flow into the adjoining valleys, forming a smooth surface, while similar asperities on the journal would not melt or break off but would probably wear down in time. This could not apply, however, with a steel combination so that inter-surface welding would be more severe.

Other desirable features of bearing alloys are a low modulus of elasticity so that the bearing can absorb small journal

deflexions when under load, a high thermal conductivity so that heat generated by friction can be dissipated rapidly, a high resistance to corrosion while at the same time being chemically active with boundary lubricants.

THE LUBRICATION OF MARINE PRIME MOVERS

In the earlier part of this paper the refining of lubricating oils has been briefly covered, together with comments on some of the more important laboratory tests carried out on lubricants. A section has been devoted to the fundamentals of lubrication and the remainder of the paper will cover more specifically the lubricant requirements of the three basic marine prime movers, namely the reciprocating steam engine, the steam turbine and the oil engine, or the Diesel engine, a name which many regard as a misnomer, but which still persists in spite of efforts to change it.

It is not the purpose of this paper to attempt to cover different types and designs of marine engines except insofar as lubrication problems are concerned. Present-day trends in ships' propelling machinery have been admirably covered by Pounder[6] and it is apparent that the Diesel engine is proving a serious rival to the steam reciprocating engine for small vessels and to a lesser degree the steam turbine for large ships. It may well be that in turn the gas turbine will supplant the Diesel engine.

RECIPROCATING STEAM ENGINES

Steam engine builders have not taken the threat of the Diesel engine lying down. Particularly since about 1925, great improvements in steam engine efficiency have been made possible by the introduction of devices such as the Bauer-Wach exhaust steam turbine, the North-Eastern Marine reheat engine, high-speed enclosed engines such as the Lobnitz, and the use of improved steam valves and valve gear so as to permit the use of high superheat. The majority of marine engineers have a great affection for the old "three legged, up-and-down job" and would be very sorry to see it falling into disuse.

It would appear, however, that in the future the reciprocating steam engine will be used essentially for powers up to about 3,000 i.h.p. for coasters, small tramp steamers and for steam trawlers, where, although less efficient than the Diesel engine, its initial lower cost, simplicity, robustness and ease of maintenance offers great advantages.

STEAM CYLINDER OILS

The lubrication of the internal working parts of reciprocating steam engines presents a rather difficult problem as there are several conflicting factors to be considered. Rubbing speeds as low, so that no matter how efficient the lubricant, it is unlikely that fluid film conditions could exist. The h.p. valve chest and cylinder are exposed to high temperatures, particularly with high superheat steam, while the i.p. and l.p. valves and pistons frequently operate under wet steam conditions, although moisture in wet steam acts as a lubricant to a limited degree. A highly compounded oil gives the most effective lubrication under wet steam conditions as it "wets" the metal surfaces in the presence of moisture much better than a straight oil, but such oils tend to emulsify with water, and surplus cylinder oil carried out of the l.p. cylinder with the exhaust steam is difficult to remove from the condensate. Furthermore, compounded oils tend to carbonize more readily when exposed to high temperatures. On the other hand, if an oil is selected which will give the minimum carbon formation and separate readily from exhaust steam or condensate, it may well be that lubrication of the l.p. valve and cylinder would be poor.

The overriding considerations are that the oil selected should give the most efficient lubrication with the minimum amount used and it should separate readily from the condensate. This is particularly important where small tube watertube boilers are fitted. Many small warships during the recent war were fitted with reciprocating steam engines and Admiralty type three-drum watertube boilers, and a considerable number

of tube failures were traced to the presence of cylinder oil in the boilers. In most cases this was due to overapplication of cylinder oil and inattention to feed water filters, rather than to any shortcomings of the lubricant.

It would thus appear that steam cylinder oils must be a compromise between oils satisfactory for high-temperature dry steam conditions and low-temperature wet steam conditions. From the lubrication point of view, the best solution would be to use different oils for the h.p. cylinder and for the i.p. and l.p. cylinders but this could not be tolerated aboard ship for practical reasons.

A recent advance in cylinder oils has been the introduction of an adhesive additive. This enables the oil to adhere tenaciously to metal surfaces, so that less compounding is required and it enables oil feed rates to be reduced, while still giving adequate lubrication to the working parts.

Steam cylinder oils are much more viscous than Diesel cylinder lubricants, which can be seen by referring back to Table I, and on the face of it this appears to be wrong, bearing in mind that steam temperatures seldom exceed 650 deg. F., whereas peak temperatures in a Diesel engine cylinder can exceed 3,000 deg. F. The reason for this apparent anomaly is, of course, that the Diesel cylinder is effectively water cooled and it is unlikely that the maximum liner surface temperature exceeds about 500 deg. F., while the average temperature is about 350 deg. F., whereas in a steam cylinder everything possible is done to maintain high cylinder wall temperatures so as to prevent wasteful condensation, therefore the average cylinder liner skin temperature in the h.p. cylinder is considerably higher than in the Diesel engine.

Methods of Cylinder Lubrication

The method of application of the lubricant plays a part in the selection of a suitable oil. With old engines using wet steam it was customary to fit tallow cup lubricators on the cylinder and valve chest covers, augmenting this by swabbing the piston rods and valve spindles. This is a "feast or famine" method of lubrication and in many vessels the lubricators are seldom used. The advent of efficient mechanical lubricators enabled oil feeds to be accurately controlled and also ensured a regular oil supply. It is rather surprising, however, to find that in quite a number of cases many of the oil feeds have been disconnected on the grounds that too much oil was being used, instead of a little trouble being taken in adjusting the feed rate.

Mechanical lubricators can be used in conjunction with quills placed in the steam line to atomize the oil, i.e. the oil is broken up into a fine mist by steam passing over the end of the quill, being carried through to the h.p. valve and cylinder in suspension in the steam. Alternatively, direct leads may be taken to the valve faces and cylinder walls. In some cases both methods are employed.

It has been found that unfiltered dark cylinder oils give the best lubrication when atomizer feeds are employed. This is due to the presence of "heavy ends" or high boiling point constituents which are not present in filtered bright stocks. Unfortunately, these constituents contribute largely to the formation of deposits which can cause trouble with sticking piston rings and valves, so that the oil feed has to be most sparing.

Engines Using Superheated Steam

The present upper limit for superheat in marine reciprocating engines is about 650 degrees F. although temperatures as high as 750 degrees F. have been used in industrial engines, a notable example being French express railway locomotives. Under high superheat conditions, exceeding, say, 600 deg. F., a heavy, steam refined, unfiltered, pure cylinder oil is most suitable, both for atomizer or direct feed, as compounded oils leave excessive deposits under such severe conditions. In the majority of engines, superheat is not usually carried past the h.p. cylinder but in engines such as the North Eastern Marine reheat engine the steam remains superheated throughout the engine. In such engines additional oil feeds to the i.p. and h.p. cylinder but in engines such as the North Eastern Marine reheat engine the steam remains superheated throughout the engine.

TABLE II—TYPICAL OILS SUITABLE FOR RECIPROCATING STEAM ENGINES

Grade	Specific gravity at 60 deg. F.	P.M. closed flash, deg. F.	P.M. open flash, deg. F.	Pour point, deg. F.	Redwood No. 1 viscosity (Seconds)			Kinematic viscosity (Centistokes)		V.I.	Carbon residue	Remarks
					70 deg. F.	140 deg. F.	200 deg. F.	100 deg. F.	210 deg. F.			
Steam Cylinder Oils High superheat (above 600 deg. F.)	0·923	540	570	30	—	1,150	260	1,185	50·2	91	2·6	High quality paraffinic unfiltered dark cylinder oil. Suitable for both atomizer and direct feed.
Medium superheat (500–600 deg. F.)	0·920	495	540	15	—	825	180	913·7	35·6	75	1·2	Filtered steam cylinder oil having minimum carbon-forming properties. Separates readily from condensate. Recommended for direct feed applications.
Low superheat and dry saturated steam	0·921	515	555	15	—	920	210	974·5	40·7	83	1·1	Compounded oil prepared from filtered base oils and incorporating lard oil and adhesive agent.
Wet saturated steam	0·909	505	545	40	—	550	150	482·8	30·0	97	2·1	Compounded oil prepared from dark unfiltered base oils and incorporating good quality tallow and adhesive agent.
Bearing Oils Open engine oil	0·940	400	420	0	2,100	209	71	162·6	13·5	82		Conventionally refined base oil with about 10 per cent blown rapeseed oil.
Enclosed crankcase oil	0·878	425	450	20	1,380	170	64	119·2	11·9	99	0·22	High quality solvent refined paraffinic oil containing anti-oxidant.

l.p. cylinder and valve faces may be required.

With lower superheat conditions of about 500-600 deg. F., a somewhat less viscous oil may be employed, an unfiltered straight cylinder oil being preferred for atomizer feed, while a filtered bright stock is recommended for direct feed applications. A good quality lightly compounded cylinder oil, preferably containing an adhesive agent, may also be used under these conditions, giving improved lubrication and enabling the oil feed to be reduced. If an exhaust turbine is installed in engines using superheated steam, a straight oil is preferred to avoid the possibility of turbine blade deposits.

Saturated Steam

In engines employing dry saturated steam, or a mild degree of superheat, a comparatively light cylinder oil may be used and in the majority of cases a compounded oil will give the best results, although here again some engineers prefer a straight oil.

With wet saturated steam, however, a compounded oil is always desirable. Atomizer feed is not generally used under such conditions but direct feed by mechanical lubricator is frequently employed. Provided that the oil supply is kept within reasonable limits, and it must be remembered that due to its superior lubricating properties a compounded oil can be used much more sparingly than a straight oil, no trouble should be experienced with deposits, providing that a high grade base oil is used with good quality lard or tallow as the compounding agent.

Table II gives inspection figures for typical steam cylinder oils for different operating conditions which may be of some interest. Engine oils suitable for use as bearing oils in both open and enclosed engines are also included in this table.

Poppet Valves

Many modern reciprocating steam engines operating on superheated steam are fitted with cam-operated poppet valves, or small matchbox valves, such as the Andrew and Cameron balanced slide valve, which permits rapid opening and closing to steam and exhaust. In some designs return springs are fitted, in others, steam return gear is used. In all designs it is essential that the valve spindles are free to operate in their guides and in certain cases trouble has been experienced due to cylinder oil deposits building up on the stem, causing the valves to stick. Where an oil feed is led to the guide the oil supply should be reduced to a minimum consistent with adequate lubrication. In some vessels a heavy type of machine oil has been used in place of a cylinder oil with very satisfactory results.

BEARING LUBRICATION

Open-fronted Engines

Although the external working parts of reciprocating engines are not subjected to the high temperatures experienced in the cylinders, nevertheless rubbing speeds are usually fairly low while bearing loads may be quite high. Bottom end bearings may be subjected to loads as high as 2,000lb. per sq. in. while top end bearings can be loaded to 3,000lb. per sq. in. Furthermore, all open-fronted engines employ "once-through" lubrication; that is, the oil after passing through the bearings drains to waste. Wick-feed or siphon oilers are generally used and in all cases the oil supply is intermittent. Under such conditions fluid film lubrication cannot exist and it is important, therefore, that the lubricating oil possesses good boundary lubrication characteristics.

The h.p. engine, particularly the top end bearings, is subjected to fairly high temperatures, mainly due to heat soak-back down the piston rod, whereas the l.p. engine may have to contend with a considerable amount of water from leaking glands.

It is important, therefore, that the oil used, besides being a good boundary lubricant, should either resist the washing effect of water or, better still, emulsify with water. Emulsifying agents are generally added to the mineral base oil, which must be of a viscosity high enough to prevent undue leakage at the bearings but not too high so as to cause difficulty in application. The most widely used compounding agent is blown rapeseed oil, which is blended with a suitable mineral oil in the order of 10-20 per cent, the larger amount being required for very wet engines.

During and immediately after the war, when fatty oils were in very short supply, experiments were carried out with synthetic emulsifying agents and a greatly reduced fatty oil content. Lubricants of this type were developed which have proved to be as good as the more conventional bearing oils and they have the advantage of being somewhat cheaper.

The base oils used must be selected for their miscibility with the compounding agent but premium quality oils are not required as the lubricants are in use for such a short period. The finished oils usually have a viscosity of about 180-240 seconds Redwood No. 1 at 140 deg. F. (see Table II) and it will be noted that despite the fact that an inexpensive conventionally refined base oil is used, the viscosity index is high. This is largely due to the presence of a fairly high proportion of fatty oil, as all such oils have high viscosity indexes.

Such an oil, besides being a good boundary lubricant under dry conditions, will also readily emulsify with water. It is therefore suitable for use in a water-in-oil emulsion which is frequently used in eccentric pans or horseshoe thrust blocks in older type engines.

Enclosed-crankcase Engines

The upper speed limit for open-fronted engines is about 150 r.p.m. as above this it is difficult to oil the bearings and guides by hand and it is dangerous to "feel around" the bearings. Although enclosed crankcase engines are not new, they have until recently been confined almost entirely to small high-speed auxiliaries such as fan engines, circulating pump engines and dynamos. The Lobnitz enclosed crankcase engine is a good example of modern main engine design, operating at speeds as high as 475 r.p.m. and being available in sizes up to 3,400 i.h.p.

As will be seen from Fig. 9, the Lobnitz engine is of the triple expansion type, suitable for a high degree of superheat. A piston valve is used for the high-pressure engine and balanced matchbox valves for the intermediate and low-pressure engines. A full oil circulation system is employed, the oil being supplied at a pressure of about 25-40lb. per sq. in. from an oil pump driven from the after end of the crankshaft. A main distribution pipe is led along the back of the engine from which external pipes supply each main bearing. Oil from the main bearings is led through drilled passages in the crankshaft to the bottom ends and eccentric straps. Oil to the top end bearings and guides is supplied through drilled holes in the connecting rods and crosshead pins. Oil for the valve gear bearings is led by external pipes from the eccentric straps. Drains from the bearings return to the sump in the bottom of the crankcase. Suitable oil filters and a small tubular type oil cooler are fitted as standard equipment. An independently-driven auxiliary oil pump is usually supplied so that oil can be circulated through the engine before starting up. In contrast to the lubrication requirements of open engines it is desirable that oils for enclosed engines have a high resistance to emulsification. In spite of the fact that wiper glands are fitted on the piston rods and valve spindles, it is difficult to exclude entirely water from the crankcase. Any water which does find its way into the sump must separate readily from the oil so that it can be drained off at suitable intervals.

Again, the oil in enclosed engines must remain in service for long periods and it must, therefore, be highly resistant to oxidation and sludging. A better quality oil than that used in open engines is required and an anti-oxidant is highly desirable.

As speeds are usually higher in enclosed engines and there is a constant supply of oil to the bearings, conditions are more favourable for the establishment of fluid film lubrication. It is possible, therefore, to use a comparatively light straight mineral oil, as will be seen in Table II.

It is important that both steam glands and crankcase-top wiper glands should be kept tight to avoid entry of water into the sump. Swabbing of the piston rods and valve spindles should be reduced to a minimum and where oil drains from save-alls are fitted, any oil drained off should not be returned to the sump as it usually contains a large proportion of cylinder oil. If a straight mineral cylinder oil is in use this would increase the viscosity of the crankcase oil and if a compounded cylinder oil is used this would greatly reduce the demulsification properties of the sump oil.

EXHAUST STEAM TURBINES

Low pressure exhaust steam turbines are not, of course, a recent development. Sir Charles Parsons introduced the idea nearly half a century ago for triple screw vessels, where the wing shafts were driven by reciprocating steam engines which exhausted into a low pressure turbine directly coupled to the centre shaft. As this arrangement was only suitable for large vessels, and efficient reduction gearing was not available

so that the slow-speed direct coupled turbine was large and cumbersome, the system fell into disuse.

Bauer-Wach System

With the introduction of the Bauer-Wach system in 1926, in which the exhaust turbine is coupled through reduction gearing and hydraulic clutch to the reciprocating engine shaft, the combination of steam engine and steam turbine became a practical proposition even for low-powered vessels. It is claimed that up to 33 per cent increased power can be obtained with the same fuel consumption or, alternatively, up to 25 per cent full economy can be achieved for the same power. Modern reciprocating engines employing superheated steam and exhaust steam turbine are competitive in overall fuel consumption with an equivalent steam turbine up to about 5,000 i.h.p.

Although the Bauer-Wach turbine, in common with other similar installations, necessarily complicates an otherwise simple steam engine, the controls are extremely simple and reliable. They have proved extremely successful in deep sea trawlers
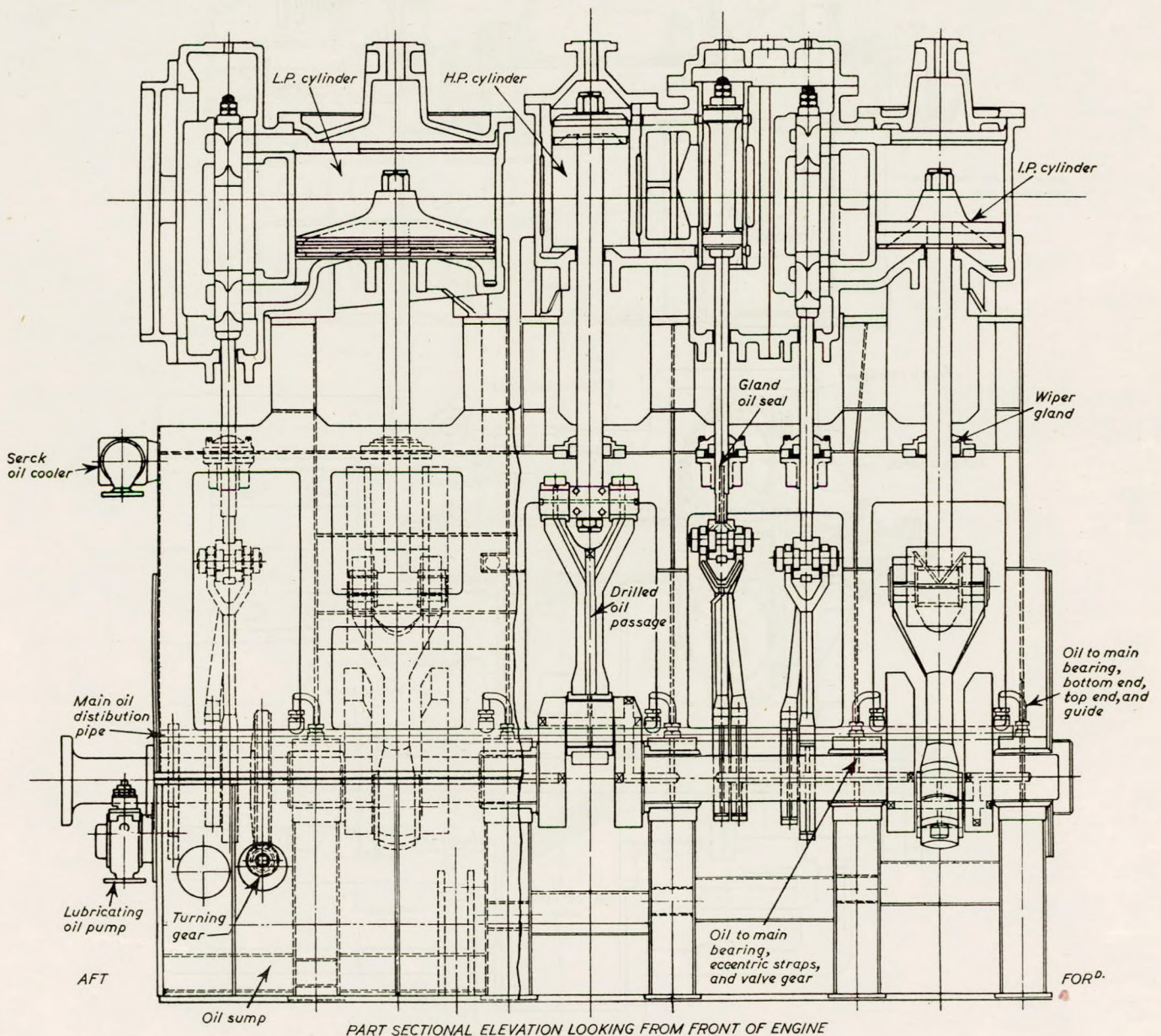

PART SECTIONAL ELEVATION LOOKING FROM FRONT OF ENGINE

FIG. 9—*Totally-enclosed Lobnitz reciprocating steam engine*

which operate with two engine-men, who are not usually quali-
fied engineers.

A longitudinal section and a plan view of a typical instal-
lation are shown in Fig. 10. The turbine is directly coupled
to the primary reduction pinion, which engages the primary
gearwheel mounted on the driving member of a Vulcan-Sinclair
hydraulic coupling. The driven member is coupled to a
secondary reduction pinion which meshes with the main gear-
wheel mounted on the main shaft, immediately aft of the main
Michell thrust block. A view of the gears with the primary
pinion and upper casing removed is seen in Fig. 11. Both
sets of reduction gears are of the single helical type but are
opposite handed. As there is no mechanical connexion between
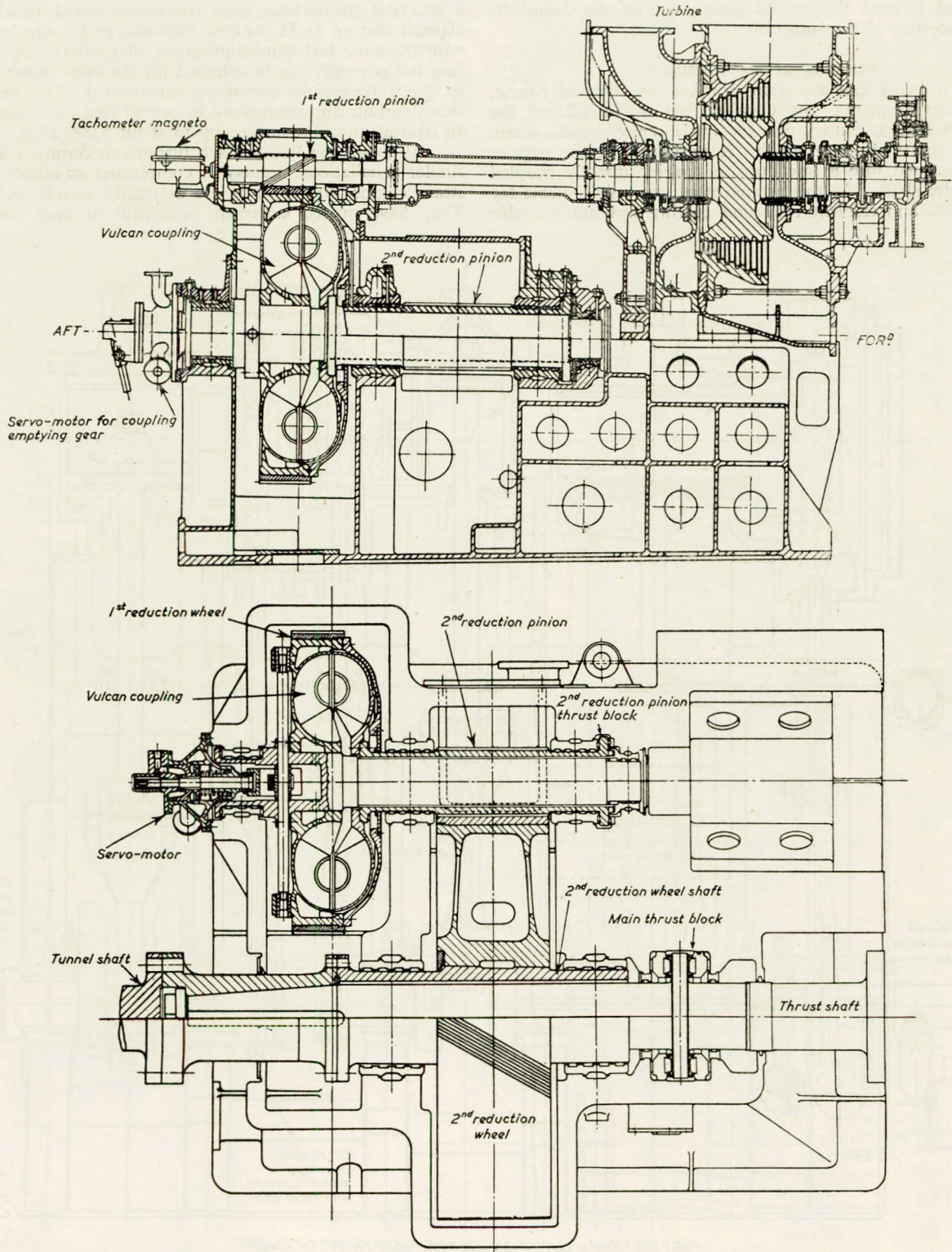the two halves of the coupling, it is necessary to locate both

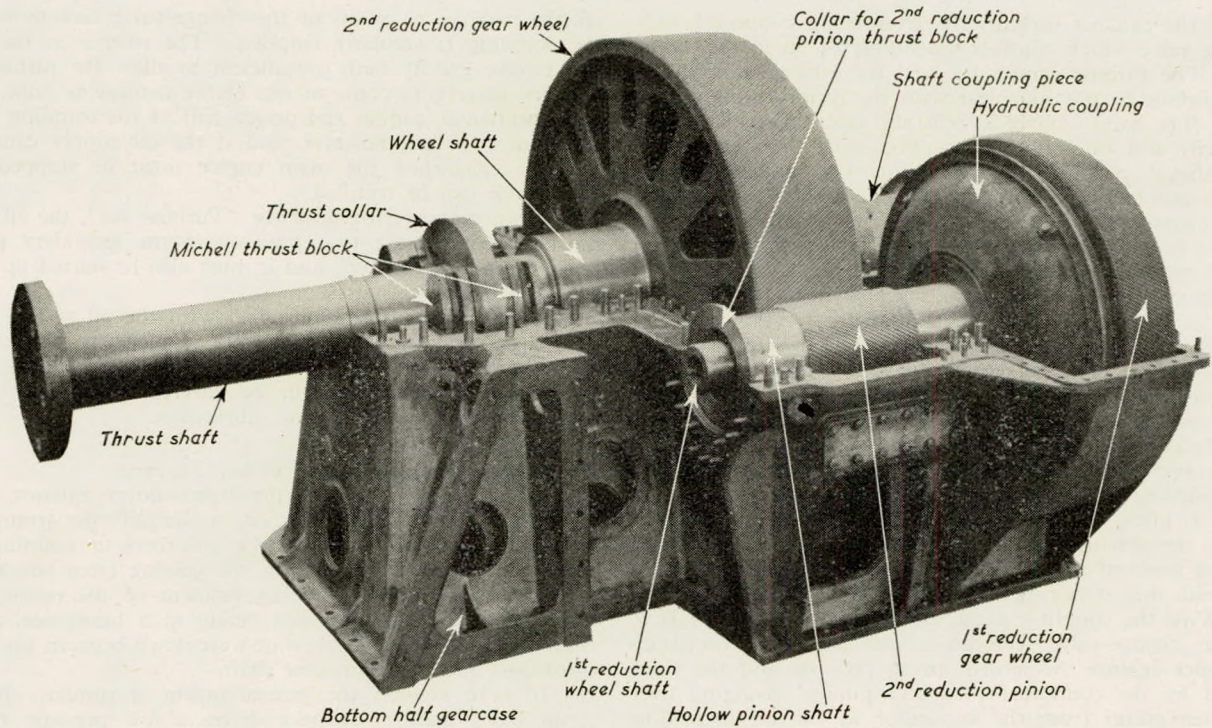FIG. 10—*Longitudinal section and plan view of Bauer-Wach exhaust steam turbine*

2nd reduction gear wheel

Collar for 2nd reduction pinion thrust block

Shaft coupling piece

Hydraulic coupling

Wheel shaft

Thrust collar

Michell thrust block

Thrust shaft

1st reduction wheel shaft

Bottom half gearcase

Hollow pinion shaft

1st reduction gear wheel

2nd reduction pinion

FIG. 11—*Reduction gears for Bauer-Wach turbine drive*

the primary and secondary pinions and the primary gearwheel by individual thrust bearings, while the main wheel is located by the main thrust block.

Lubricating Oil System

The turbine is fitted with a self-contained lubricating oil system, as will be seen in Fig. 12. Duplicate reciprocating oil pumps are installed, either of which is capable of supplying the system. In addition to lubricating the bearings and gears, the oil is used as a hydraulic medium to operate servomotors which control the exhaust steam changeover valve and also the hydraulic coupling. A single lever is used to engage or
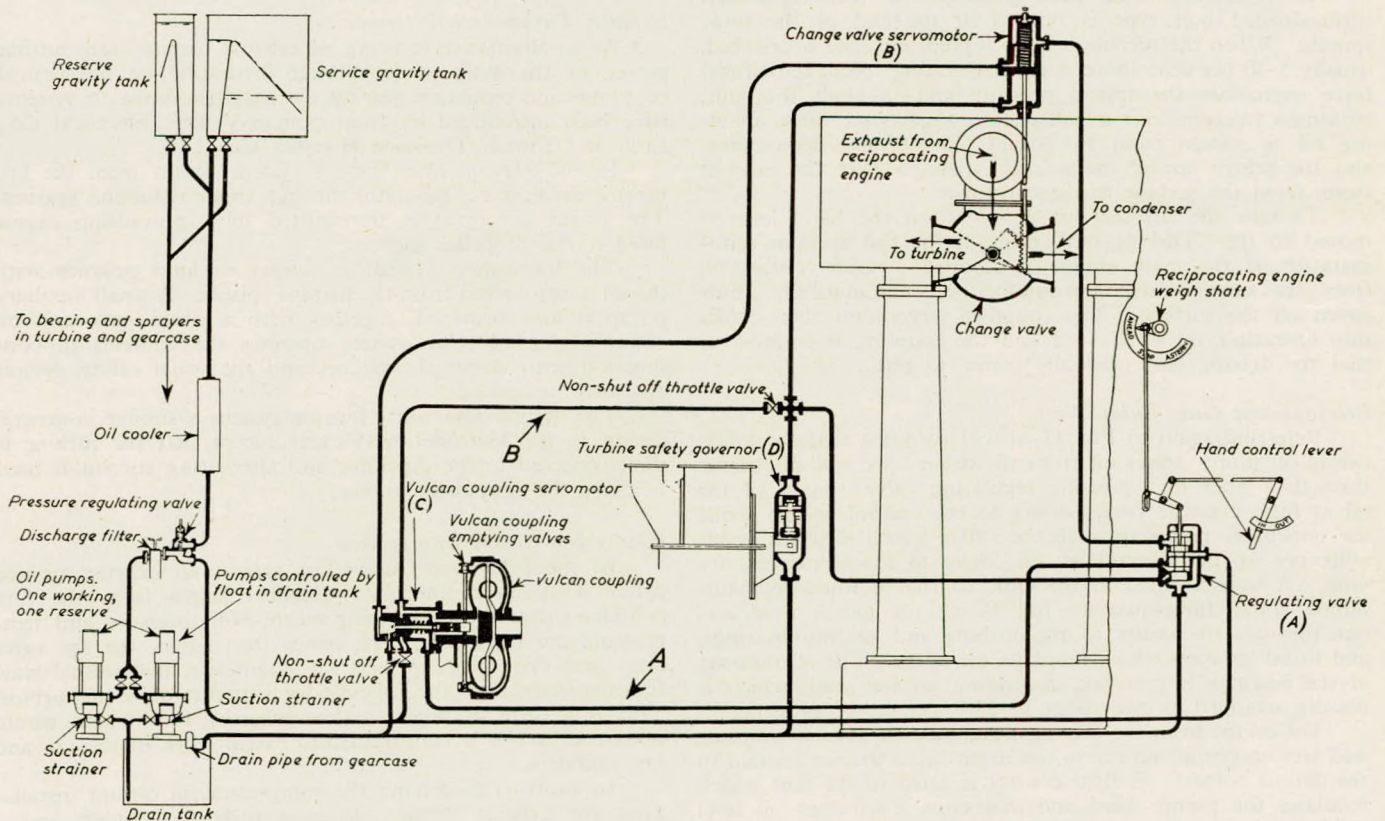

Reserve gravity tank

Service gravity tank

To bearing and sprayers in turbine and gearcase

Oil cooler

Pressure regulating valve

Discharge filter

Oil pumps. One working, one reserve

Pumps controlled by float in drain tank

Non-shut off throttle valve

Suction strainer

Suction strainer

Drain pipe from gearcase

Drain tank

Change valve servomotor (B)

Exhaust from reciprocating engine

To condenser

To turbine

Reciprocating engine weigh shaft

Change valve

Non-shut off throttle valve

B

Turbine safety governor (D)

Vulcan coupling servomotor (C)

Vulcan coupling emptying valves

Vulcan coupling

Hand control lever

Regulating valve (A)

A

FIG. 12—*Lubrication and oil control systems of Bauer-Wach turbine*

disengage the exhaust turbine, the lever being connected to a regulating valve which controls the supply of oil to the servomotors. The turbine cannot be used for astern running nor can the turbine be put into gear with the reciprocating engine stopped. The hand control lever is also coupled to the engine weigh shaft and can only be operated with the weigh shaft in the "ahead" position. In the event of a sudden stop the turbine is automatically thrown out of gear.

To engage the turbine with the main engine running ahead, the control lever is moved to the "Turbine in" position and high-pressure oil from the lubricating oil system passes through ports in the control valve, via pipe A, to the hydraulic coupling servomotor where the oil pressure forces the piston to the left. A rack coupled to the piston rod turns a pinion which closes the coupling emptying valves, and at the same time high-pressure oil direct from the pump discharge is admitted to the hollow shaft of the coupling. The driven half of the coupling, being mechanically connected to the main shaft, revolves with the main engine, but until oil is admitted to the coupling the driving half remains stationary. As the coupling is filled, centrifugal force circulates the oil from the driven to the driving half which commences to rotate due to the torque moment of the circulating oil and gradually comes up to speed, thus motoring the primary pinion and the turbine rotor. With the coupling filled, oil now passes via pipe B to the steam change-valve servomotor, the piston is forced up the cylinder against the return spring pressure and the valve is moved by the connected rack and pinion, changing over the exhaust steam from the condenser to the turbine. The turbine immediately picks up speed above the motoring speed, and in turn commences to drive the primary pinion and gearwheel. The primary half of the coupling, now rotating at a higher speed, takes over and drives the secondary half of the coupling which transmits the drive through the secondary pinion and main gearwheel to the main shaft. The shaft revolutions increase some 12-15 per cent above those obtained with the main engine alone.

An emergency over-speed governor of the unbalanced spring-loaded bolt type is housed in the end of the rotor spindle. When the turbine reaches a predetermined over-speed, usually 5-10 per cent above normal operating speed, centrifugal force overcomes the spring pressure and the bolt flies out, striking a fulcrum lever which releases a governor valve, allowing oil to escape from the steam change-valve servomotor, and the return spring immediately changes over the exhaust steam from the turbine to the condenser.

To take the turbine out of operation the hand lever is moved to the "Turbine out" position (or this is done automatically if the main engine is stopped), which releases oil from the change-valve servomotor and automatically shuts steam off the turbine. The coupling servomotor then comes into operation, oil is shut off and the coupling is emptied so that the driving half gradually comes to rest.

Bearings and Gear Lubrication

Referring again to Fig. 12, it will be noted that the lubricating oil pump draws oil from the drain tank and discharges through a filter to a pressure regulating valve. Some of the oil at high pressure passes direct to the control system while the remainder passes through the valve, which is set at about 50lb. per sq. in., through an oil cooler to the service gravity tank. A weir is fitted in the tank, so that it must be maintained about three-quarters full before oil passes over and out through the outlet to the turbine and gearing bearings, and to oil sprayers which lubricate the gears. The oil pressure at the bearings is constant, depending on the head, which is usually arranged to give about 10-15lb. per sq. in. pressure.

Oil drains from the bearings, gear case, hydraulic coupling and servomotor all return to the drain tank, usually located in the double bottom. A float control is fitted to the tank which regulates the pump speed and maintains a constant oil level in the tanks.

In the event of oil failure, steam is automatically shut

off the turbine by means of the change-valve servomotor and the coupling is similarly emptied. The reserve in the main and reserve gravity tank is sufficient to allow the turbine and primary gearing to come to rest before damage is done. The main gearwheel, pinion and driven half of the coupling would continue to rotate, however, and if the oil supply cannot be quickly replenished the main engine must be stopped until the trouble can be rectified.

When manœuvring with the "Turbine out", the oil pump must be kept in use to supply oil to the secondary gearing bearings and oil sprayers, and it must also be started up before the main engine is started.

A small centrifuge is installed with the oil system and it is essential that this should be used regularly to maintain the oil in good condition and remove water. The question of turbine oil maintenance will be covered more fully in the next section on steam turbine lubrication.

Parsons and Brown-Boveri Exhaust Turbines

In both the Parsons and the Brown-Boveri exhaust turbine systems an attempt has been made to simplify the arrangement by substituting mechanical shock absorbers or couplings for the Vulcan coupling to protect the gearing from shock loading due to the uneven turning moment of the reciprocating engine, or from the propeller racing in a heavy sea, and to allow for minor misalignment or wear down between the engine crankshaft and the main gear shaft.

In both systems the general layout is similar. Exhaust steam from the l.p. cylinder drives a low pressure turbine, the power being transmitted to the main shaft through double reduction gearing. Parsons employ double helical gearing whereas Brown-Boveri fit opposite-handed single helical gears.

Forced lubrication systems are employed, with servomotor-operated controls employing the lubricating oil as a hydraulic medium. A gravity tank system supplies oil to the bearings and gears, as previously described for the Bauer-Wach installation.

Exhaust Turbo-electric Drives

As an alternative to using an exhaust turbine transmitting power to the main shaft through hydraulic or mechanical couplings and reduction gearing, electrical transmission systems have been introduced by Metropolitan-Vickers Electrical Co., Ltd., and British Thomson-Houston Co., Ltd.

In the Metropolitan-Vickers system, steam from the l.p. turbine drives a d.c. generator through single reduction gearing. The power generated is transmitted to a propulsion motor fitted to the propeller shafting.

The lubrication system is similar to land practice with the oil pump driven from the turbine spindle. A small auxiliary pump is also provided, together with a drain tank and oil coolers. An oil relay system operates the controls in conjunction with electrical switches and the usual safety devices are incorporated.

The British Thomson-Houston system is similar in general design to the Metropolitan-Vickers except that the turbine is direct coupled to the generator and alternating current is used to drive the propulsion motor.

Götaverken Steam Compressor

In the Johansson-Götaverken system an exhaust turbine drives a steam compressor. Exhaust steam from the h.p. cylinder enters the compressor where both pressure and temperature are increased. The steam then enters the i.p. valve chest and drives the i.p. and l.p. engines in the normal way. Exhaust steam from the l.p. cylinder is used to drive the turbine. This is a form of reheat system whereby heat which would otherwise be lost is converted into useful work in the i.p. and l.p. cylinders.

In addition to driving the compressor, in certain installations the exhaust turbine drives a turbo-feed pump and a dynamo from extensions of the spindle. In all cases, however, the turbine is directly connected to the compressor.

The system is probably the most simple of all exhaust turbine installations as there is a complete absence of reduction gears, couplings or electrical transmissions.

The lubrication system is also simple, the oil pump being driven from the turbine spindle. The exhaust changeover valve is kept open to the turbine by the oil pressure. If this fails the exhaust passes direct to the condenser and the turbine stops. The usual type of overspeed trip is also fitted to the turbine spindle.

Exhaust Turbine Oils

As turbine oils will be discussed more fully later, it is only necessary at this stage to point out the necessity of employing high quality, straight-mineral turbine oils, preferably containing both an anti-oxidant and a rust preventive, for all types of exhaust turbines. With direct coupled systems and single reduction units a heavy grade of oil is recommended, whereas for double reduction gears an extra heavy grade is preferred.

There is one important point to bear in mind in installations combining open-fronted reciprocating engines with exhaust turbines and this is to take every precaution to avoid mixing the turbine oil with the compound engine oil or steam cylinder oil. In many vessels where the oil is stored in bulk tanks in the engine room there is only a single filling line for all tanks from the deck. If a compounded oil has been passed through the line and is followed by the turbine oil, a considerable amount of contamination of the latter can occur. Such contamination can seriously affect the demulsibility properties of the turbine oil and also reduce its oxidation-resisting properties. Where such filling systems are installed it is preferable to store the turbine make-up oil in barrels rather than risk contamination.

STEAM TURBINE LUBRICATION

Geared marine turbines have for many years enjoyed a high reputation for reliability and simplicity. They pack the maximum of power into the minimum space and are, therefore, particularly suitable for major warships and large liners. Competition from the Diesel engine has spurred on designers to improve thermal efficiencies, and during recent years high-pressure watertube boilers generating high-temperature steam have been widely used in liners, cargo liners and large oil tankers. Gear efficiencies have been improved and tooth loading stepped up so that shorter and lighter gears and gear cases can now be used. Locked train double reduction gears have proved very successful in warships and will no doubt be used extensively in merchant vessels in the future. Overall fuel consumption figures as low as 0·55lb. per s.h.p. per hr. are now quite feasible, as compared with about 0·40lb. per s.h.p. per hr. for a similar Diesel-engined installation.

Marine Turbine Oils

Turbine oils used in marine geared turbines have a more arduous duty to perform than in large direct coupled land turbo-alternators, as in addition to lubricating and cooling the bearings they must also lubricate the gears. Unfortunately, the requirements of bearings and gears are most dissimilar. The steam turbine represents one of the best examples of machinery operating under fluid film conditions. Speeds are high and bearing loads low, so that a low viscosity oil which reduces fluid friction to a minimum is the most suitable lubricant. Furthermore, with present high steam temperatures, cooling of the bearings, particularly at the high pressure end of the turbine, is becoming increasingly important, and the lower the viscosity the more efficient is the cooling. On the other hand, particularly with modern gear sets employing high tooth loading, a low viscosity oil would not provide adequate lubrication of the gears, and wear would be excessive. A high viscosity oil is therefore preferable for gear lubrication.

It would appear, therefore, that the most satisfactory solution from the oil point of view would be to employ two separate systems, with the turbine system using a low viscosity oil and the gear system a much more viscous oil. Serious consideration has been given to this by engine builders but the idea has always been rejected on the grounds of additional complication, increased weight (particularly important in warships) and higher initial cost.

Another way out of the difficulty would be to employ low viscosity oils incorporating oiliness, or mild extreme-pressure additives, but unfortunately none of the normal compounded oils possess the oxidation resistance necessary in turbine oils which must remain in service for many years. At the present time investigations are being carried out to see if improved load-carrying agents can be developed which will themselves be stable and which will not affect the stability of the base oil. Results of tests so far carried out are inconclusive, particularly as the only satisfactory way to test such oils is in the actual equipment under running conditions, and this is obviously a long-term project. At present, therefore, marine turbine oils must be a compromise between an oil sufficiently viscous to lubricate adequately the gears and a lighter oil suitable for the turbine bearings.

In addition to having adequate load-carrying properties, the other essential requirements in marine turbine oils are:—
(1) A high resistance to oxidation.
(2) Good demulsibility characteristics and anti-foaming properties.
(3) Good anti-corrosion properties.

Oxidation Resistance

In service, turbine oils, in common with all petroleum products, undergo chemical change due mainly to oxidation, with the subsequent formation of both soluble and insoluble oxidation products. The oil-soluble products eventually form weak oxyacids while the insoluble products form complex solid or semi-solid sludge, resins and varnishes, all of which are highly undesirable.

It has been found that paraffinic and naphthenic hydrocarbon components of lubricating oils when oxidized tend to form oil-soluble compounds of an acidic nature[7] while oxidation of aromatic constituents tend to form oil-insoluble sludges and varnishes. As has already been discussed, the solvent refining process reduces aromatic constituents to a minimum and, therefore, greatly reduces sludge-forming tendencies in an oil. It is generally accepted that high viscosity index paraffinic oils possess the greatest natural resistance to oxidation, but nevertheless, regardless of the method or degree of refining, the type of crude from which the oils are refined or the addition of oxidation inhibitors, oxidation will take place to a greater or lesser degree. Oxidation can be retarded but cannot be entirely stopped. The important thing is that by carefully refining suitable base oils and incorporating powerful oxidation inhibitors, the rate of oxidation can be retarded so that it presents little danger.

The major factors in promoting oxidation are the presence of oxygen, heat, and the catalytic effect of certain metals. It is impossible to prevent air from coming in contact with oil in the system but by suitable design this can be reduced to a minimum.

Modern turbines employing high steam temperatures have greatly increased the risk of oxidation. Oxidation is a chemical reaction and it has been shown that its rate is doubled for about every 18 deg. F. rise in temperature, but below about 170 deg. F. the rate of oxidation of a high quality oil is negligible. Although the observed temperature of oil leaving the bearings seldom exceeds 170 deg. F., the molecules of oil in contact with the journal at the high pressure end of a turbine may be subjected to temperatures as high as 400-450 deg. F., due to heat soak-back along the rotor spindle but, of course, these represent only a small proportion of the oil flowing through the bearing at any given time. Possibly the most severe oxidizing conditions exist when the turbine is being warmed through in turning gear, when only boundary lubrication conditions can exist, due to the low speed of rotation.

The most active catalysts (a catalyst is a substance which

although remaining unchanged itself, by its presence, affects or promotes a chemical action in other substances) found in turbine lubricating oil systems are copper and finely divided iron oxides, the latter frequently being present in semi-colloidal form in the oil. Copper piping is gradually being eliminated in modern turbine installations, mainly because solid drawn steel piping is stronger and this is also an advantage from the catalyst point of view.

Until the early 1930's practically all turbine oils were produced by carefully refining paraffinic distillates by the acid treatment method. Such oils, a typical good example being Admiralty S.M.L.O., have been in service for many years and have given excellent results. Operating conditions in modern turbines, however, are becoming increasingly severe. In addition to increased steam pressures and temperatures already mentioned, other factors include the tendency to have less oil in the circulation system, partly to save space and weight and partly because in large liners there is frequently insufficient headroom to allow the use of the gravity tank system. With less oil in the system it must be circulated more rapidly so that it is exposed to heat more frequently and aeration is increased. Heavy loading of modern gear sets also increases the possibility of oxidation.

Experience gained by oil technologists with improved lubricants in large high-pressure, high-temperature land turbo-alternators has been put to advantage in producing better marine turbine oils. The solvent extraction method of refining was perfected, which greatly reduced sludge formation in turbine oils, due to the almost complete elimination of unsaturated constituents. It was found, however, that over-extraction, besides removing the undesirable constituents, also removed certain natural oxidation inhibitors, and the oils lost some of their natural resistance to the formation of soluble acidic decomposition products. Although the oils could be used for longer periods before acidity increased, once it did commence it progressed fairly rapidly. As a result of this, after intensive research, chemicals were developed, which when added to good quality solvent refined turbine oils, retarded oxidation almost indefinitely under normal operating conditions. These chemicals were termed "anti-oxidants" or "oxidation inhibitors".

Anti-oxidants vary considerably in composition and in their effectiveness with different base oils, and as a general rule, should be developed to suit any particular base oil concerned. The action of anti-oxidants is not fully understood, but it would appear that they act as chain breakers or de-activators and are thus gradually consumed in service but they are replaced to some degree in fresh make-up oil. It is essential that the additive is not readily decomposed by heat, nor must it be water soluble as it is impossible to exclude entirely water from turbine oil systems.

Demulsibility

Good demulsibility characteristics are particularly important in turbine oils especially in modern sets with limited oil capacity. As water leakage into the system is an ever-present danger, it is essential that if it does occur, separation should be as rapid as possible. Solvent refined good quality oils have excellent demulsibility properties and these are practically unaffected by most anti-oxidants. Certain corrosion inhibitors can affect laboratory demulsibility test results but do not appear to affect adversely the oil in service.

Foaming

Foaming of turbine oils is somewhat similar to emulsification, except that air replaces water as the dispersed phase, oil being the continuous phase in each case. Cases of severe foaming are rare and where it has occurred has usually been traced to a mechanical fault allowing excessive local aeration. Under normal conditions it is impossible to avoid a certain degree of aeration, particularly with reduction gear oil sprayers, but with a suitably designed system and clean oil, entrained air will separate readily from the oil, especially in the drain tank. Foaming is objectionable on two main grounds; if excessive it will promote oxidation and where oil is used as a hydraulic medium in control systems and fluid couplings, as in the Bauer-Wach exhaust turbine, the entrained air can cause erratic operation.

As has been discussed earlier, oils are much more susceptible to emulsification when oxidized and this applies equally to foaming. This is another good reason for employing oils with the maximum stability. Entrained solids will also adversely affect both emulsification and foaming, apart from being injurious to finely finished working parts, and should be eliminated as far as possible.

Corrosion of Turbine Systems

At first glance, the likelihood of corrosion in turbine lubricating oil systems would appear to be improbable. On closer examination of conditions, however, it will be seen that there is every possibility that it will occur to some degree. There is always a possibility of salt water entering the system from a leaking oil cooler tube or possibly a leak in the tank top where the drain tank is located in the ship's double bottom. It is usual, of course, for the lubricating oil pressure to be higher than the cooler circulating water pressure so that any leakage should be from oil to water, but many engineers start circulating water through the cooler before starting the lubricating oil pump and, when shutting down, stop the oil pump first. Under these conditions a leaking tube would allow water to enter the oil system.

There is also the possibility of condensed steam entering the system from leaking turbine glands and condensation of moisture from the atmosphere can be troublesome, particularly during cold weather. The latter causes of water entry are common but luckily salt water leaks are more rare.

Assuming then that moisture is always present to some degree in the turbine oil system, its effects will be considered. Corrosion has been found to be particularly severe in areas which are not constantly oil bathed, such as the upper parts of gear cases and oil reservoirs. This type of corrosion, sometimes termed primary corrosion, is mainly due to condensation. Drops of moisture accumulate on the metal surfaces, where, in the presence of air, rusting occurs. The droplets increase in size and finally drop into the oil, carrying with them finely divided rust particles. These particles are of two types, namely, ferric oxide (Fe_2O_3) and ferrous oxide (FeO). Ferric oxide is an active catalyst while ferrous oxide particles tend to coalesce and can damage bearings and gears. Ferrous oxide is magnetic and can readily be removed by means of magnetic strainers.

Secondary corrosion takes place much more slowly and is caused by the formation of weak water-soluble organic acids during oxidation of the lubricant. Acidic vapours above oil level can rapidly increase the corrosion already started by condensation. Corrosive attack can also take place below oil level and is particularly severe on machined surfaces such as journals and gear teeth. Although the most severe corrosion is believed to be caused by secondary corrosion, primary corrosion could also play a part.

Primary corrosion is entirely independent of oil oxidation and can be severe in completely new engines but secondary corrosion is a direct result of oil deterioration.

There are several ways whereby corrosion can be reduced to a minimum. Engine builders can help by designing systems wherein the possibility of water leakage is reduced to a minimum, while efficient ventilation of oil reservoirs and gear cases reduces condensation. Ships' staffs can help by keeping the surrounding atmosphere as dry as possible by careful attention to steam leaks of all types, and also by the regular use of the centrifuge to keep the oil moisture content to a minimum.

Oil technologists have helped materially by producing oils which greatly reduce the possibility of oxidation and, therefore, the formation of acidic materials. Although this reduces the risk of secondary corrosion, the problem of primary corrosion remains.

A desirable feature of any turbine oil, particularly in the

presence of moisture, is that it should possess good metal-wetting properties, but unfortunately it is the presence of polar bodies in a mineral oil which imparts such properties and highly refined turbine oils are entirely free from such matter. It is perhaps true that some of the undesirable constituents, from the oxidation point of view, which are removed by solvent refining, acted to some extent as rust inhibitors. Acid refined oils could therefore be expected to have better metal-protective properties and this is borne out by experience. Another reason for the superior rust prevention properties of the latter type of oil could be that initial oxidation, and subsequent formation of polar molecules, is much more rapid. It would appear, therefore, that by perfecting oils which remain highly stable in service the formation of a protective coating on metal surfaces in the oil system is being prevented and the possibility of primary corrosion will be increased.

Solvent refining of turbine oils originated in the U.S.A. and they were developed essentially to meet the increasingly severe demands of large power station turbo-alternators. These oils were introduced in the mid-1930's and after a few years in operation complaints were received, particularly from operators of new machines, that severe rusting was being experienced. The Westinghouse Electric Corporation[8] were one of the first turbine builders to investigate this problem and they evolved a laboratory method of assessing the anti-rusting properties of turbine oils. This was later adopted by the American Society of Testing Materials as Test D665-42T (later amended to D665-44T) and consisted essentially of stirring a sample of the test oil at room temperature in a beaker in which a flat, polished, steel specimen was suspended. After ten minutes stirring, 1 per cent of distilled water was added and stirring continued for eight hours per day for five days, after which the specimen was removed and examined for rusting.

The major oil companies were concurrently examining the problem and it was soon found that by adding small quantities of various polar bodies that the oils could pass the A.S.T.M. test. Unfortunately, however, most of the materials of this type adversely affected the demulsibility properties of the oil. After intensive research, new types of corrosion preventives were developed, which, when added to turbine oils in very small quantities, plated out gradually on metal surfaces to form thin protective coatings on oil-bathed surfaces, while at the same time the demulsibility properties were not affected to any degree.

A solvent refined oil containing an anti-oxidant, having a demulsification number (I.P. 19/47) of about $1\frac{1}{2}$ minutes, could have the number increased to about five minutes when a rust-preventive additive of the type described was added. In practice, however, no difference in demulsibility characteristics would be detected.

Oils containing an anti-oxidant and a rust inhibitor are usually referred to as "double-inhibited oils" and are, of course, in widespread use at the present time, both ashore and afloat.

With the introduction of double-inhibited oils the problem of both primary and secondary corrosion of oil-bathed surfaces had been virtually solved for industrial turbines, but the problem of primary corrosion of non-oil-bathed surfaces inside the oil system still remained. This aspect will be discussed later.

Following the success of double inhibited oils in the U.S.A., their use was extended to this country and they are, of course, now widely accepted, particularly for power station turbines. The United States Navy also showed a marked interest in solvent-refined oils as long ago as 1938 but they also experienced several cases of corrosion with them. Even with rust-preventive additives present, which were successful for land machines, trouble was still experienced but to a lesser degree. The difference in performance was attributed to the presence of salt water in the oil system, which is not, of course, common in land turbines. The standard specification for naval turbine oils at that time was known as Symbol 2190 which did not include a salt water corrosion test.

Salt water has been proved to be much more highly corrosive than fresh or distilled water, possibly due to the forma-tion of hydrochloric acid by hydrolysis under high temperature conditions.

The United States Navy, recognizing that salt water was largely responsible for corrosion troubles experienced with journals, gears and other parts of the oil system, altered their specification for turbine oils in August 1945 and the new U.S.N. Department Specification 14-0-15 for Turbine Lubricating Oil, Non-Corrosive (Symbol 2190-T) was introduced. This included a salt water corrosion test similar to A.S.T.M. D665-44T except that 10 per cent of synthetic sea water having a fixed composition was used in place of distilled water. D665-44T differed from the original D665-42T insofar as a 10 per cent solution of distilled water was used in place of 1 per cent and the stirring speed of 1,000 r.p.m., ± 50 r.p.m., was carried out at 140 deg. F. in place of room temperature.

Towards the end of the war the British Admiralty became increasingly interested in inhibited turbine oils for naval vessels and introduced a new specification known as O.M.88. Oils meeting the new, more rigid specification replaced the older type S.M.L.O. turbine oils used for so many years. This new specification was later amended somewhat and the current specification known as O.M.100 was introduced in November 1951. This specification is interchangeable with U.S.N. Symbol 2190-T.

It must not be thought, however, that the major oil companies were remote from the new developments; in fact their representatives sit on both the Royal Navy and the United States Navy turbine oil panels. These specifications merely lay down the requirements for naval turbine oils; it is the responsibility of the oil suppliers to develop oils which will meet the specifications. Commercially available, branded, marine turbine oils are equal to the new specifications and in certain cases exceed their requirements from the oxidation stability, demulsibility requirements and anti-corrosion property points of view.

Admiralty research departments have investigated a number of serious cases of corrosion caused by the ingress of excessive amounts of salt water into turbine oil systems in naval vessels, due in some cases to enemy action, and in others to lack of maintenance and severe operating conditions in war time. They developed a method[9] of counteracting salt water under such extreme conditions, employing sodium nitrite in aqueous solution as an inhibitor. The results obtained are reported to be very successful, but it must be appreciated that the conditions under which it is used are abnormal, and the treatment must be carefully handled. Too weak a solution, instead of acting as a corrosion inhibitor, can, in certain circumstances, actively promote local corrosion.

Under normal operating conditions, with a clean oil in use, limited amounts of both distilled and salt water should rapidly separate out but this does stress the importance of using an efficient centrifuge regularly.

Protective Coatings

Appreciating the fact that while efficient double-inhibited solvent refined oils greatly reduced the possibility of corrosion below oil-wetted surfaces, the problem of primary corrosion in other parts of the system still remained, one leading oil company developed a special oil-resisting, acid-resisting and water-resisting protective paint to protect the system. Oil-resistant paints were not, of course, new, but many such paints had certain defects. While resisting new oils they were not so effective against organic acids formed by oil oxidation. Furthermore, if the paint were not correctly applied, or if the metal surface were not thoroughly cleaned, it tended to peel off in sheets and in extreme cases could cause trouble with choked oil filters.

The new type of paint, consisting essentially of an aluminium base bonded with special synthetic resins and incorporating petroleum solvents, was readily applied by brush or spray gun, and in addition to excellent protective properties, had the added advantage that if applied to a dirty surface it would not peel off. If any of the paint left the metal it was
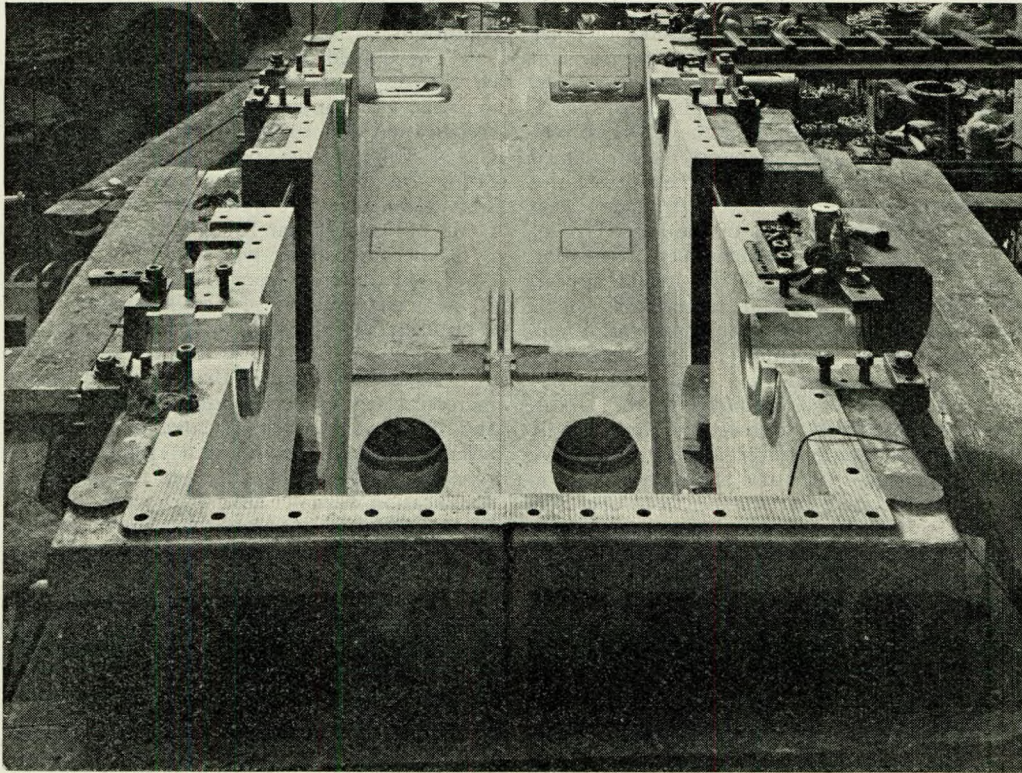
FIG. 13—*Lower half of gear case of a modern cargo liner treated with special protective paint*

in the form of tiny flakes which were too fine to choke filters or damage bearings or gears. An example of a single reduction gear case treated with this paint is shown in Fig. 13, while Fig. 14 shows the main gearwheel with the paint applied to the entire surface except the journals, flanges and teeth. The paint is also applied to the interior of the bearing housings, oil drain tank and gravity feed tank. It is not possible to treat the inside of oil pipes owing to the difficulty of obtaining

TABLE III—INSPECTIONS OF MODERN MARINE TURBINE OILS

Test	Inst. of Petroleum test no.*	Commercial branded oils		Typical Admiralty O.M. 100
		S/R. geared turbine	D/R. geared turbine	
Method of refining	—	Solvent	Solvent	Solvent
Type of additives	—	Anti-oxidant corrosion-inhibitor	Anti-oxidant corrosion-inhibitor	Anti-oxidant corrosion-inhibitor
Specific gravity at 60 deg. F.	59/52	0·872	0·874	0·896
Open flash point, deg. F.	35/42	445	465	445
Closed flash point, deg. F.	34/47	425	435	420
Pour point, deg. F.	15/42	15	15	—15
Kinematic viscosity at				
100 deg. F. c.s.	71/52	68·4	84·8	84·0
210 deg. F. c.s.	71/52	8·62	9·94	8·31
Redwood 1 viscosity at				
70 deg. F. secs.	70/49	690	900	1,000
140 deg. F. secs.	70/49	111	132	120
200 deg. F. secs.	70/49	52	57	52
Viscosity index	73/47	107	105	67
Neutralization value, mgm. KOH/gm.	1/46(A)	0·05	0·05	0·08
Mineral acidity	1/46(C)	Nil	Nil	Nil
Corrosion (salt water)	135/51	Pass	Pass	Pass
Corrosion (copper strip)	64/52(B)	Pass	Pass	Pass
Ash (sulphated residue), per cent	5/42(B)	Negligible	Negligible	Negligible
Water content	74/45	Nil	Nil	Nil
Demulsibility, secs.	19/51	140	180	240
Foaming: limit ML after 10 mins. of foam collapse	A.S.T.M. D.892–467			
Sequence 1		220	260	100
Sequence 2		Nil	20	Nil
Sequence 3		260	260	120
Oxidation test : neutralization value after 90 hours, mgms. KOH/gm.	114/47P.	0·10	0·10	0·15

*Institute of Petroleum. Standard Methods of Testing Petroleum and its Products.
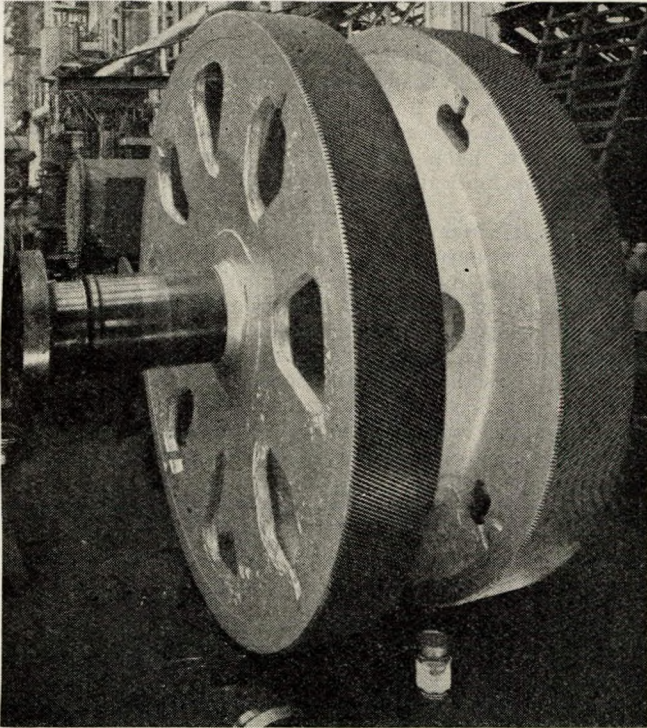
FIG. 14—*Main gearwheel treated with special aluminium paint*

a clean surface but it is estimated that about 90 per cent of the interior surfaces can be adequately protected by this method.

The use of a good oxidation and rust-inhibited turbine oil in an oil system protected by a suitable paint has reduced the risk of corrosion to a point where it can no longer be regarded as a serious hazard.

Table III gives typical inspections of (a) an oil suitable for use in single-reduction geared turbines and preferred by some builders of Bauer-Wach exhaust turbines; (b) an oil suitable for double-reduction geared turbines; and (c) an oil meeting Admiralty O.M.100 Specification.

It will be noted that in addition to the normal physical tests, acidity, corrosion, demulsibility, foaming and oxidation tests are included, all of these being stipulated in the O.M.100 Specification. All the oils quoted are well inside the specification figures.

The branded oils have a better viscosity index, better demulsibility characteristics and better oxidation resistance, whereas the O.M.100 oil has superior anti-foaming properties.

Lubricating Oil Systems

All large marine turbines are fitted with self-contained lubricating oil systems which are of two main types, namely, gravity feed systems and direct pressure feed systems. The former is more popular for merchant ships with the exception of large liners, while the latter is invariably used for warships, where head room is strictly limited.

Gravity Feed Systems

A typical gravity feed system as installed in many merchant vessels is illustrated in Fig. 15. Duplicate rotary motor-driven lubricating oil pumps are installed which take their suction from a drain tank built into the double bottom. The oil passes through suction strainers, one of which is of the magnetic type, and is discharged through duplicate discharge filters to vertical tubular type oil coolers. From the oil coolers the oil passes to a four-way piece, one branch of which leads direct to the turbine bearings and gears, while the other two lead to an overhead gravity tank.

By closing the gravity tank inlet valve, the circuit can be

used as a direct system, in which the pressure at the bearings is controlled by the discharge pressure at the pump. It is common practice to operate on direct supply when first circulating oil through the system, in order to prove all oil lines clear and to ensure that all air pockets are eliminated.

In normal service, however, the gravity tank inlet valve is open and the pressure on the bearings then depends on the static head of the gravity tank. The pump speed is regulated so that there is only a slight flow visible at the overflow sight glass, indicating that the gravity tank is full and the maximum pressure head on the bearings. This is usually in the region of 10-15lb. per sq. in.

From the journal bearings, thrust bearings and gears, the oil returns to the drain tank for recirculation. It will be noted that the drain tank is large enough to hold all the oil in the system plus the oil in the gravity tank and emergency reserve tank, so that if repairs are necessary to any part of the system all the oil can be run down into the drain tank.

As marine turbines are variable speed engines, it is necessary to fit an overspeed device, the most commonly used being the well-known Aspinall overspeed governor in which two spring-loaded valves are located in the end of the turbine spindle. The governor works in conjunction with a steam pilot valve control and an emergency steam stop valve, and it is operated by oil pressure from the main system, as can be seen in the illustration. Up to normal maximum turbine speeds, the springs in the governor keep the valves closed, but when the predetermined overspeed is reached, centrifugal force overcomes the spring pressure and the valves are opened, allowing oil in the circuit to drain back direct to the drain tank. With the oil pressure in the circuit reduced, the needle control valve opens, and steam passes into a cylinder in the emergency valve, exerting pressure on a piston mounted on the valve spindle. As the piston is larger in area than the valve, the latter is rapidly closed. Steam cannot be readmitted to the turbine until the governor is reset by means of a hand lever allowing the valves to close, thus building up pressure again in the oil circuit until it is high enough to shut the needle control valve. This in turn shuts steam off the emergency valve piston and boiler steam pressure again opens the main valve. Any interruption of the normal oil supply also automatically shuts steam off the turbine by means of the oil control system.

It will be seen that the drain tank is built into a cofferdam in the ship's double bottom, so that water or fuel can be carried without coming in contact with the sides or bottom of the tank. It is important that regular soundings should be taken of the cofferdam so that leakage from the double bottom through its sides, or leakage of sea water through the ship's bottom, can be quickly detected. The drain tank has a sloping bottom, so that solids or water pass down into a drain hat from which the water may be pumped by means of a hand pump. The drain tank top is built above the main tank top, a good feature, as it reduces the possibility of bilge water entering through a leak in the top.

In many vessels, reciprocating steam pumps are used in place of the rotary type, but in all systems one pump is capable of supplying all oil requirements, with a second pump held in reserve. The gravity system is more suitable for use with reciprocating pumps, as the pump pulsations are not reflected in the oil supply at the bearings, due to the pressure head in the gravity tank.

Direct Feed System

This is similar to the gravity system described except that the overhead tank is eliminated and oil from the pump passes direct to the turbine bearings and gears. It is usual to fit a spring-loaded discharge valve in the system, adjusted to about 30lb. per sq. in., so that the pressure in the system is kept constant. Emergency governors, controls and emergency stop valves are fitted as in the gravity system.

Bearings

The bearings of both turbines and gears consist of gun-

metal or steel shells lined with high quality, tin base, white-metal. Careful attention is paid to diametral clearance to ensure correct convergence for the establishment of a fluid film, and also to control end leakage. The average clearance is about 2/1,000 inch per inch shaft diameter, as above 3/1,000 inch per inch diameter vibration may occur, probably due to the convergence angle being too great, leading to excessive oil pumping. If, however, the clearance is below 1/1,000 inch per inch diameter, end leakage is too low, resulting in high operating temperatures caused by insufficient oil flow and also to too acute a convergence angle.

During recent years several cases of excessive turbine shaft vibration have occurred, and have been attributed to instability in the oil film. In some instances this has been cured by shortening the length of the bearings, in others by altering the oil viscosity. In some cases it has been attributed to unbalance in the rotor. It is unlikely that the oil itself can cause the vibration, but being elastic in nature, it can react to outside forces. Luckily extreme cases of this type are rare in modern machines.

Turbine bearing loads are low, as, if the rotors are in perfect balance, the only load when running at speed is the weight of the moving parts. Loading on the gear shaft bearings, particularly the pinion bearings, can be considerably higher, but even here it is much lower than in other prime movers, so that fluid film lubrication usually exists.

Thrust Bearings

Thrust bearings of the Michell or Kingsbury type have entirely replaced the older multi-collar thrust, or adjusting block, as a means of locating the turbine rotor relative to the casing, and to take up unbalanced end thrust. Similarly, this type of main thrust block is universally fitted in modern turbine vessels. Even with main shaft speeds as low as 100 r.p.m., fluid film conditions exist. Both turbine and main thrust bearings are supplied with oil from the circulating oil system.
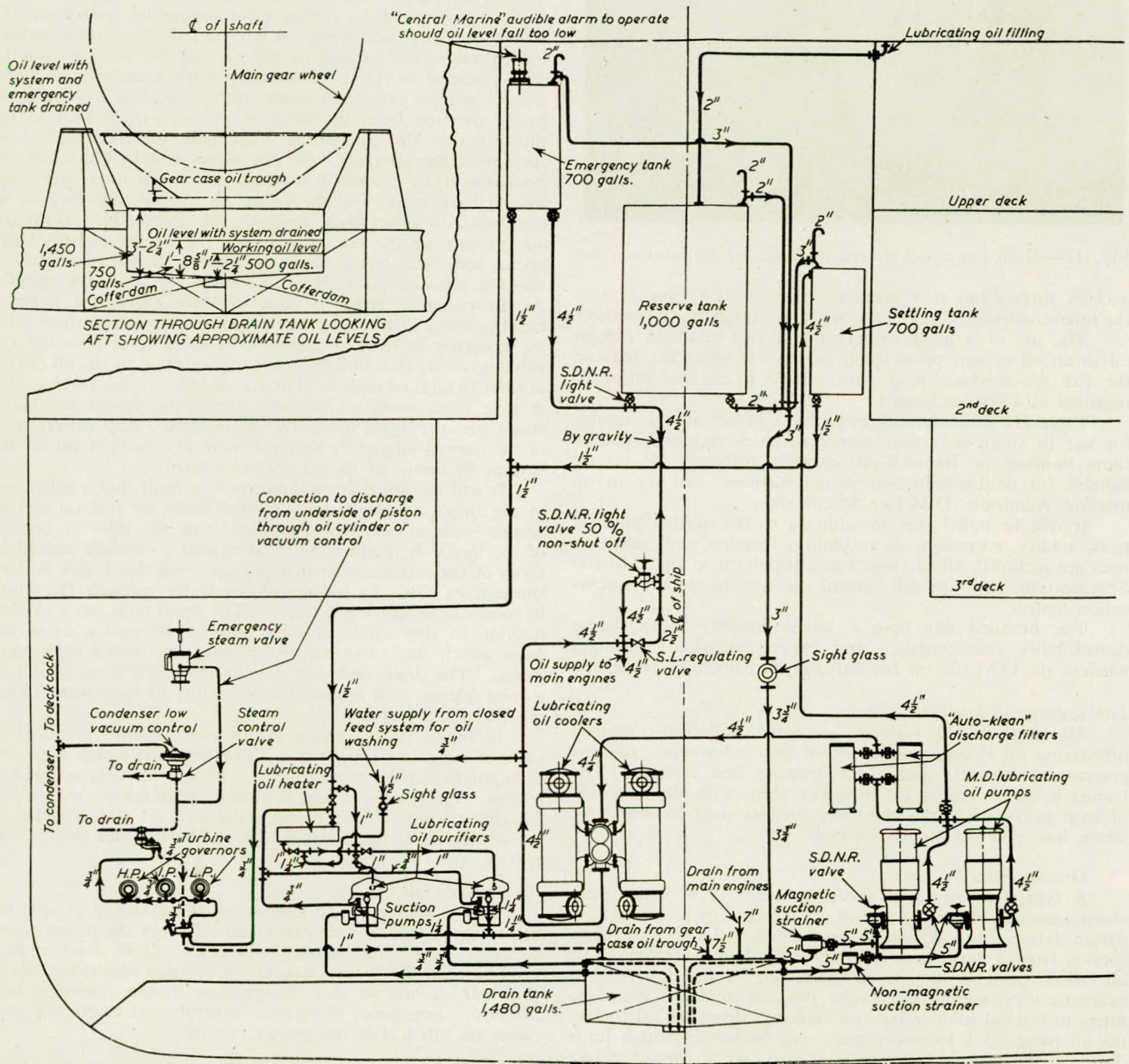

FIG. 15—*Typical arrangement of turbine pressure-gravity forced lubrication system*

It is essential in this type of bearing that the block is entirely oil filled, otherwise cavitation may be set up near the root of the thrust collar due to contrifugal force, thus starving the inner part of the pads of lubricant. It is usual, therefore, to fit a restriction plate in the bearing outlet pipe. It is of particular importance in thrust bearings that the oil used is perfectly clean, as if rust particles or other solids enter with the oil stream they may score the white metal faces of the pads and, in extreme cases, cause breakdown of the oil film.

Gear Lubrication

Both single and double reduction gears are employed for turbine propelled vessels, the former being by far the most widely used. Turbine gears are probably the most efficient and the most carefully cut in industry, but despite the care taken in manufacture, failures still occasionally occur. Some excellent papers on this complex subject have been presented during recent years[10, 11] and the author will only comment briefly on lubrication aspects.

Gear scuffing was particularly troublesome during and immediately after the recent war, mainly in naval vessels, but this now appears to have cleared up and few cases have been reported during the last two years. Scuffing has been attributed to several causes, including badly cut gears, misalignment in gear shafts, unsuitable material and last, but by no means least, unsuitable lubricants.

It is apparent that if a perfect oil film can be maintained between mating gear teeth, metal-to-metal contact and subsequent local surface welding cannot occur. The lubrication of gears is a particularly difficult problem under the best of conditions, as theoretically, only line contact occurs between mating teeth and pressures are, therefore, exceptionally high. In practice, of course, due to elastic deformation of the metal, area contact occurs, although the width of the contact area is very small. Even with the most efficiently cut gears of the helical type, a certain amount of sliding motion takes place and this, allied to the high loading, places severe demands upon the lubricant. Undulations on the gear faces accentuate the problem as the load is necessarily carried on the high spots.

Under such conditions extreme-pressure lubricants could be expected to greatly improve lubrication, but as previously mentioned, other considerations have prevailed and a straight heavy turbine oil is still universally used. It might well be that with improved methods of gear finishing, such as shaving or lapping, although permitting higher designed loads to be carried, local intensity of loading will be reduced as undulations will be virtually eliminated and demands on the lubricant will be correspondingly less severe.

A typical modern marine turbine unit is shown in Fig. 16 with the turbine casing and upper half gear covers removed. It will be noted that double reduction gears of the double helical type, with quill drive for the second reduction pinions, are fitted. The turbine Michell thrust blocks and the claw type flexible couplings can also be seen. The gear oil spray connexions are shown at the upper left of the illustration.
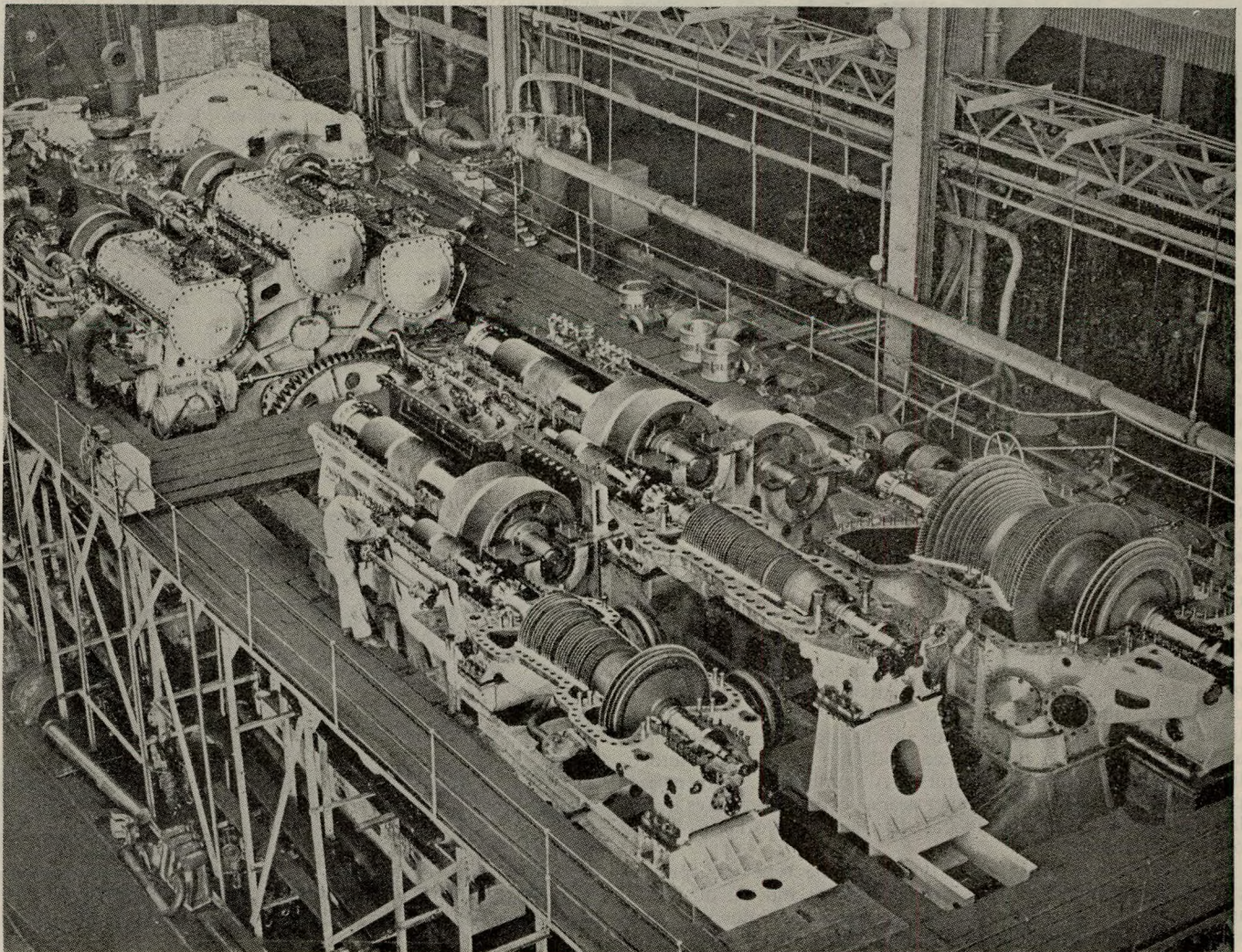
In order to obtain the maximum efficiencies for both

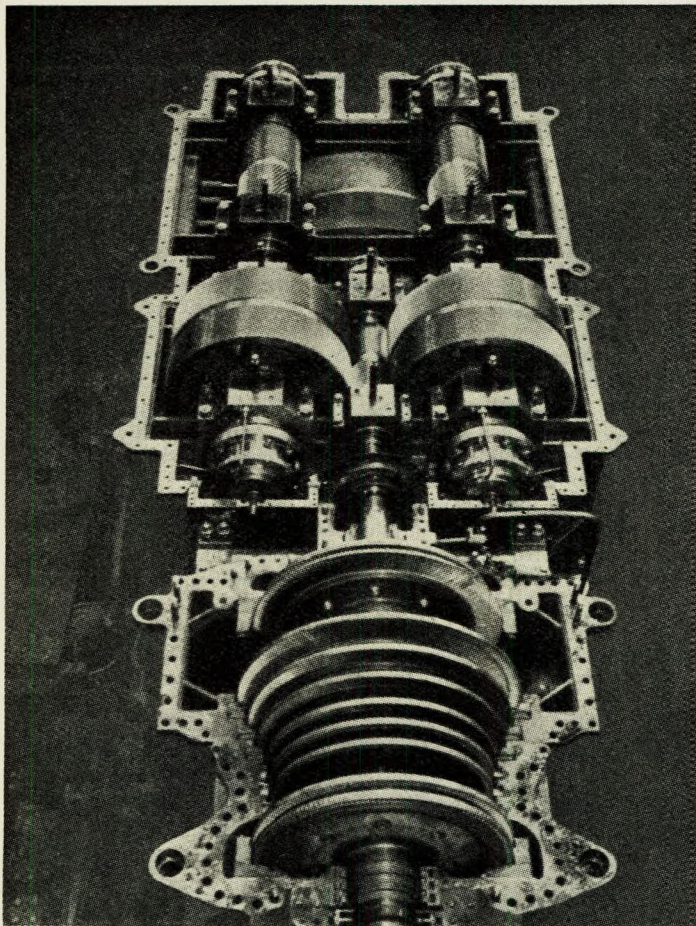Fig. 16—*Modern double-reduction geared turbine set*

with oil and there is little risk of aeration which could cause local failure of the oil film.

It is essential that the oil used for gear lubrication be completely free of any solids, as these would be carried into the meshing gears and serious damage could result. Similarly, moisture in the oil could, under the extreme loads and local temperatures encountered, flash into steam, and this has been put forward as a possible cause of gear pitting. In any event water would break down the oil film locally, resulting in metal-to-metal contact.

Oil Coolers

These are essential parts of the system and either of the duplicate coolers is capable of handling all the cooling requirements, even in tropical climates. This enables a clean cooler to be kept in reserve, and in emergency they can be changed over without shutting down the oil system.

Careful attention is now paid to both the design and location of oil coolers. In older machines the coolers were usually horizontal, and of small capacity, frequently located in inaccessible positions, but in modern installations vertical coolers are fitted, usually on the bottom platform.

Sudden chilling of the oil is avoided by introducing the oil into the cooler at the opposite end to the cooling water inlet, which greatly reduces the possibility of sludge deposition on the tubes, with eventual choking up of the cooler. Care, however, must be taken not to maintain too low a cooler oil outlet temperature, as this causes unnecessarily high fluid friction losses in the bearings. In far too many cases the oil is kept too cold, and it is quite a common practice to fully open up the cooling water valve as soon as the turbines are warmed through. This is particularly true in cold weather. Overcooling is one of the main causes of choked oil coolers, as sludge will always settle out in the coldest part of the system, and in this case it is difficult to remove. Providing that bearing temperatures are not too excessive, the cooler oil outlet should be maintained around 100-110 deg. F. Bearing outlet temperatures are kept much too low in many cases, and considerable additional power can be obtained by careful control of bearing inlet and outlet temperatures. Observed outlet temperatures should be about 135-145 deg. F., although, of course, provided that there is an adequate oil supply, individual bearing temperatures may be considerably higher without any harmful effect to the oil.

Cleaning of Turbine Oil Systems

The cleaning of steam turbine lubricating oil systems falls into two categories:—

(1) Cleaning of systems in a new engine preparatory to filling with final oil charge.
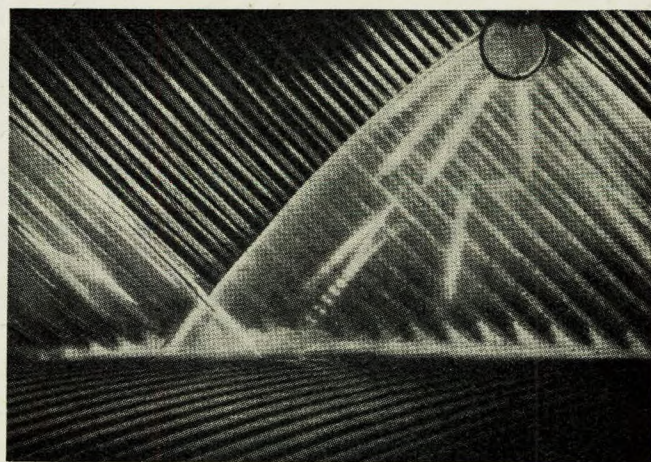(2) Cleaning of systems in old turbines when the oil has

Fig. 17—*Locked train double reduction gears for small warship*

turbines and propeller, particularly in modern installations employing high-pressure steam, double reduction gears are again coming into favour. Locked train double reduction gears have been used, particularly in large American warships, with very satisfactory results. A typical set of double reduction locked train gears as fitted in a small British warship is illustrated in Fig. 17. It will be seen that the power is transmitted to the main gearwheel through four pinions instead of two, so that tooth pressures being lower, a shorter and more compact unit is possible. A further advantage is that the tangential thrust of the first reduction pinion is neutralized, so that the pinion-bearing loads are greatly reduced. To assist in equalizing the loads on the gear teeth, a quill shaft with flexible couplings is used to connect the first reduction gear wheel to the second reduction pinion.

Gear Sprayers

Sprayers are universally employed to supply oil to the gears. A copious oil supply is essential, as the oil must cool as well as lubricate the gears over their entire length. It is usual to apply the oil to the gears on the ingoing side, although application of oil on the outgoing side has proved satisfactory in small, geared turbo-generators. A number of sprayers are provided for each set of meshing gears, so that the sprays overlap, giving an added margin of safety against blockage of any of the oil outlets.

The sprayers are of two types, in one type the sprayer ends have a number of holes drilled in them and the oil issues in a series of fine streams, while in the other, known as the spoon-type sprayer, a flat fan-shaped spray of oil is directed on to the gears. Fig. 18 shows the latter type of sprayer in use, and it will be seen that the gears are completely flooded

FIG. 18—*Spoon-type nozzle spraying oil into meshing gear teeth*

become unfit for further service, preparatory to filling with new oil charge.

Much valuable information concerning the cleaning of both new and old oil systems is contained in a paper published by the American Society for Testing Materials[12], copies of which are available to engine builders and superintendent engineers.

Cleaning of Oil Systems of New Turbines

The condition of the oil system in new turbines is frequently far from satisfactory when fitting out is completed. Oil pipes, bearing housings, gear cases and drain tanks often contain rust, welding scale, lint, dust, dirt and other foreign matter. Even where all parts leave the shops clean, they remain open for long periods during erection, and inevitably contaminants find their way into the system. Particularly during winter months, rusting of ferrous parts of the system can proceed to a surprising degree, and once started, it is difficult to stop, even when a corrosion-inhibited oil is finally used.

To prevent this initial corrosion, all machined parts should be protected, preferably with a petroleum-base rust preventive of which many effective types are now available. Rusting of non-working parts can be effectively taken care of by the application of a suitable hard drying protective paint as previously described.

It is essential, however, that all temporary rust preventives should be removed before the final oil charge is put in. Wherever possible they should be removed by kerosine or petroleum solvents immediately prior to filling the system with a flushing oil. It is advisable to coat the exposed working parts with a corrosion-inhibited turbine oil as soon as the rust preventives are removed.

The question of whether or not a flushing oil should be used, and which is the most suitable type, is frequently discussed. In the author's opinion, a separate flushing oil charge is highly desirable, and the most suitable oil is the same as the final charge. No matter how carefully temporary rust preventives are removed by hand, it is practically impossible to remove them entirely, and even a small amount of such materials in the final oil charge can seriously affect its demulsibility properties. Other soluble contaminants can also adversely affect new turbine oil.

A poor quality flushing oil cannot be tolerated as it is practically impossible to remove it all after flushing is completed. It is possible to "lose" several barrels of oil in a system after draining is completed, and this, too, could adversely affect the performance of the final oil charge.

Different builders have their own methods of flushing the oil system but, briefly, the oil should be circulated for at least forty-eight hours with an efficient centrifuge in continuous use. Without a suitable filter, foreign matter is merely moved from one part of the system to another. It is preferable to heat the oil during flushing to a temperature higher than that normally experienced in the circuit. Besides being a better solvent when hot, the oil will cause the metal surfaces to expand to a greater degree than during normal service, and this liberates both soluble and insoluble contaminants. The oil can be heated by passing hot water or low pressure steam through the water side of the oil cooler, or by temporarily installing a steam heating coil or electric immersion heater in the drain tank.

After initial flushing, temporary wire gauze strainers placed between the oil pipe flanges in suitable positions should be examined, and if it appears that the system is fairly clean, if possible the engine should be turned over in turning gear with the oil circulating. This should remove any remaining contaminants on the working parts.

In land practice, it is quite common for the turbine to be put on load when it is decided that the system is fairly clean, keeping the flushing oil in use as the lubricant for several months before draining and refilling with the final charge. This is not, of course, so easy with marine installations, but it might be worth considering in certain cases.

A laboratory test of the flushing oil often provides useful information regarding the contaminants removed, and examina-

tion of the centrifuge bowl will indicate when no further foreign matter is being extracted.

When flushing has been completed it is usual to open up bearings, gears and drain tank for examination. Any solids flushed through to the drain tank should be removed manually. The final oil charge should be put into the system as soon as possible after draining off the flushing oil.

Cleaning of Systems in Old Machines

The cleaning of the oil system in an old machine where the oil has been in service for many years and has reached the end of its useful life, is usually a more difficult problem than cleaning the system in a new machine. Obviously, the amount of cleaning required depends on the condition of the oil system. Ideally, the oil should be taken out of service *before* sludge and other solids are deposited but, unfortunately, this is rarely the case. Apart from oil degradation products, the system will probably contain water, metallic oxides, lint, ash and dust, and these should all be removed before the fresh oil charge is put into service.

CLEANING WITH OLD OIL

When the system is known to be reasonably clean the old oil can be used to clean up the mess it has helped to make! The best way of doing this is to employ an efficient centrifuge and a hot water wash. Most of the oil oxidation products are water soluble, and they act as binders for finely divided solids of all types. By using hot water to wash the oil, the water soluble materials are gradually dissolved and pass out with the centrifuge water discharge. With the removal of the binders the solids are deposited in the centrifuge bowl, from which they can be removed periodically. Hot water has the additional advantage of coagulating finely divided dust, ash and metallic oxides, forming larger particles which are more readily removed in the centrifuge.

The higher the temperatures of both oil and water, the more efficient is the cleaning process, and it is suggested that, if possible, the oil temperature should be maintained at 150-160 deg. F. and the water temperature 160-180 deg. F., with the centrifuge throughput reduced to about half its rated capacity to ensure maximum efficiency. This method of cleaning, provided it is carefully carried out, may well be done with the vessel in service, over a period of several weeks, until it is considered that no more can be done. By this time the centrifuge water discharge should be quite clear, and the bowl should be practically free of deposits.

The oil should then be drained out of the system while still warm and the system opened up for examination. As far as possible any remaining deposits should be removed manually.

When cleaning is completed, a new charge of double-inhibited turbine oil should be put into the system and preliminary flushing carried out as outlined for a new machine. The centrifuge should be kept in continuous use during the flushing period but water washing should be discontinued. Modern turbine oils have high solvency properties for any remaining oil soluble, water insoluble, oxidation products, and after a week or so in service the turbine should be quite clean. If, after laboratory tests, the oil is shown to be in good condition it can remain in service, but if contaminated it will have to be removed and a fresh charge put in.

If, on examination, after hot water washing with the old oil the system is found to be still dirty, it will probably be necessary to resort to more drastic methods of cleaning. These can be divided into three main groups:—

    (a) The use of low-pressure steam or hot water.
    (b) Aqueous alkaline degreasing solutions.
    (c) Non-petroleum aromatic solvents.

Steam or Hot Water. Owing to the complexity of marine turbine oil systems, hot water or steam washing is usually restricted to coolers, tanks, bearing housing and gear cases. Hot distilled water should be used at a temperature of about 180 deg. F., and with careful application by hose most of the

remaining solids in the main parts of the unit can be washed out.

Steam has been used successfully for the cleaning of oil tanks and oil coolers. Neither method has proved very popular aboard ship, mainly because it is difficult to avoid washing dislodged solids into inaccessible parts of the system from whence they will later enter the oil stream.

Alkali Solutions. Several proprietary alkali degreasing compounds are available and some have proved most effective for cleaning very dirty oil systems. Great care, however, must be taken to remove all traces of alkali when cleaning is completed, otherwise troublesome emulsions might occur when the engine is put back into service.

A steam coil should be temporarily fitted into the drain tank so that the solution can be heated to about 180 deg. F., and a temporary circulating pump connected up to the current. The system should be filled with distilled water, the temperature raised to about 160 deg. F. and the alkali slowly added until the desired concentration is reached. The cleaning solution is circulated for 6-12 hours depending on the state of the system, and this is usually sufficient to completely clean it. The system is then emptied and immediately refilled with distilled water, which should be circulated at approximately 180 deg. F. The increased circulating temperature ensures that any remaining alkali is liberated from the pores of the metal. At least two clean water washes are required, and they should be continued until the water is completely neutral as indicated by phenolphthalein.

The metal surfaces will now be chemically clean and they should be immediately wiped down with a rust inhibited turbine oil, the set reassembled, and the final oil charge put in without delay. Where oil coolers are very dirty it may be necessary to clean them separately with a more concentrated alkali solution.

Solvent Degreasers. Non-petroleum solvents have successfully cleaned some very dirty oil systems. The three solvents most widely used are trichlorethylene, carbontetrachloride and tetrahydronaphthalene (trade name "Tetralin"). The two former solvents are toxic, and care must be taken when they are used to avoid personnel inhaling fumes. "Tetralin" does not present a toxic hazard, and for this reason is safer to use unless the cleaning is carried out by specialists. A further disadvantage of chlorinated solvents is that if pockets are left behind in the system, they could ultimately combine with water to form corrosive hydrochloric acid. All three solvents are non-inflammable in the liquid phase.

Several firms specialize in the cleaning of oil systems with solvents, which can be used either in the liquid or the gaseous phase. It is usual to blank off bearings and gear sprayer pipes to avoid dislodged solids being trapped where they could later cause trouble. After degreasing is completed the system should be thoroughly blown through with dry air, preferably warm, to remove all traces of liquid or gaseous solvent. As the exposed surfaces are now very susceptible to corrosion, they should be coated with turbine oil and the system filled with the new oil as soon as possible.

Turbine Oil Maintenance

The majority of modern turbine vessels are fitted with efficient oil-clearing equipment as, unless it is maintained in good condition, the useful life of the oil can be considerably shortened. As has already been mentioned, the oil is constantly exposed to contaminating influences and these should be removed as soon as they enter the system. Coarse suction and discharge strainers are fitted as standard, which will remove the larger solid contaminants down to about 3/1,000 inch in size. Magnetic suction strainers are useful as they remove all magnetic ferrous oxides, or fine particles of steel from the gears which could cause damage to working parts.

Centrifuges

Centrifuges are by far the most widely used method of cleaning turbine oils aboard ship and, provided that efficient machines are installed and operated correctly, they play an important part in keeping the oil in good condition.

Centrifuges can be operated either as "clarifiers" or as "purifiers" (sometimes termed "separators"). In the former case there must be no water present as all matter removed from the oil remains behind in the bowl. When used as purifiers, however, independent water and oil outlets are provided. As it is impossible to exclude water entirely from turbine oil systems, the centrifuges are invariably used as purifiers.

A modern centrifuge is a most efficient machine and can remove solids down to as fine as 3 microns. It is usual to fit a machine capable of cleaning about 10 per cent of the total volume of oil in the system per hour, that is, a turbine with 2,000 gallons of oil in circulation will require a centrifuge capable of handling 200 gallons per hour. The term centrifuge "capacity" is frequently misused; in this connexion it does not refer to the maximum amount of oil that can be passed through the machine in a given time, but to the volume it can centrifuge at maximum efficiency. In many cases the maximum efficiency is obtained when the machine is handling about 50 per cent of its maximum rated throughput.

Differences of opinion exist as to the most suitable oil temperature at which centrifuging should be carried out. The higher the temperature of an oil the greater is its capacity for dissolving sludge, and from this point of view centrifuging at low temperatures would give the greatest sludge removal. On the other hand, the higher the temperature the more readily will solids and water separate out. If the oil is known to be fairly heavily oxidized, on balance it would be preferable to centrifuge without heating. With a clean system and the oil in good condition, however, it is recommended that centrifuging should be carried out at a temperature of 150-160 deg. F., as this will not harm modern inhibited oils but it will give maximum cleaning efficiency. With non-inhibited oils it might be advisable to lower the temperature to 140 deg. F., as aeration in the centrifuge bowl at high temperatures could hasten oxidation.

Water washing, which may be used with advantage for cleaning the older type non-inhibited oils, particularly when oxidized, is not necessary or desirable with inhibited turbine oils. Certain additives are water soluble and the use of hot water in the centrifuge would gradually remove them.

Referring back to Fig. 15, the oil cleaning system can be seen, consisting of two duplicate centrifuges, a small oil heater and the necessary pipe connexions. For normal routine operations, oil is drawn from the sump in the base of the drain tank by the small centrifuge pump and discharged through the heater to the bowl of the machine. After cleaning, the oil is discharged back to the drain tank.

If the oil is known to be very wet, batch filtration can be carried out, preferably in port. Oil is pumped to the settling tank which is usually fitted with a steam heating coil. After allowing the oil to settle for about twenty-four hours at a temperature of about 120 deg. F., most of the free water can be run off to bilge. The connexions to the purifier are then opened and the oil cleaned before returning it to the drain tank.

The amount the centrifuge needs to be used depends on several factors. If the set is a "wet" one, due to leaking steam glands or excessive condensation, it may be necessary to run it daily for two or three hours. If, however, the set is in good mechanical condition and the oil fairly clean, it may only be necessary to run the centrifuge for about four hours per week. If there is the slightest salt water leak into the oil system the centrifuge should be operated continuously until the leak can be stopped.

Laboratory Evaluation of Used Turbine Oils

The major tests carried out on lubricating oils have already been briefly described. The evaluation of used turbine oil is of great importance to the operator, as laboratory tests will indicate whether or not the oil is fit for further service. The

keeping of a careful record of the condition of oil in a turbine is important, as a single test could well give a false picture. Samples should be drawn at regular intervals, and it is suggested that 2,000 hour intervals are sufficient once the oil has settled down in a new engine.

No doubt there will always be differences of opinion among chemists and turbine operators as to the value of different laboratory tests on used oils, but the main purpose of such tests should be borne in mind, which is to follow closely the performance of the oil in service, so that if necessary, it may be removed *before* sludge, varnish, or solids have formed in the oil system. In too many cases the oil is changed only *after* the system has become badly fouled.

There is little point in carrying out any but the minimum number of tests which will indicate the condition of the oil. If preliminary tests indicate that the condition is unsatisfactory, then an additional test may be called for. The routine tests usually carried out are:—

> Neutralization value (acidity).
> Viscosity Redwood I at 140 deg. F. or kinematic
> viscosity at 100 deg. F.
> Water content—amount and nature.
> Sediment content.

Another test, the interfacial tension test, has been found valuable in assessing the condition of an oil, but it is essential that the composition of the oil is known, as certain corrosion inhibitors can greatly affect the readings.

A 16-oz. sample is usually sufficient for routine testing and special sample bottles should always be used. Probably the ideal container is a dark coloured glass bottle fitted with a ground glass stopper. Care must be taken to ensure that the bottles are clean and dry before drawing the sample. Coloured bottles are preferred to clear white bottles, as it has been found that daylight, particularly strong sunlight, can affect the test results. Natural rubber stoppers are unsuitable as they are attacked by oil, and corks, although usually satisfactory, can crumble and small pieces may get into the sample. Corks covered with tinfoil have been found very satisfactory. Tins are not really suitable as it is difficult to see when they are clean and with soldered tins it is possible for flux to contaminate the oil.

Dry lintless cloths should be used for cleaning out sample bottles as cotton or wool waste invariably leaves fibrous material behind. As a precaution, if the bottle has been previously used for samples, it is advisable to use a little benzene or petroleum solvent for cleaning.

The method and place of drawing off samples from the oil system is important. For example, if a sample were taken from the bottom of the drain tank it would possibly consist mainly of water and sludge and if this were taken as a representative sample, the laboratory results would indicate that the oil was unfit for service. Samples should be drawn from the system with the turbine running and the oil circulating, a suitable place being from the oil pump discharge. If not already fitted, a small draw-off cock could easily be fitted in the pump discharge pipe. It is advisable to take two samples, so that if one is suspect the other may be used as a check.

MARINE DIESEL ENGINE LUBRICATION

The satisfactory lubrication of Diesel engines presents some of the most difficult problems encountered by oil technologists. This is especially true of large marine engines, where, due to low speeds and high loads, it is difficult to establish fluid film lubrication. Cylinder lubrication is particularly difficult due to the extremely high temperatures encountered. In addition, the contaminating influences of partially burnt products of combustion on the crankcase oil have to be considered. The ever-present risk of water leakage into the crankcase oil either from condensation or from leakage of the cooling system, influences and often restricts the use of otherwise suitable additives.

Cylinder Lubrication

At first glance it would appear that no mineral oil could withstand the extremely high temperatures encountered in a Diesel engine cylinder. Gas temperatures may exceed 3,000 deg. F. at the top of the stroke, whereas the fire point of a typical, good quality, lubricating oil used as a cylinder lubricant is in the region of 460 deg. F. and thermal cracking occurs about 750 deg. F. It must be borne in mind, however, that these are typical figures at atmospheric pressure and they rise with increase of pressure, although unfortunately no reliable figures appear to be available of the fire point and cracking temperatures at pressure similar to those normal in Diesel cylinders. A further factor which must be considered is that the lubricant is present as a very thin film on a cooled surface. Reliable figures for liner skin temperatures are most difficult to obtain but it would appear that the maximum skin temperature encountered at the beginning of the working stroke is in the order of 450-500 deg. F., while at the lower end of the stroke it may be as low as 250 deg. F. The oil film is subjected to the maximum gas temperature for only a fraction of a second, however, so that it has hardly time to be completely burnt before it is renewed.

The essential properties required in an oil used for cylinder lubrication are:—

(1) It must reduce sliding friction and wear to a minimum, it must possess adequate viscosity at the cylinder working temperature, it must spread rapidly over a hot surface and still form a tough adsorbed oil film.

(2) It must form an effective piston ring seal, preventing "blow-by" and loss of compression.

(3) It must burn cleanly, leaving as little and as soft a deposit as possible.

(4) If a detergent type oil, it must effectively disperse "carbon" from the ring zone and prevent build-up of deposits in the ports of two-cycle engines.

With mean piston speeds seldom exceeding 1,500ft. per min. and stoppage and reversal of direction at top and bottom of the stroke, allied with high temperatures, it is impossible for fluid film lubrication to exist except perhaps about mid-stroke where speeds are highest. Cylinder lubrication must, therefore, be considered as a boundary lubrication problem.

Viscosity Requirements of Cylinder Oils

We have seen that viscous steam cylinder oils of the paraffinic type, possessing long chain molecules, are efficient boundary lubricants, and from this aspect would seem to be suitable for use as Diesel cylinder lubricants. Unfortunately, however, such viscous lubricants tend to form excessive carbonaceous deposits and, furthermore, the question of application must be considered.

In all large Diesel engines the cylinder lubricant is supplied by mechanical lubricator to two or more points on the cylinder walls. It is essential that the oil spreads rapidly around the cylinder to completely cover the walls, otherwise dry spots will occur with subsequent disastrous results. The greater the number of oil feeds, the better will be the distribution, but practical difficulties usually restrict the number to a maximum of six and it is considered that four should be the minimum permissible. Shallow circumferential grooves assist in spreading the oil but are often objected to on the grounds that they can cause ring breakage and also weaken the liner.

The vertical positioning of the oil feeds and the timing of lubricant injection are important features and, as is usual in such matters, there are differences of opinion in the matter. Perhaps the most suitable position is between the second and third compression rings with the piston at top dead centre and injection carefully timed so that it commences as the top ring covers the feed point on the upward stroke and is cut off as the top ring passes the hole on the downward stroke.

Low viscosity oils spread more rapidly than viscous oils and, therefore, they more easily form a complete oil film on the cylinder walls. An additional argument in favour of light oils is that they form the minimum of deposits. The dis-

advantages of light oils are, of course, that they are poor boundary lubricants and also that they are more volatile than heavy oils. They tend, therefore, to boil off the cylinder walls more readily. It will thus be seen that we already have conflicting requirements for an ideal cylinder lubricant and in most cases a compromise in viscosity is arrived at.

Allied with viscosity, the viscosity index of the lubricant must be carefully considered. A high viscosity index oil, showing the minimum change of viscosity with rise in temperature, has much to commend it. In addition, such oils have the greatest natural stability and are less volatile than low viscosity index oils of the naphthenic and aromatic types. From the viscosity point of view then, a high viscosity index oil having the highest viscosity commensurate with satisfactory distribution and the greatest resistance to volatility is the obvious choice.

Considering cylinder lubrication as a boundary lubrication problem under high temperature conditions, lessons learnt in other fields of lubrication can well be taken into account. We have seen that selected oiliness agents greatly improve the lubricating properties of mineral oils, and that under very severe conditions extreme-pressure agents are most valuable. The question is then, can such compounding agents or additives increase the lubricating properties of mineral oils used as Diesel cylinder lubricants. Unfortunately, many fatty oils possessing desirable polar properties would decompose at cylinder temperatures and form objectionable deposits. In addition, it is likely that in some cases their effectiveness would be lessened by the high temperatures encountered. Extreme-pressure agents might well improve lubrication but could form abrasive ash deposits under such conditions. It will thus be seen that the field to select from is considerably narrowed.

Work carried out using esters of fatty acids in mineral oils has, however, proved extremely promising. No increase in cylinder deposits has been observed and in many cases liner and piston ring wear has been halved. These oils are most stable under normal temperature conditions, and in the cylinder, the polar molecules form a tenacious adsorbed film on the walls. It is likely that the chemical action between these molecules and the hot metal is accelerated, resulting in an oil film with increased load-carrying properties, thus reducing metal-to-metal contact and wear.

The addition of bright stock to a medium viscosity base oil is common practice, the amount of bright stock used depending on the final viscosity required in the blend, but 10-30 per cent is usual. If, however, suitable oiliness additives can be developed which give superior lubricating properties to a distillate base oil, bright stock can be eliminated with a corresponding reduction in carbonaceous deposits.

Cylinder Wear

The causes of cylinder and piston ring wear are probably the most controversial subject connected with Diesel engines. Cylinder wear is becoming increasingly important as higher output engines are developed. The use of residual fuels, with higher sulphur and ash contents, has further aggravated the problem. That there are several factors contributing to cylinder wear is obvious and the effect of the different factors varies with the design and operating conditions of the individual engine.

The main causes of cylinder wear can be classified as follows:—

(1) Abrasive wear caused by foreign matter, mainly introduced with the induction air, and also by hard particles of carbon, asphalt, or ash, from incomplete combustion of fuel or lubricating oil.

(2) Corrosive wear caused by products of combustion, particularly when starting up or when running under light load.

(3) Wear caused by metal-to-metal contact due to breakdown of the oil film, as a result of an unsuitable oil being used or an inadequate oil feed.

All three causes may be interrelated and can all occur at the same time in an engine cylinder. As, however, these can

be encountered in all Diesel engines they can be described as normal wear hazards.

Other factors influencing wears are:—

(1) Unsuitable material used for liners, pistons and rings.

(2) Poor workmanship in machining cylinder bores, piston rings and pistons.

(3) Distortion of pistons or liners due to bad design or poor local cooling.

(4) Dirty fuel injectors causing uneven spray pattern increasing risk of distortion.

(5) Poor design, insufficient, or too great, piston ring vertical clearances.

(6) Poorly designed or leaking fuel injectors causing excessive dilution of lubricating oil and subsequent breakdown of oil film.

(7) Inadequate piston cooling causing excessive ring zone temperatures.

Corrosive Wear

Many authorities consider that corrosion is the main cause of cylinder wear. This can be attributed to the presence of moisture and acidic materials formed during combustion. For every pound of fuel burned, roughly a pound of H_2O in vapour form is produced within the cylinder. Liquid fuels, of course, consist mainly of hydrogen and carbon, but sulphur contents as high as four per cent are common in heavy residual fuels. In addition, magnesium and sodium chlorides may be present in small quantities due to fuel contamination by salt water. During combustion, large volumes of carbon dioxide are formed, with lesser amounts of sulphur oxides, and traces of nitrogen oxides and chlorine compounds. If these combustion products combine with H_2O, carbonic acid, sulphuric and sulphurous acids, nitrous and nitric acids, and hydrochloric acids can be formed.

Effect of Sulphur Compounds

Of the several corrosive compounds likely to be present it is recognized that sulphur is the most important. Although it is common to talk of sulphur content of fuels, it must be remembered that this does not refer to pure sulphur but to complex sulphur-organic compounds, which in many cases form part of the hydrocarbon molecules. If the sulphur were present in free form, it would be relatively easy to remove it during refining and it would be a valuable raw material for other industries. Sulphur compounds can be removed from distillate fuels of the gas-oil type and modern refineries install special plant to remove them by a liquid extraction process. Unfortunately, this is an expensive process, and if applied to marine Diesel fuels of the heavy distillate type would considerably increase their price. It is extremely difficult to remove sulphur compounds from residual fuels, and although it can be done, the increased cost of the fuels would rule it out.

Much of the crude oil available from sterling, or semi-sterling areas, comes from oil fields in the Middle East and these crudes have high natural sulphur contents. Marine Diesel fuels refined from such crudes have sulphur contents as high as 1·5 per cent, although 1·0-1·2 per cent is more common, while residual fuels from the same source can have sulphur contents as high as 4·5 per cent.

Marine Diesel fuels come under Class B of the British Standard Specification No. 209:1947. Class B fuels may be wholly heavy distillates or they may be a blend of distillate and straight run (not cracked) residuum, provided that the blend meets the specification requirements. As would be expected, wholly distillate fuel is usually lower in sulphur content than a blended fuel, but this also depends on the source of the crudes used. The maximum sulphur content permissible for Class B fuels is 2·0 per cent, so that all commercially available marine Diesel fuels can be expected to be well inside the limit.

Sulphur contents of many of the marine Diesel fuels currently available at bunkering ports are considerably higher than in pre-war days and this is even more marked with heavy residual fuels. If, therefore, sulphur is largely responsible for

cylinder wear, it is obvious that increased wear must now be accepted. This is borne out by figures available and it would also appear that where ships are using residual fuels, cylinder wear is approximately 60-100 per cent greater than when using marine Diesel fuel. This is confirmed by Humphreys,[13] Arnold,[14] Kilchenmann,[15] and Jackson,[16] but Lamb,[17] found little difference in wear between the two fuels.

During combustion, sulphur can burn to form gaseous sulphur dioxide (SO_2), or sulphur trioxide (SO_3). It is extremely difficult to determine the relative amount of each gas present in the cylinder, but recent experimental work[18] has indicated that of the total sulphur oxides present in the exhaust gases, 90-95 per cent are sulphur dioxide and 10-5 per cent sulphur trioxide. This is fortunate, as the former is relatively harmless from the corrosion point of view. Certain authorities contend that sulphur trioxide is harmless to metals in the gaseous state. If, however, condensation takes place within the cylinder, the sulphur trioxide combines with water to form highly corrosive sulphuric acid.

At first glance it would appear that condensation could not take place within the cylinder but on closer investigation this appears to be quite feasible. It must be borne in mind that the liner skin temperature rarely exceeds 500 deg. F., that sulphur trioxide can raise the dew point[19] (or condensation temperature) considerably, and that the dew point is considerably raised at high pressures. Unfortunately, no reliable figures are available giving dew points of sulphur compounds at pressures encountered in Diesel cylinders. It is interesting, however, to compare the effect of pressure on the boiling point of water. At a pressure of 500lb. per sq. in. (equivalent to the compression pressure in many Diesel engines) the boiling point of water is 473 deg. F. Assuming that the boiling point at this pressure corresponds to the dew point, it could be possible for some of the water vapour present in a Diesel cylinder to condense on the walls, if the surface temperature falls below this figure. If the dew point of sulphur trioxide is similar to that of steam, then conditions appear to exist where sulphuric acid would form on the cylinder walls.

As sulphuric acid is used in refining lubricating oils, it is possible, particularly if a poorly refined cylinder oil is used, for the acid to attack the oil, removing the colour-forming constituents, thus forming acid sludge in addition to damaging the oil film. It is not considered, however, that this materially contributes to corrosive wear.

If the hypothesis regarding the formation of sulphuric acid on the cylinder walls is correct, engines running under light load and/or low jacket temperatures would be expected to show the greatest cylinder and ring wear, and this is borne out by experience, particularly in small engines where the walls are thin and consequently skin temperatures lower.

It is generally accepted that engines which are being constantly started and stopped show more rapid wear than engines which run for long periods. Typical examples of the former case are Diesel-engined ferries, tugs and other river craft. Such engines never really get warmed through and consequently cylinder wall temperatures are low, so that surface condensation is much more likely.

A disproportionate amount of wear takes place in starting an engine from cold. Some of the fuel injected into the cylinder strikes the cold walls and condenses. As in most cases the walls are devoid of lubricant when starting up, conditions are then ideal for attack by corrosive by-products of combustion.

Sulphur has another adverse effect in Diesel engine cylinders. It has been shown[20] that sulphur oxides can convert loosely packed fluffy soot particles left by incomplete combustion into hard brittle deposits. These are particularly noticeable on the top land of the piston. Lubricating oil is frequently blamed for these deposits but even with a good detergent oil, such deposits persist. As in most cases the top land is given a considerable taper to allow for expansion when hot, it is clear of the liner, particularly in crosshead type engines. Lubricating oil, under such conditions, cannot contact the top land and any deposits found in this zone must come almost entirely

from the fuel. The danger is that if such deposits are excessive, they will eventually build up on the top land until the clearance space is taken up. As the deposits are hard and brittle they then break up and the particles cause abrasion to the liner and piston. They also work their way past the rings and can materially affect ring sticking. Top land scuffing, as it is termed, is quite a serious problem in small, high-speed engines but is of less importance in large engines where clearances are greater.

As already mentioned, sulphuric acid and also sulphur dioxide will react with lubricating oils to form acid sludge. This sludge will bake hard under high temperatures to form varnish and carbonaceous deposits in the ring zone and on the piston skirt. A slight degree of varnishing can be tolerated and may even help to prevent metal-to-metal contact, but fouling of the rings with eventual ring sticking is highly objectionable.

Methods of Combating Effects of Sulphur

As high sulphur fuels have to be accepted, particularly when burning residual fuels, all practical steps must be taken to minimize its effects. Liner material necessarily plays an important part and although a considerable amount of research has gone into improving liner metals, the ideal material has not yet been found.

Chromium plating has greatly reduced wear in small and medium-sized engines but unfortunately the cost of chromium plating large liners has greatly restricted its use. Porous chrome appears to give better results than solid chromium plating, probably because lubricating oil is absorbed into the surface thus assisting in the retention of the oil film. The main reason why chromium plating gives reduced wear is probably because it is highly resistant to acidic attack and sulphur oxides or sulphuric acid have little effect on it.

Chromium plated top and second compression rings have been widely used in small engines, and appear to reduce liner wear but not to the same extent as chromium plating the liner. In some cases liner wear has been reduced about 80 per cent by the use of chromium plated rings, but chromium plated liners have given four to ten times the life of normal, good quality, cast iron liners.

The higher the liner skin temperature the less the possibility of condensation and, therefore, jacket temperatures should be kept as high as reasonably practicable. Too many marine engines operate at very low jacket temperatures, this being particularly true of small vessels, such as trawlers, tugs, etc., where direct salt water cooling is employed. Jacket water outlet temperatures as low as 80 deg. F. have been recorded in fishing vessels in North Atlantic waters. It is, of course, difficult to maintain high jacket temperature in such installations without severely restricting the water flow. This can be dangerous, as uncooled pockets may be left in the cylinder heads, finally causing fracture. With large engines using a closed fresh water cooling system it is much easier to maintain suitable inlet and outlet temperatures. It is desirable to maintain the maximum water flow with the minimum difference between inlet and outlet temperatures. About a 10 deg. F. differential between inlet and outlet should be aimed at. Outlet temperature of about 160 deg. F., which is considerably higher than many engines operate at, should be treated as quite normal.

If possible, heat should be applied to the cooling water before starting (as is usual with Doxford engines), so that the liners are warmed through. This materially reduces starting wear and also lessens the risk of fracture of the engine parts due to too rapid expansion when first started.

Suitable lubricants can materially reduce corrosive wear in two ways. Firstly, if a tenacious oil film can be maintained on the liner walls, there is less likelihood of corrosive attack on the metal. The addition of oiliness agents should materially assist in this respect, particularly after the engine is stopped. Straight mineral oils tend to drain fairly rapidly from hot surfaces, but polar bodies which form tenacious adsorbed oil

TABLE IV—INSPECTION DATA ON TYPICAL HEAVY FUEL OIL USED IN MARINE DIESEL ENGINES

Sampling point	ex bunker	After purifier	After clarifier	Main engine fuel pump
Specific gravity at 60 deg. F.	0·963			
Redwood 1 viscosity at				
100 deg. F. secs.	1,595	—	—	1,615
122 deg. F. secs.	743	—	—	755
140 deg. F. secs.	438	—	—	436
200 deg. F. secs.	116	—	—	112
Flash point P.M. closed deg. F.	218	—	—	—
Pour point upper, deg. F.	15	—	—	—
Conradson carbon, per cent	11·5	—	—	—
Water, per cent (Dean and Stark)	0·2	0·15	Less than 0·1	Less than 0·1
Sediment by extraction, per cent	0·04	0·03	0·02	0·02
B.S. & W., per cent	0·5	0·35	0·1	0·1
Ash, per cent	0·06	0·04	0·03	0·03
Sulphur, per cent	2·3	—	—	—
Asphaltenes, per cent	7·6	—	—	—
Calorific value, BTU/lb. gross	18,800	—	—	—

films, remain on the metal for much longer periods. This frequently prevents starting up again with a dry liner.

The second way in which lubricants can combat corrosion is by including acid-neutralizing additives in the oil. Many detergent oils incorporate additives which to a degree act as neutralizing agents besides reducing ring zone deposits.

Abrasive Wear

Generally speaking, there is less chance of abrasive foreign matter being drawn into the engine with the induction air in marine engines than in equivalent engines operating ashore, as the atmosphere is much cleaner. The problem of abrasive materials formed by incomplete combustion, however, is common. Engines operating on residual fuel are bound to suffer higher abrasive wear due to the higher initial asphaltene and ash contents of this type of fuel. No matter how carefully the fuel is cleaned it is impossible to remove entirely materials of this type; in fact, to remove all the asphaltenes would involve a considerable loss of fuel, in some cases up to 12 per cent! Complete combustion of asphaltenic matter is possible but it is likely that some of the heavier constituents are only partially burnt and the oxidized asphalt can be hard and abrasive.

It would appear that the main cause of increased abrasive wear in engines burning residual fuels is the comparatively high ash content. Ash is material which will not burn completely even at temperatures exceeding 3,000 deg. F. in a Diesel engine cylinder. It is an extremely difficult matter to analyse ash forming constituents in fuels, partly due to their complexity, and partly to the small amounts present. For example, the ash content of a typical 1,500 second residual fuel is in the order of 0·04-0·08 per cent. An added difficulty is that to extract such materials from a fuel sample, the oil must be burnt off and the various elements remaining are combined with oxygen. It is possible, therefore, to identify the basic elements but not necessarily the compounds originally present in the fuel. As a simple example, sodium chloride after ashing is converted into sodium oxide. Elements such as iron, aluminium, barium, vanadium, silicon, calcium, sodium, magnesium and several others have all been identified but not necessarily from any particular fuel. The percentage of individual compounds present can vary considerably, but sodium sulphate, sodium chloride, iron oxide and silica are the most important. Some ash compounds may be present as finely divided foreign matter in the fuel, but the majority are from the fuel itself.

Ash-forming constituents can be present in both soluble and insoluble form in residual fuels, the former being essentially organic compounds of metals while the latter consist mainly of finely divided iron oxides and silica. Efficient centrifuges will remove the greater part of the insoluble ash, but soluble ash will be carried through to the engine in the fuel. Table IV gives inspection data on a typical residual fuel before and after centrifuging and it will be noted that the sediment by extraction has been considerably reduced and that the ash content has been halved. Although the figures expressed as a percentage appear small, when translated into actual weights they look much more significant. For example, in the case of a motorship burning 35 tons per day, this would represent about 23½lb. of incombustible abrasive material entering the engine per day, apart from foreign matter brought in with the induction air.

As a matter of interest, analyses and weights of sludge taken from the purifier and clarifier during an engine trial on this fuel are given in Table V. In this case it will be seen that the purifier is doing the bulk of the work in removing solids and water from the fuel.

TABLE V—ANALYSES AND WEIGHTS OF SLUDGE SAMPLES TAKEN DURING TRIALS USING FUEL SHOWN IN TABLE IV

Origin of Sample	Sludge from purifier	Sludge from clarifier
Weight of fuel treated, tons	15	15
Centrifuging temperature, deg. F.	177	175
Approximate treating rate, tons per hour	2·75	2·75
Total weight of sludge, lb.	4·2	0·3
Total weight of sludge, per cent of fuel treated	0·013	0·001
Analysis		
Free water, per cent	6·3	Nil
Percentage of sample after removal of free water		
Water	23	20
Oil	30	53
Ash	28	10
Carbonaceous material*	19	17
Lb. of dry solids removed per ton of fuel treated	0·13	0·005

*Determined by loss on ignition.

It has usually been found that liner wear in large engines burning marine Diesel fuel was fairly heavy during the first few hundred hours' operation, after which it settled down at a much lower rate per 1,000 hours' running. It would appear, however, that this is not the case in similar engines burning residual fuels. Fig. 19(a) shows typical wear figures shown in the form of a profile diagram from a Doxford engine which has operated on fuels ranging from 500 to 1,500 seconds Redwood No. 1 at 100 deg. F. since new. Fig. 19(b) shows maximum wear figures from two cylinders of the same engine plotted in more conventional form. Wear during the first 1,000 hours is high, after which the rate falls slightly, then increases more gradually until after about 12,000 hours the liner is reaching the end of its useful life.

It may well be that abrasive material prevents the formation of a work-hardened surface and that the abraded surface is susceptible to corrosive attack. It is interesting to note that the maximum wear is greatest in way of the top compression ring at inner dead centre.
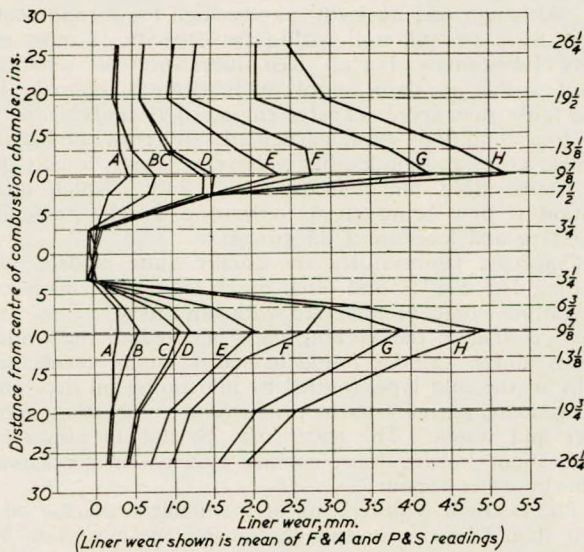
FIG. 19(a)—*The effect of total operating time on liner profile*

(A) 994 hr;   (B) 2,661 hr.;   (C) 4,261 hr.;   (D) 5,863 hr.;
(E) 7,654 hr.;   (F) 8,725 hr.;   (G) 11,000 hr.;
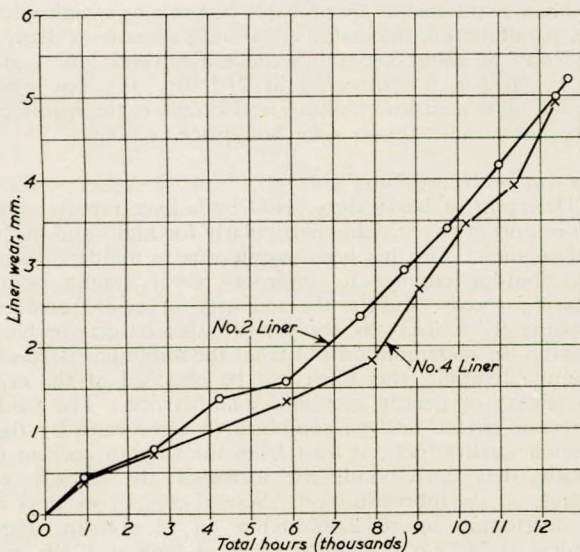(H) 12,683 hr.

FIG. 19(b)—*Maximum liner wear plotted against operating hours*

Scuffing

Scuffing is a form of roughening of the liner surface and is caused by metal-to-metal contact between piston rings or piston skirt and the liner. As previously mentioned, two similar hard metals do not form a good bearing combination and, of course, rings, piston and liner are usually of cast iron, although different compositions are usually employed. If metal-to-metal contact occurs, surface asperities may break off and the dislodged particles are highly abrasive, so that once the surface is broken, the scuffing can quickly spread. In extreme cases the local temperatures become so high that surface welding occurs with eventual seizure.

A frequent cause of scuffing, often overlooked, is that oil scraper rings may be given too sharp an edge. In addition, if the ring pressures are high the sharp edge can literally cut through the oil film. The fine edge is easily broken off and the particles readily cause abrasion. Similarly, compression rings having 90 degree machined corners tend to scrape off the oil film instead of spreading it. To avoid this, both top and bottom edges should be given a definite chamfer.

Obviously, the better the lubricant, the less chance there is of metal-to-metal contact and subsequent scuffing. Surface finish of liner and rings also has an important influence, as the rougher the surface the thicker must be the oil film to prevent metal-to-metal contact. It has been found, however, that in some cases a superfine finish has resulted in high initial wear. The method of finish-machining the liner appears to have an important bearing on this matter. Finish-grinding, for example, gives an excellent surface finish but it is difficult to obtain a parallel liner from top to bottom, as the grinding wheel wears as it progresses from one end to the other. It is possible, too, that abrasive material is left in the pores of the metal, which is subsequently released when the engine is working. Honing gives possibly the best results after finish machining, as surface asperities are gradually smoothed over and the resultant surface is highly retentive for lubricating oil. Unfortunately, honing is too expensive to apply to large liners. An excellent finish is obtained by taking a high-speed, fine-finishing cut with a diamond or carbide tipped tool. This seems to give a retentive surface for lubricant and yet not too rough so as to present a risk of surface contact under normal running conditions.

Other Factors Influencing Wear

It is only possible to cover very briefly some of the other wear hazards in a paper of this type as many factors are outside its scope. For example, liner material is a study in itself and is the joint province of the metallurgist and engine designer, while workmanship is the responsibility of the engine builder.

Piston Design and Cooling

Design of pistons and piston rings is a subject to which much more attention could well be paid. Heat from the piston crown is removed either by transfer to a cooling medium such as water or lubricating oil, or down the side walls to the rings and piston skirt and thence through the oil film and liner walls to the jacket-cooling water. In a suitably designed piston the greater part of the heat is removed by the coolant. This is highly desirable, as with a reduced amount of heat transferred through the piston rings there is much less chance of excessive deposits causing ring sticking.

Water is several times as efficient a cooling agent as lubricating oil and from this point of view is to be preferred for piston cooling. Unfortunately, however, it is extremely difficult to prevent leakage from telescopic pipe for swinging arm glands and water in the crankcase oil is most undesirable. For this reason the majority of engines now employ oil cooling. Where water is employed, as in the Doxford engine, treated distilled water is used in a closed circuit.

In oil-cooled pistons a high rate of circulation is essential to avoid overheating of the piston crown. The lower the viscosity of an oil the better is its heat transfer properties, and from this point of view a high viscosity oil as sometimes used for

cylinder lubrication would be unsuitable for piston cooling. A poor quality oil tends to form carbonaceous deposits on the underside of the piston crown which can seriously reduce heat transfer and lead to cracking of the metal. Detergent oils are much superior to straight oils for piston cooling, but where the oil is drawn from the bearing system other factors might prevent their use.

Piston ring clearance is important; too great a vertical clearance can cause excessive ring fluttering and eventual breakage, too small a clearance causes the rings to bind in their grooves when hot, and even a small amount of deposits can cause ring sticking.

Ring Zone Deposits

Probably the most common trouble in Diesel engines of all types is sticking or sluggish piston rings. Some of the causes have been briefly covered but the influence of the lubricating oil itself must now be considered although this cannot be dissociated from incomplete combustion deposits from the fuel. Attempts have been made to distinguish between carbonaceous deposits from the fuel and those from the lubricating oil, but these have not been very satisfactory. It is generally accepted, however, that the greater part of the deposits formed in the ring zone are from the lubricating oil.

Partially-burnt carbonaceous matter, ash, and possibly sulphur compounds from the fuel, foreign matter introduced with the induction air, together with carbonaceous compounds from the lubricant, accumulate behind and on the sides of the rings and bake there, causing sluggish action, and finally prevent movement. When this occurs, engine efficiency is reduced, combustion troubles may be experienced due to lack of compression, "blowby" becomes troublesome and the crankcase oil becomes contaminated.

In two-cycle engines, additional trouble can be caused by deposits building up in the ports, restricting air and exhaust gas passages and reducing efficiency still further. Although the actual carbon content of ring zone and port deposits may be relatively small, it is usual to call all such deposits by the general term "carbon". The thin brown to black deposits frequently found on piston skirts are termed varnish and are mainly caused by lower temperature oxidation of the lubricant. Varnish deposits are not usually harmful and may actually be an added protection against metal-to-metal contact between piston and liner.

The nature of the lubricating oil used can have an important influence on the nature and amount of ring zone deposits. Paraffinic oils are less volatile than naphthenic or aromatic type oils, they have high viscosity indices, they possess excellent resistance to oxidation, and strongly resist sludging. Carbonaceous deposits form rather more slowly with paraffinic oils than with naphthenic oils but, unfortunately, the carbon formed is hard, close grained and slaty in nature. Naphthenic oils, on the other hand, while not being so stable nor possessing such good temperature/viscosity characteristics, form a much softer carbon and usually less of it.

Selection of a Cylinder Lubricant

From the cylinder point of view, therefore, the most suitable oil would combine the good properties of both paraffinic and naphthenic oils whilst eliminating their weak points. Considerable advances have been made in this direction and whilst it is impossible to change the nature of the deposits formed by paraffinic oils, it has been found possible by solvent refining selected naphthenic crudes and then adding suitable antioxidants, greatly to improve their oxidation stability and raise the viscosity index to about 70.

The advantages and disadvantages of including bright stocks have been touched on but these are still widely used to improve lubricity and to raise the viscosity index when blended with naphthenic oils. As an alternative, distillate paraffinic oils can be blended with naphthenic oils to increase the viscosity index to about 80. Probably the best solution is to incorporate a suitable oiliness agent in a blend of solvent-refined naphthenic and paraffinic distillate base oils.

Viscosity requirements have been outlined but this is also influenced by whether the same oil is required to lubricate both bearings and cylinders. In large engines fitted with separate mechanical cylinder lubricators this is largely a question of convenience in storage and handling, but in smaller engines employing splash, spray, or mist lubrication from the crankcase, there is no choice in the matter. Oils with a viscosity as high as 450 seconds Redwood No. 1 at 140 deg. F. (25·7 centistokes at 210 deg. F.) are still recommended by one leading engine builder but the tendency now is to use a single oil for both engine and crankcase and the viscosity is usually in the order of 160-250 seconds Redwood No. 1 at 140 deg. F. (10·8-16·84 centistokes at 210 deg. F.) for large marine engines.

Bearing Lubrication

Lubrication of bearings, gears, chain drives and other working parts of larger or medium sized Diesel engines is a comparatively simple matter compared with cylinder lubrication. Although bearing loads may be high for a reciprocating engine, these are still well within the capacity of most good quality lubricants. Forced feed lubrication is universally applied so that an ample supply of lubricant is assured. Due to the fairly slow speeds of large engines it is unlikely that full fluid film conditions exist, particularly in crosshead bearings, eccentric straps and guides, but as bearing wear is usually quite low, partial fluid film lubrication is likely. Much greater attention is now being given to bearing design, particularly to the type and location of oil grooves.

Crankcase temperatures are usually quite modest, rarely exceeding 160 deg. F. and often much lower. In many cases temperatures could be increased with advantage. This would reduce oil drag in the working parts and reduce the tendency to form sludge. Sludge found in marine engine crankcases is usually of the cold type, formed by lubricating oil decomposition products, partially burnt products of combustion, foreign matter and water. The water may be due to atmospheric condensation, leakage from cooling systems or condensation of blowby water vapour.

The essential requirements of a suitable crankcase oil are that it should be viscous enough to carry the maximum loads encountered but not so viscous as to cause unnecessary viscous drag, it must possess the maximum resistance to oxidation and formation of sludge, it must not emulsify readily with water, it must not form excessive deposits when used for piston cooling, and it must be readily pumpable at all normal temperatures.

These requirements are probably best met by a high viscosity index paraffinic oil, preferably containing an anti-oxidant, with a viscosity of about 160-180 seconds Redwood No. 1 at 140 deg. F. (10·8-12·9 centistokes at 210 deg. F.), but where a common oil is used for crankcase and cylinders the requirements of the latter must always take first place.

Detergent or Heavy Duty Oils

Detergent or heavy duty (H.D.) oils have rapidly come to the fore during recent years, particularly for high- and medium-speed engines. This has been mainly due to builders designing higher output engines to improve power/weight ratio, by increasing speeds, and in the majority of cases considerably increasing the b.m.e.p. by the use of superchargers or blowers. Although the maximum output from the minimum sized engine is highly desirable, this must not be obtained at the expense of reliability or greatly increased maintenance. The tendency to increase engine ratings, combined in some cases by the use of poorer quality fuels, at least from the sulphur content point of view, has correspondingly increased the already severe demands on the lubricating oil. Several engines are now available developing up to 2,400 b.h.p. in 16 × 10-in. diameter cylinders at 750 r.p.m., with b.m.e.p. at high as 125lb. per sq. in. Engines of this type are being widely used for driving generators and in Diesel-electric locomotives and will

undoubtedly be used for marine purposes on an increasing scale in the future.

Mainly due to the greater heat release in the cylinders, ring zone temperatures have increased and there is a much greater possibility of ring sticking and piston skirt lacquering. Crankcase temperatures are higher and oxidation of the lubricant is correspondingly more rapid.

In the late 1930's certain American high-speed Diesels, mainly of the automotive type, gave a considerable amount of trouble with excessive ring sticking, port blockage in two-cycle engines and rapid oil deterioration in the crankcase. Copperlead and other hard alloy bearings, which were introduced to carry the heavier engine loads, also gave considerable trouble with corrosion. As a result of this, new types of lubricants combining special additives were developed which were much more stable at high temperatures and also had the property of dispersing carbon deposits in the ring zone, the deposits being kept in an extremely fine suspension in the oil. In some cases separate dispersant or detergent additives as they are usually termed, and active oxidation inhibitors were blended in a suitable base oil, while in other cases a dual purpose antioxidant and detergent additive was used. Some oils also incorporated a bearing corrosion inhibitor. These new oils were termed H.D. oils as they were intended for heavy duty use.

U.S. Army Ordnance Specifications

The value of the new oils was recognized by the American Military Authorities who introduced a specification known as 2-104A in April 1942, which was later superseded by 2-104B in May 1943. It might be more accurate to say that a set of engine test conditions were prescribed which could only be met by one of the then recently introduced heavy duty oils, combined with certain physical requirements. This specification was the first serious attempt to set up a standard of performance for detergent oils. Oils submitted for test were subjected to a thirty-six hour screening test, essentially for bearing corrosion, in a Chevrolet petrol engine, followed by further tests in Caterpillar and General Motors Diesel engines. Following the tests the engines were stripped and the parts examined by a committee who decided whether or not the oil was up to standard. If passed by this reviewing board, the oil was given a numbered certificate and placed on the approved list.

It was found, however, that certain engines of American origin did not give satisfactory performance when operating on 2-104B oils for long periods under heavy loads. It also became apparent that the actual engine test procedure left a lot to be desired. For instance, the specification did not stipulate the ignition quality of the test fuel, nor its sulphur content. It was possible, therefore, by using a specially prepared low sulphur fuel for a poor quality detergent oil to obtain approval.

The U.S. Army Ordnance then introduced an amendment to the 2-104B Specification which became known as 2-104B Supplement 2 (sometimes referred to as Series 2). More severe engine tests were prescribed, including a 500-hr. test in a single-cylinder supercharged four-cycle Caterpillar engine, and other field and bench tests specified by Caterpillar, using fuels of 1 per cent minimum sulphur content. Whereas about $2\frac{1}{2}$ per cent of suitable detergent additive in a good base would meet the requirements of 2-104B, 12-16 per cent of additive was required to meet the more stringent conditions of Supplement 2. The cost of the new oils was greatly increased and in some cases the useful life was considerably reduced.

After considerable experience with both 2-104B and 2-104B Supplement 2 detergent oils, the Army authorities came to the conclusion that the majority of the petrol and Diesel engines which they operated did not require oils of such extreme detergency as Supplement 2, and yet required better oils than 2-104B. A compromise was decided upon and an intermediate level of detergency was introduced under the designation 2-104B, Supplement 1 (or Series 1). Engine test conditions were relaxed somewhat, although the Caterpillar engine test on 1 per cent sulphur fuel was retained. The test period, however,

was reduced to 480 hours and an unsupercharged engine used. To pass this new test about 6-10 per cent detergent additive was required.

At this stage, there were thus three different Army specifications for detergent oils and the situation was most confusing. The matter was thoroughly investigated and field results on the three different types of oil were closely examined. As a result of this the U.S. Army Ordnance introduced in 1951 a single new specification, designated MIL-0-2104, to take the place of all three earlier specifications and until recently this was used by the U.S. Army for all types of internal combustion engines. The specification covers tests on S.A.E. 10, 30 and 50 grades, the S.A.E. 30 grade being the oil most widely used. The quality of the oil is specified in terms of conventional physical tests and, in addition, before an oil is approved it must give a certain standard of performance in two engine tests. The first is a thirty-six hour test in a Chevrolet petrol engine, known as the L-4 engine test, in which the sump temperature is maintained at 280 deg. F. and the jacket water outlet temperature at 200 deg. F. This test is essentially a high temperature oxidation test to determine the oil stability and tendency to corrode copper-lead bearings. The second test of 480 hours' duration, known as the L-1 engine test, is carried out in a single-cylinder unsupercharged Caterpillar engine. The test fuel is carefully defined and must have a minimum sulphur content of 0·35 per cent.

MIL-L-2104A

The MIL-0-2104 specification has been under review for some time and slight modifications have now been made. A new specification, MIL-L-2104A, dated February 1954, has now superseded MIL-0-2104. The essential alterations which have been made are that the S.A.E. 10 grade must have a viscosity (extrapolated) not exceeding 12,000 S.U.S. at 0 deg. F. and the S.A.E. 30 grade must not exceed 150,000 S.U.S. (extrapolated) at 0 deg. F. Furthermore, the S.A.E. 10 grade must now have a pour point not exceeding 20 deg. F. as this grade is intended essentially for operations under arctic conditions.

It will be appreciated that 2-104B, Supplement 1 and Supplement 2 detergent oils have no longer any official significance as the specifications are obsolete. They do represent, however, a standard of performance which is still recognized by the petroleum industry, by engine builders and by operators.

It is the author's opinion that an oil which passes the MIL-L-2104A Specification is suitable for all current British medium- and high-speed Diesel engines operating on light distillate fuels, unless the conditions are exceptionally severe.

It is not intended that the actual composition of detergent additives should be discussed here, as many of them are trade secrets, but in general the most effective types contain metallic compounds of barium, calcium, or phosphorus. All these compounds can form ash compounds when burnt in a Diesel cylinder and although these are removed from oil-bathed surfaces of the liner, piston and piston rings, it is possible for such compounds to build up on exhaust valve faces, seats and stems. It is advisable, therefore, to take all the operating factors into consideration and use an oil with the lowest additive content which will give satisfactory engine performance.

Navy Specifications

About the same time as the U.S. Army Ordnance introduced the 2-104B specification the U.S. Naval Authorities brought out the 9000 Series of oils known as Navy Symbol 14-0-13A. As with 2-104B, the Naval specification laid down physical limits for the oil, together with engine test requirements, and after an oil had been tested the component parts of the engine were examined by a reviewing board. The engine tests included a submarine engine test together with a salt water test. The additive requirements for the oil were about the same as for 2-104B oils. The Navy 14-0-13A specification was superseded in July 1952 by Interim Military Specification Lubricating Oil, Diesel Engine MIL-L-9000A (Ships). The

new interim specification covers two classes of Diesel engine lubricating oils, namely Class A and Class B.

Class A covers an oil suitable for use in Diesel engines operating under severe conditions conducive to high rates of wear and engine deposits, such as low temperature operation, high load operation and the use of high sulphur content fuels. This oil is intended to provide satisfactory lubrication and protection against rusting in Diesel engines and associated equipment in which crankcase oil contamination with either salt or fresh water is likely to occur.

The engine tests have to be carried out on one or more viscosity grades to determine the effectiveness of the oil in prevention of excessive ring sticking, clogging of oil channels, lacquer, carbon or sludge deposits and the tendency of the oil to minimize engine corrosion and wear. In addition, oils intended for submarine service have to show satisfactory performance in a prototype submarine engine test. The standard tests include a 480-hr. test without oil change in a G.M.-71 engine using 1·0 per cent minimum sulphur fuel, a jacket temperature of 130 deg. F. and an engine back pressure of 2-in. mercury. Furthermore, 2 per cent sea water is added to the crankcase at intervals during the test. A Caterpillar engine test is also included.

Class B covers lubricating oils suitable for use in Diesel engines (including submarine type) operating under normal loads and temperature conditions with low deposit forming (low sulphur content) fuel. The engine tests include a 300-hr. test in a G.M.-71 engine with a 0·35 per cent minimum sulphur content fuel. Test conditions are less severe than for Class A fuels as the jacket outlet temperature is increased to 175 deg. F. and the oil is changed at 120-hr. intervals. The inclusion of 2 per cent salt water is retained, however. An additional Caterpillar 1-A engine test is also included.

The requirements for Class A lubricants are extremely severe and are intended for engine conditions more difficult than those so far encountered in naval vessels in peace time. In submarines operating under wartime conditions using a Snorkel tube for air induction, such lubricants might, however, be necessary. So far the specification for Class A oils is in abeyance but could be invoked when required. It is expected that a detergent additive content of at least 10 per cent would be needed for this type of lubricant, plus an effective corrosion inhibitor, whereas about half this additive content is required for Class B oils. The latter is, therefore, roughly in line with MIL-L-2104A except for the more stringent corrosion tests (including a silver corrosion test).

The Royal Navy introduced a 9000 Series oil in 1943 for all types of internal combustion engines, including submarines. So far, these comparatively low additive content oils have given satisfactory service. The question of whether or not the requirements for oils submitted for approval should be increased is at present under review and it is thought likely that they will eventually be brought into line with U.S. Navy MIL-L-9000A, Class B lubricants.

DEF/2101 Specification

This specification was introduced by the Ministry of Defence in August 1951 to cover the requirements of the British Forces for detergent oils and it is generally similar to MIL-L-2104A. It covers S.A.E. 20, 30 and 50 grades which are designated OMD-60, OMD-110 and OMD-330 respectively. The engine tests used are based on the L-1 Caterpillar and the L-4 Chevrolet tests as for MIL-L-2104A, but the test results and engine parts are assessed by the Ministry of Supply Review Board.

Unfortunately, although the specifications are so similar, an oil approved by the U.S. Army Ordnance under MIL-L-2104A is not necessarily accepted by the Ministry of Supply and *vice versa*. It is hoped, however, that in time the two specifications will become interchangeable.

DEF-2101A

The original DEF-2101 specifications has been superseded by DEF-2101A (effective June 1953). The new specification includes an S.A.E. 10 grade and the viscosity requirements for the S.A.E. 10, 30 and 50 grades are now almost identical with those of the MIL-L-2104A specification. Similarly, a maximum viscosity at 0 deg. F. has been stipulated for the S.A.E. 10 grade. Physical requirements for oils complying with DEF-2101A and MIL-L-2104A are given in Table VI and the similarity is obvious.

Use of Detergent Oils in Marine Engines

Shipowners generally have been somewhat slow to employ these new types of oils but not without some reason. So far, heavy duty oils have been used in the main for high- or medium-speed engines, and although Diesel generators come within the latter category, the main propulsion unit is a somewhat different problem. For convenience in storage it is obviously better if the same lubricant can be used in both main engines and Diesel auxiliaries. Furthermore, the possibility of water leakage into the oil is always more likely aboard ship

TABLE VI—PHYSICAL REQUIREMENTS OF OILS MEETING MIL-L-2104A AND DEF-2101A SPECIFICATIONS

MIL-L-2104A.

Property	Grade 10 (SAE. 10)	Grade 30 (SAE. 30)	Grade 50 (SAE. 50)
Viscosity at 210 deg. F. Kinematic, centistokes	5·44 to 7·29	9·65 to 12·98	16·83 to 22·75
Saybolt Universal, Seconds	44 to 50	58 to 70	85 to 110
Viscosity at 0 deg. F. (extrapolated). Kinematic, centistokes, maximum	2,614	43,570	—
Saybolt Universal, Seconds	12,000	200,000	—
Viscosity index, minimum	—	—	75
Pour point, deg. F., maximum	minus 20	0	Plus 15
Stable pour point, deg. F., maximum	minus 20	—	—
Flash point, deg. F., maximum	360	390	400

DEF-2101-A

Property	OMD-40 (SAE 10)	OMD-60 (SAE 20)	OMD-110 (SAE 30)	OMD-330 (SAE 50)
Viscosity at 210 deg. F. Kinematic, centistokes	5·5 to 7·3	7·3 to 9·7	9·7 to 13·0	16·8 to 22·8
Saybolt Universal, Seconds	44·2 to 50·0	50·0 to 58·2	58·2 to 70·1	84·9 to 110·2
Viscosity at 0 deg. F. (extrapolated). Kinematic, centistokes	2,600	10,000	33,000	—
Saybolt Universal, Seconds	12,080	46,460	153,300	—
Viscosity index, maximum	—	—	—	75
Pour point, deg. F., maximum	minus 20	minus 5	0	15
Flash point, deg. F., minimum	360	360	390	400

than in similar engines ashore and the effect of water, particularly salt water, on detergent oils must be considered.

If, as is fairly common practice, used oil from the generators is batch filtered and used as make-up for the main engine, what would be the effect of adding used detergent oil to a straight oil?

Consider the effect of water on detergent oils first. It is rather difficult to be specific on this point as different detergent additives vary considerably in their solubility in water. Those with which the author is most familiar are virtually insoluble in water but there are other additives which are highly soluble. Furthermore, although some detergent oils when new possess good demulsibility, as the oil does its work, the insolubles content increases and, of course, natural oxidation of the oil takes place at the same time. It is therefore true to say that, generally speaking, a used detergent oil separates less readily from water than a used straight oil, but the difference can vary considerably. In the case of generators or similar types of Diesel engines used aboard ship, the risk of serious water leakage in a well-maintained engine is not very great as compared with the main engine.

On balance, therefore, suitable detergent oils could well be used for auxiliary Diesel engines, enabling the engines to be run for much longer periods without the need for decarbonizing.

Maintenance can thus be greatly reduced, an important point these days when experienced engineers are in short supply. Oil change periods can usually be extended, and with the normal type of ship's Diesel generator operating at speeds up to about 500 r.p.m. there is no reason why the oil should not be used for periods of upwards of 2,000 hours running. Provided that the oil is changed before the end of its useful life it should only be necessary to drain the old oil when hot and refill with new oil, without stripping the engine.

With the main engines several factors have to be considered. With engines operating on marine Diesel fuel, ring sticking is not usually a serious problem, provided that a suitable straight oil is used in the cylinders. Wear, although fairly high, is acceptable. In the case of engines burning residual fuel, increased deposits and higher wear rates are becoming a serious problem, as already discussed. The question is, therefore, will detergent oils keep rings free, reduce port deposits in two-cycle engines and reduce wear. Unfortunately, our knowledge on this particular subject is somewhat scanty. Until recently it was generally accepted that a good oil flow over the rings was essential to flush away deposits in the oil. Such conditions exist in trunk piston engines relying on splash or spray lubrication of the cylinders. In large engines employing separate cylinder lubricators, however, it was believed that
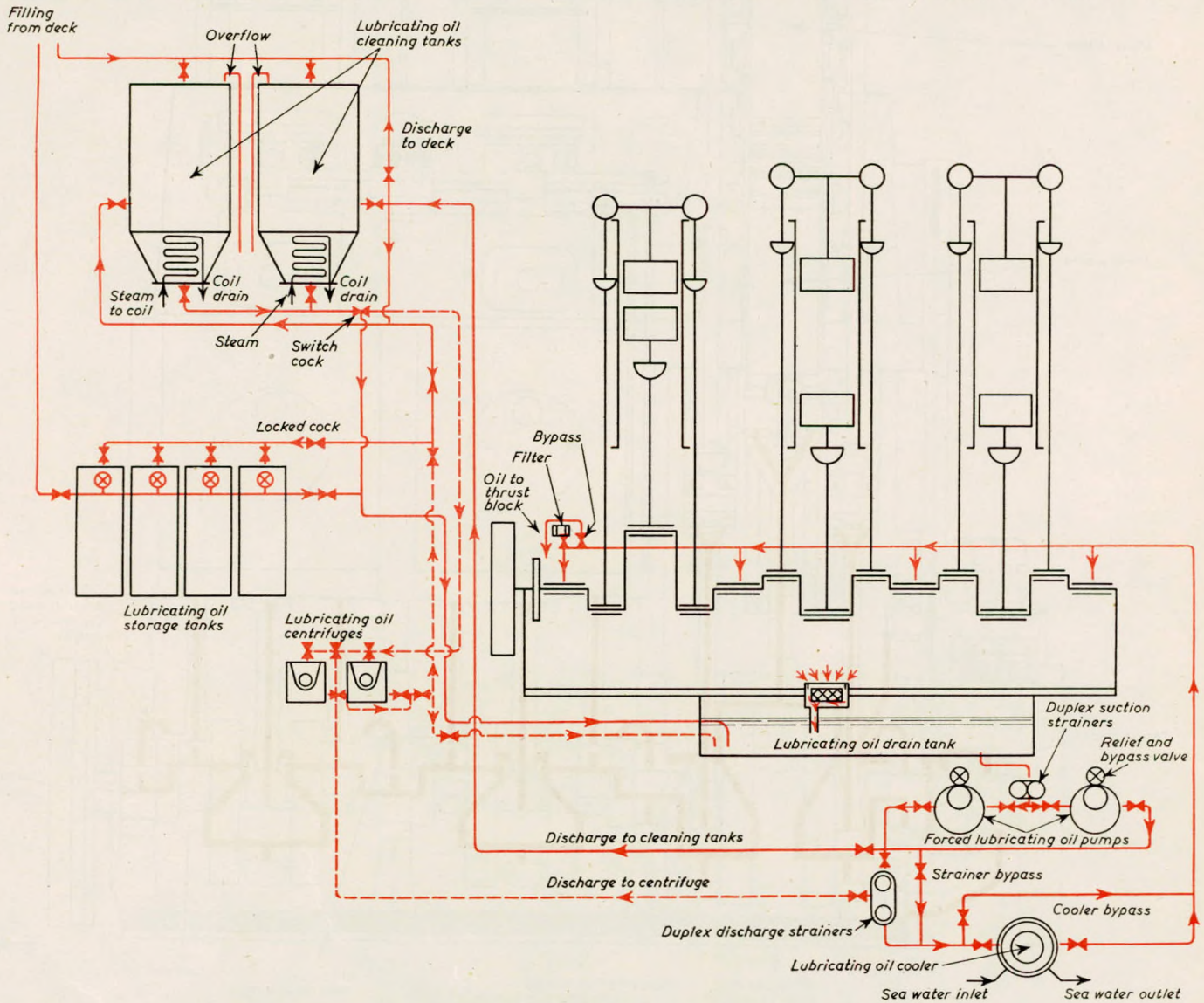

FIG. 20—*Diagrammatic layout of Doxford lubricating oil system*

there was insufficient oil flow over the rings to remove deposits. A number of tests carried out have indicated, however, that good detergent oils do considerably reduce deposits although it is somewhat early to claim that improvements can be obtained in all types of engines. More extensive tests are required in different engines and it is hoped that shipowners will co-operate in this matter as it may well be that detergent oils will greatly improve engine performance.

On the question of wear in smaller engines, detergent oils have been found to reduce liner wear but not to any great degree. The incorporation of load-carrying additives into a good detergent oil has, however, in several cases reduced wear by 50 per cent. Information regarding the effectiveness of

such oils in large engines is again scanty but one or two preliminary tests have shown encouraging results.

It is not considered necessary or desirable to use detergent oils as crankcase lubricants in large engines, mainly due to the increased possibility of water leakage, particularly where water-cooled pistons are used. In oil cooled pistons, however, the inside of the crowns is kept completely clean with detergent oils. Although where a detergent oil is employed as a cylinder lubricant with a straight oil in the crankcase, there is a possibility of some of the former finding its way into the sump, this is not likely to be detrimental. On the other hand, it is not advisable for used detergent oil from auxiliary engines to be used as make-up for the main engines. In such cases there
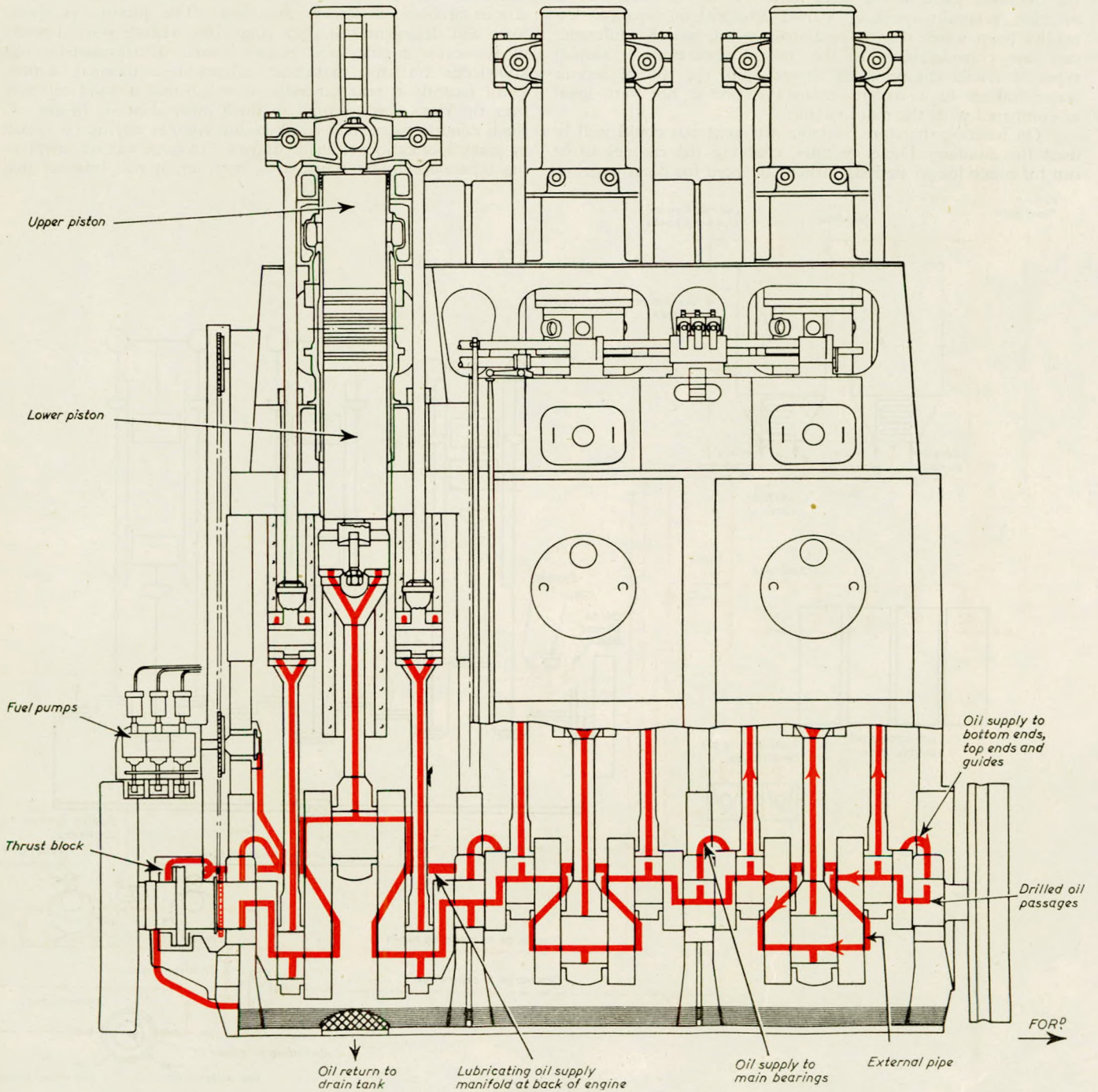

Fig. 21—*Lubricating system of typical Doxford engine*

would be a danger of the detergent oil freeing deposits previously formed in the crankcase. These could well choke oil filters or pump suctions with serious results. For the same reason a detergent oil should never be put into a dirty engine of any type.

The fact that the Royal Navy has successfully used detergent oils in marine Diesel engines, including submarine engines, with very satisfactory results, proves that this type of oil is eminently suitable for use aboard ship and will undoubtedly be used much more widely in the future in merchant ships, for both main and auxiliary engines.

Lubricating Oil Systems

As space only permits a limited description of the forced lubrication systems employed in large Diesel engines, three types widely used in the British Merchant Service, namely the Doxford engine, the Sulzer engine and the Harland-Burmeister and Wain engine, will be covered in brief.

Doxford Engine

A diagrammatic sketch of a typical Doxford engine lubricating oil system is shown in Fig. 20. This illustrates the three after-cylinders of a large six-cylinder engine, in which three scavenge pumps are driven by rocking levers from the engine crossheads. In some engines the lubricating oil pumps are driven by the scavenge pump crosshead, but, in this instance, duplicate independent rotary type pumps are used. One pump is capable of handling the engine requirements with the second kept as standby. The pump takes its suction from a large oil drain tank located in the double bottom, and the oil passes through suction strainers before being discharged through discharge strainers to the oil cooler. From the cooler the oil passes to a main distribution pipe located at the back of the engine. Oil is fed to the top half of each main bearing by external pipes, then through drilled holes and external pipes to the side and centre bottom-end bearings. From the bottom-end bearings the oil passes up drilled passages in the connecting rods to the crosshead bearings and guide shoes.

The passage of the oil can be more clearly followed in Fig. 21. From the bearings the oil drains back into the crankcase and thence through a coarse strainer back into the drain tank. The Michell thrust block, located at the after end of the engine casing, is supplied from the main manifold, the oil first passing through a fine filter to exclude abrasive material which would damage the thrust pads. The timing chain, sprockets and bearings, the scavenge-pump rocking-lever bearings and the Doxford-Bibby detuning-wheel grid spring are also lubricated from the main system.

The upper piston transverse-beam centre bearing is oil lubricated by means of flexible rubber hoses led over circular grooved quadrants. The return is led back into the crankcase. The upper piston tubular guides are grease lubricated by means of grease cups fitted to the outside of the guides.

Both upper and lower pistons are lubricated by mechanical lubricators, four to six oil feeds, depending on the size of the cylinder, being located equidistantly around the liner circumference. Lubricating oil injection is timed so that the top and second rings straddle the feed points at the inner end of their stroke. Mechanical lubricators are also used to supply oil to the scavenge pump pistons.

Two oil cleaning tanks are fitted in the upper part of the engine room, being capable of holding the complete oil change and also the usual clean oil storage tanks. Duplicate centrifuges are fitted for oil cleaning. If it is desired to batch filter the oil in port, the oil is pumped by the main oil pumps into the cleaning tanks, each of which is fitted with a steam heating coil. The oil is heated to about 120-140 deg. F. for a period depending on the condition of the oil and any water or sludge present is drained off at intervals. The cleaned oil can be returned by gravity back to the drain tank or it can be passed to the centrifuge where it is further cleaned before entering the tank.

Deck connexions are fitted to enable the storage tanks and

engine tank to be filled direct by gravity and the latter line can also be used to pump out the entire oil charge if necessary.

During normal operation the centrifuges are used on the usual bypass system, an oil lead from the main discharge strainers supplying the oil to a small heater before entering the machine. The cleaned oil is then returned to the drain tank. Further notes on crankcase oil maintenance are given later in this paper.

Sulzer Engine

Longitudinal and transverse views of a typical Sulzer Type SD two-cycle engine showing the lubricating oil system in detail are given in Figs. 22(a) and 22(b). The system is necessarily more complicated than in the Doxford engine as oil is used for piston cooling, fuel valve cooling and guide cooling. The engine is of the dry sump type, the oil drain tank being located beneath the engine in the double bottom. Independent, electrically operated, duplicate-gear type lubricating oil pumps, together with filters and a large, vertical, oil cooler, are fitted. Reaching the engine, the lubricating oil divides into high and low pressure circuits, as can be seen in Fig. 22(b), the pressures being about 250lb. per sq. in. and 43lb. per sq. in. respectively. The high-pressure oil passes through a telescopic pipe to the crosshead, the major quantity passing up into the piston head and, after cooling, returns via another telescopic pipe through a tundish and back to the drain tank. A minor quantity of oil is led off at the crosshead to lubricate the guide shoes and the top- and bottom-end bearings. It will be seen that the hollow connecting rod is fitted with an internal pipe, the oil supply to the bottom end passing down the pipe, while the oil drains from the crosshead flow through the annular space and passages in the bottom-end bearing to the sump. Returning the oil in this way greatly reduces the formation of oil mist and keeps the crankcase clean.

The low-pressure oil supplies the main bearings, thence by external pipes to the scavenge pump rocking lever bearings and to the crosshead guides, where it is used for cooling before returning via a tundish back to the sump and drain tank.

A rather unusual feature of the Sulzer engine is that high-pressure oil is used for cooling the fuel valves located centrally in the cylinder head. An oil feed is led direct from the main supply to the fuel valves and, after cooling, return pipes convey the oil back to the sump.

The main pistons and scavenge pump pistons are lubricated by mechanical lubricators which supply oil to suitably located feed points. Six or eight oil feeds are provided for the main pistons, situated about level with the bottom compression ring at top centre. Timed oil injection is used.

The open tundishes enable the oil flow from each piston and main bearing circuit to be checked and thermometers are provided for recording temperatures. It will be appreciated that although the piston cooling oil return temperature is accurate, the return from the main bearing has subsequently passed through the guides so is not indicative of the bearing temperature.

In this Sulzer engine, inspection openings are provided at the lower end of the cylinder through which part of the piston skirt can be observed with the engine running. The lower oil scraper rings are fitted in cast iron trays beneath the inspection openings. These trays collect oil scraped off the piston from which the dirty oil is drained to the bilge, thus preventing contamination of the crankcase oil.

Harland-B. and W. Engines: Two-cycle Opposed Piston Engine

A longitudinal view, in part section, of a six-cylinder Harland and Wolff opposed-piston single-acting two-cycle engine is shown in Fig. 23. The engine is similar to the Doxford engine in general principles but eccentrics are used to drive the upper exhaust pistons in place of cranks. In addition, mechanically driven rotary scavenging air blowers are used in place of scavenge pumps.

The lubricating oil drain tank is built into the double

Fuel valve cooling oil
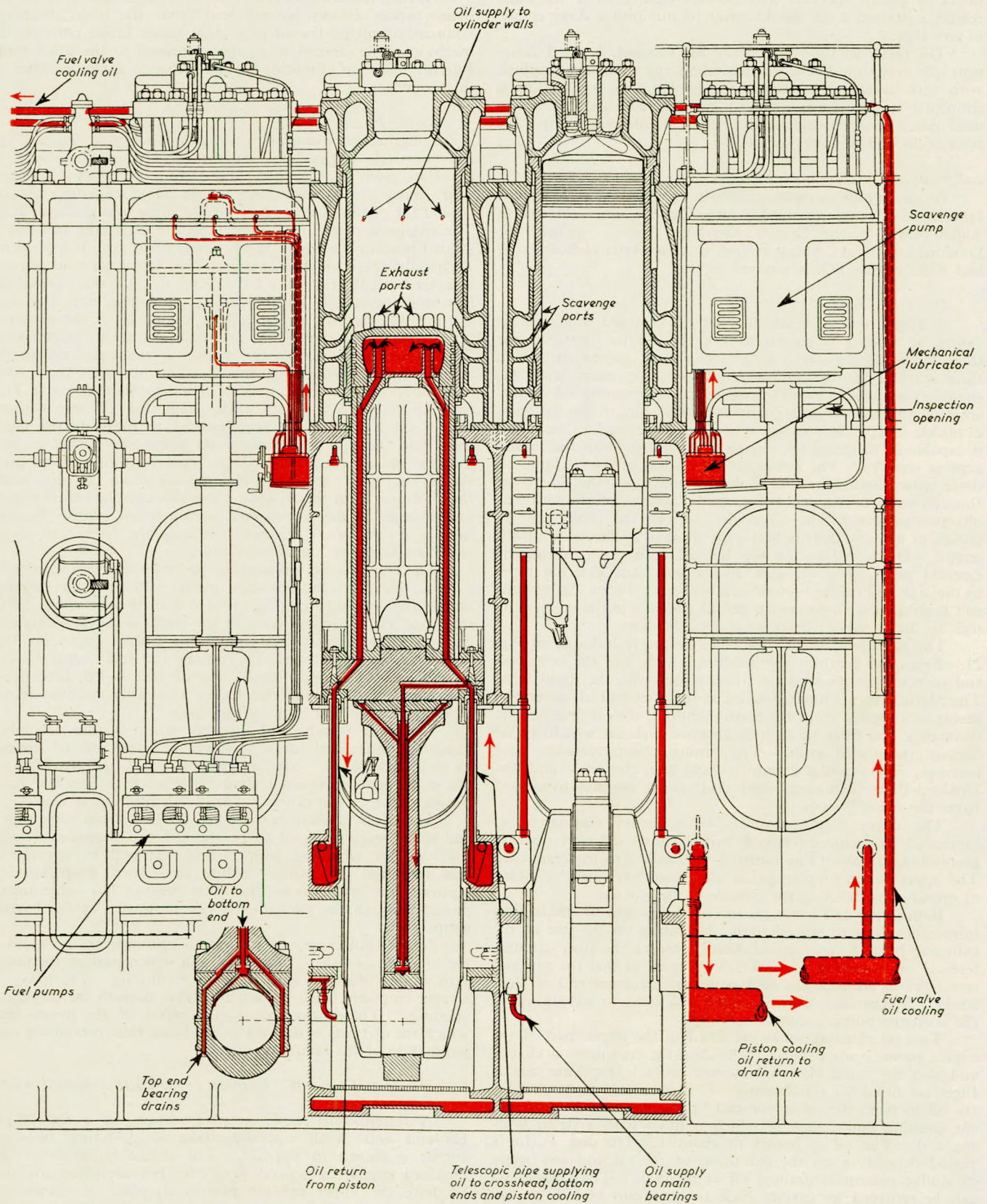
Oil supply to cylinder walls

Scavenge pump

Exhaust ports

Scavenge ports

Mechanical lubricator

Inspection opening

Oil to bottom end

Fuel pumps

Top end bearing drains

Fuel valve oil cooling

Piston cooling oil return to drain tank

Oil return from piston

Telescopic pipe supplying oil to crosshead, bottom ends and piston cooling

Oil supply to main bearings

FIG. 22(a)—*Longitudinal view of Sulzer Type SD two-cycle engine showing lubricating oil system*

Fuel valve coolant supplied from
high pressure oil manifold

Oil feed to scavenge
pump cylinder

Oil feed to
cylinder

Exhaust
manifold

High pressure lubricators for
pistons and scavenge pumps

Cooling oil return
from crosshead guides

Oil to
guide shoe

Tundish

Oil for guide cooling and
rocking lever bearings

Oil supply for
piston cooling

Outlet piston
cooling oil

High pressure oil

Low pressure oil

Supply manifold
for main bearings

Return to drain tank

FIG. 22(b)—*Transverse view of Sulzer Type SD engine*
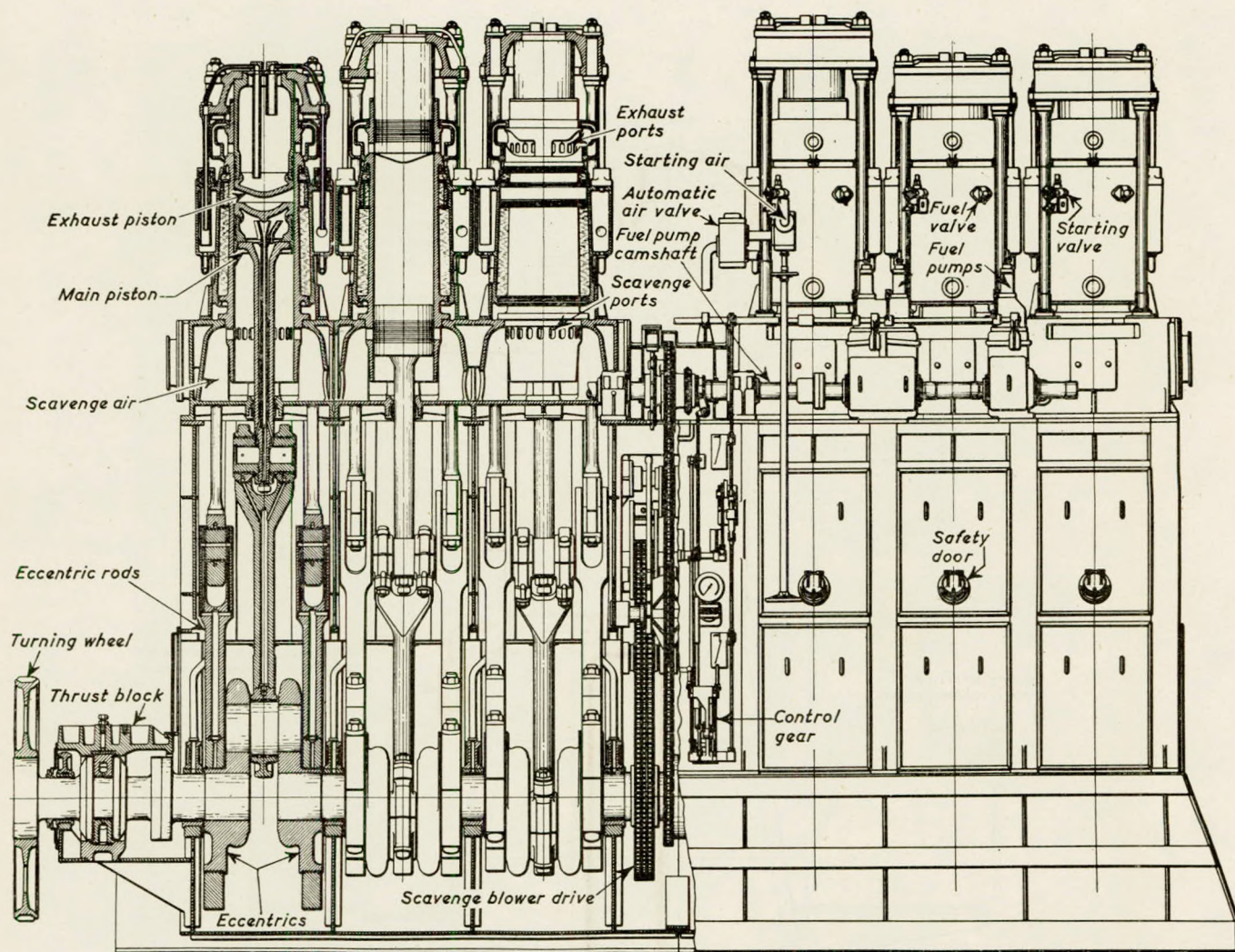
FIG. 23—*Harland and Wolff single-acting opposed piston two-cycle engine*

bottom beneath the engine. Duplicate lubricating oil pumps are provided, in some cases driven from the main engine in others independent, electrically-operated pumps are fitted. The usual suction and discharge filters and oil cooler are provided. Oil is pumped to a main distribution manifold from which it is fed to each main bearing by external pipes, then through drilled passages in the crankshaft to the bottom-end bearings. The oil then passes up the hollow connecting rod where it lubricates the top-end bearings before draining back into the sump. The eccentric straps and side crosshead bearings are lubricated in a similar manner.

Lubricating oil from the main supply system is used for piston cooling. Oil is conveyed to the main piston crosshead by telescopic pipe from which it passes up the hollow piston rod to the piston head and, after cooling, it returns via an internal pipe in the piston rod back to the crosshead and return telescopic pipe to the sump. A tapping is taken at the crosshead to supply oil to the ahead and astern guide shoes. The exhaust pistons are also oil cooled by separate telescopic pipes connected to the piston yoke.

Two scavenge blowers of the four-lobe rotor type are mounted on framework at the back of the engine, being chain driven from a sprocket wheel on the crankshaft. There is a clearance of about 20/1,000 inch between the vanes, the upper rotor being driven by spur gear from the lower rotor, which is directly coupled to the sprocket shaft. The rotor bearings and gears are lubricated from the main manifold, suitable

baffles and oil throwers preventing the oil leaking into the rotor casing.

A separate chain is used to drive the fuel pump camshaft, and where engine driven lubricating oil pumps are fitted, these are driven by a third chain from the crankshaft. Suitable guide sprockets and chain-tightening gear is fitted and all chains and bearings are lubricated from the main oil system.

Mechanical lubricators, chain driven from the fuel pump camshaft, supply oil to four feed points located at the outer ends of the cylinder. Timed injection is employed so that oil is injected when the pistons are towards the outer ends of their stroke, when cylinder pressures are lowest. Although it is easier to inject oil into the cylinder at these points it is questionable whether placing the feed points below the scavenge ports and above the exhaust ports is sound practice. As the pistons return to inner centre, much of the cylinder lubricant is swept into the ports instead of being spread evenly over the surface of the liner. If moderate oil feed rates are employed this could lead to dry spots on the liner. Furthermore, oil forced into the ports tends to carbonize and build up objectionable deposits.

To avoid the necessity of altering the lubricator adjustment with changes of oil viscosity due to varying engine room temperatures, a small thermostatically controlled, electric-heating element is fitted in the lubricator oil well. This maintains a constant oil temperature and, therefore, oil viscosity, which is particularly important in cold weather when starting

up, as it ensures correct injection timing and oil spread around the liner.

It is interesting to note that, unlike the two engines previously illustrated, in this case a short main piston is used, the lower end of the liner projecting into the enclosed scavenge air box. A piston rod scraper box, or gland, is fitted on the top of the crankcase, which prevents contaminated cylinder oil finding its way into the crankcase oil.

Four-cycle Single-acting Engine

Although the two-cycle engine is being increasingly used for all types of merchant vessels, there are still many adherents of the four-cycle engine. Fig. 24 shows the well-known Harland-B. and W. single-acting engine fitted with under-piston supercharging. This type of engine is simple, robust, and of proven design. The under-piston method of supercharging is extremely simple and effective, air pressures being usually about 3·5-4·75lb. per sq. in. The lower end of each cylinder is enclosed, the piston rod passing through a stuffing box in the top of the crankcase. The packing consists of cast iron scraper rings of the segmental type, held in place by garter springs. Two large diameter suction and delivery valves are fitted to each cylinder, the supercharged air feeding a manifold common to all cylinders.

The piston rod gland effectively prevents supercharged air or cylinder oil from entering the crankcase on the downward stroke of the piston, and also prevents oil vapour from the crankcase entering the cylinder on the upstroke (suction) of the piston.

A fuel oil circulating system is fitted with the necessary filters, pumps and cooler. Lubricating oil is supplied to a distribution manifold and thence to each main bearing in the usual manner. From the main bearing the oil passes through drilled passages in the shaft to the bottom end bearing, then up the connecting rod to the crosshead, where it lubricates top-end bearings and guides. Following normal Harland-B. and W. practice, the pistons are oil cooled by telescopic pipes, the oil being supplied from the main system.

Drain Tanks

The drain tank capacity in large marine Diesel engines which, of course, determines the amount of lubricating oil in the system, varies considerably from ship to ship. In some engines it is as little as about ½ gallon per b.h.p. while in others it is as much as 1¾ gallons per b.h.p., or, in round figures, for a 10,000 b.h.p. engine it can vary from 5,000 gallons to 17,500 gallons. In the latter case there is much less chance of the oil oxidizing rapidly, and if water leakage occurs, it
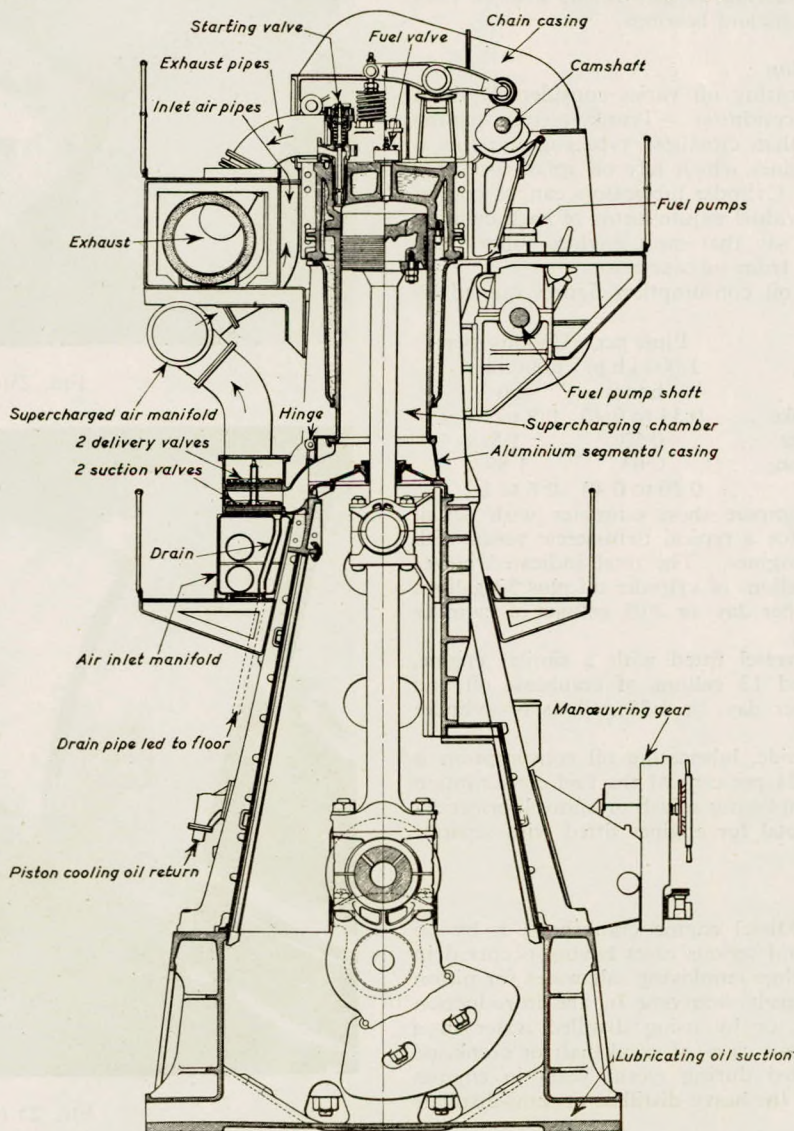

Fig. 24—*Harland-B. and W. single-acting pressure-charged four-cycle engine*

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has more chance to settle out before being recirculated through the engine.

Drain tanks in the double bottom should always be surrounded by a cofferdam, with the tank bottom at least twelve inches clear of the ship's hull to reduce the possibility of salt water leaking into the oil. Care should be taken to ensure that the drain tank top is watertight and all connexions kept completely tight to prevent the ingress of dirty bilge water and fuel oil into the tank.

Lubricating Oil Pumps

Independently-driven gear type pumps are now being widely used for bearing lubrication and piston cooling; in some cases a single pump carries out both duties, while in others high- and low-pressure pumps are installed. They have the advantage over engine driven pumps in that they can be started up before the main engine so that oil is thoroughly circulated before the engine starts and can be run after the engine is shut down, thus allowing working parts to cool down gradually. Furthermore, the oil pressure can be regulated independently of the engine speed.

Pump capacity, too, can vary considerably; for example, in an 8,000 b.h.p. engine the capacity ranges from about 90 tons per hour to more than 300 tons per hour. High capacity pumps are always to be preferred as they ensure a more than ample supply of oil to pistons and bearings.

Lubricating Oil Consumption

Consumption of lubricating oil varies considerably, both with engine design and condition. Trunk piston engines usually consume more oil than crosshead type engines; this is especially true of small engines which rely on splash or spray lubrication for the pistons. Cylinder lubricators can, of course, be adjusted to suit the individual requirements of each cylinder but it is generally true to say that most engines suffer more from over-lubrication than from oil starvation.

Approximate cylinder oil consumption figures quoted by Pounder[21] are:—

	Pints per 1,000 i.h.p. hour	Gallons per 1,000 i.h.p. day
Single-acting four-stroke ...	0·34 to 0·40	0·9 to 1·2
Single-acting two-stroke ...	0·50	1·5
Double-acting two-stroke ...	0·63	1·89
Trunk engines ...	0·20 to 0·40	0·6 to 1·2

It is interesting to compare these estimates with actual cylinder oil consumptions for a typical twin-screw vessel with opposed-piston two-cycle engines. The total indicated horsepower was 11,830 and 24 gallons of cylinder oil plus 32 gallons of crankcase oil were used per day, or 2·03 gallons of cylinder oil per 1,000 i.h.p. per day.

Another single-screw vessel fitted with a similar engine, developing 3,732 i.h.p. used 12 gallons of crankcase oil and 7 gallons of cylinder oil per day, or 1·88 gallons of cylinder oil per 1,000 i.h.p. per day.

As a rule-of-thumb guide, lubricating oil consumption is usually in the order of 1½-2½ per cent of the fuel consumption for trunk piston engines employing splash or spray lubrication, and about 2-3½ per cent total for engines fitted with separate cylinder lubricators.

Crankshaft Corrosion

Corrosion of marine Diesel engine crankshafts is by no means a new problem, several serious cases having occurred in the past, particularly in engines employing salt water for piston cooling. This has been largely overcome by the introduction of oil cooling for pistons, or by using distilled water in a closed circuit, and few serious cases of crankshaft or crankcase corrosion have been reported during recent years in engines using marine Diesel fuels of the heavy distillate or semi-distillate type.

During the last two years, however, several serious cases of severe crankshaft corrosion have occurred, and in all cases known to the author the vessel has been operating on residual fuel. To bring the matter into perspective it must be remembered that there must now be about 500 Diesel-engined vessels operating on heavy fuel, so that the proportion experiencing shaft corrosion is less than 5 per cent.

Where corrosion has occurred it has been most severe and rapid in action. In one case the shaft had to be removed and machined when the vessel was barely six months old. In another case it became necessary to machine the shaft after about a year in service. The corrosive attack does not appear to follow any fixed pattern, but is most severe on crankpins
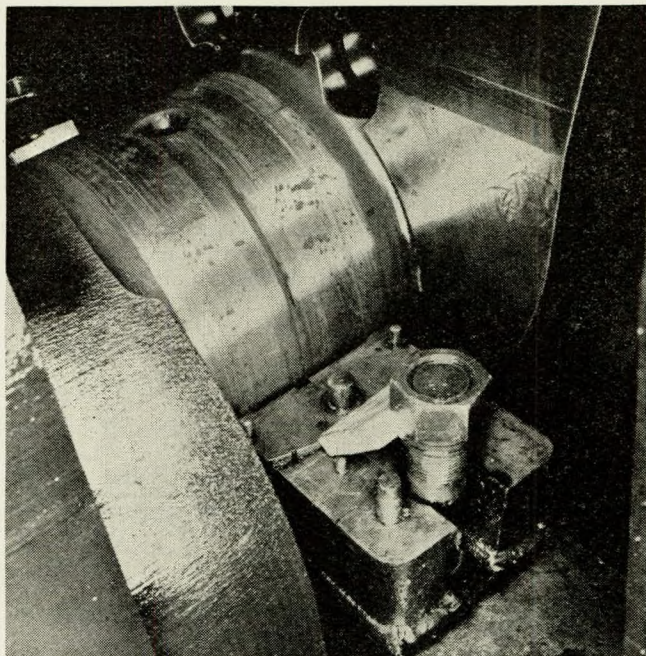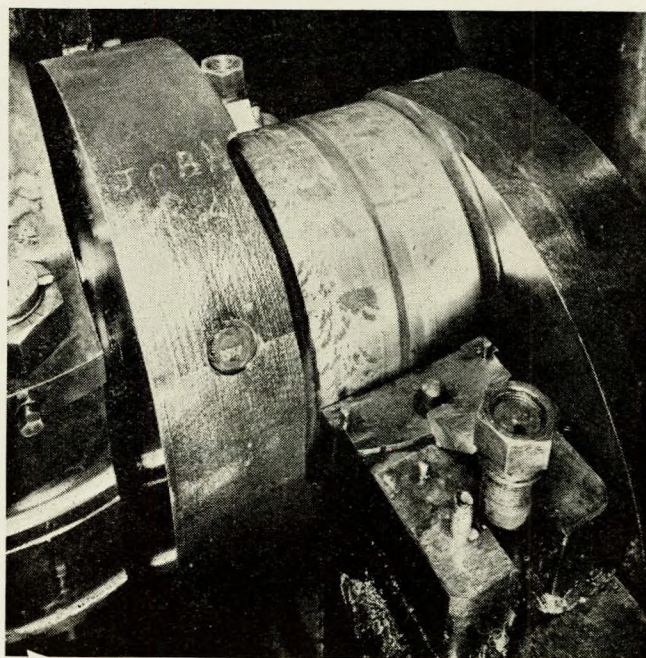

FIG. 25(*a*)

FIG. 25(*b*)

FIGS. 25(*a*) and (*b*)—*Typical examples of severe crankshaft corrosion in an engine operating on residual fuel*

known to the author the vessel has been operating on residual fuel. To bring the matter into perspective it must be remembered that there must now be about 500 Diesel-engined vessels operating on heavy fuel, so that the proportion experiencing shaft corrosion is less than 5 per cent.

Where corrosion has occurred it has been more severe and rapid in action. In one case the shaft had to be removed and machined when the vessel was barely six month' old. In another case it became necessary to machine the shaft after about a year in service. The corrosive attack does no appear to follow any fixed pattern, but is most severe on crankpins and main bearing journals, although in one case the crosshead pins were also severely attacked. In one engine examined the corrosion to crankpins and journals were severe in a band about an inch wide around the circumference of the journals and there were irregularly shaped patches at intervals on the rest of the journals. The depth of the corrosion was about $\frac{1}{16}$ inch maximum and about one-third of the bearing surface was affected. The corroded areas did not have the appearance of local pitting, which subsequently spread to cover several adjacent pit holes, but rather seemed to consist of fairly even area corrosion. Some of the affected areas were quite small but others were roughly two inches across.

Typical examples of the severity of the corrosive attack on main bearing journals are shown in Figs. 25(a) and (b). In this case the vessel was about eighteen months old when the photographs were taken and had operated on residual fuel since new, the average viscosity being about 1,500 seconds Redwood I at 100 deg. F. and an average of about 3 per cent sulphur.

In addition to crankshaft corrosion, several cases of severe corrosion of lubricating oil centrifuge discs and bowls have been reported; in the majority of cases the engines have been using residual fuel. In one instance the bowl and tinned discs of the centrifuge were severely attacked after the vessel had been in service for only a few weeks. The tinning on the discs below the holes had been almost entirely removed and the mild steel below also attacked. The bottom of the bowl was also severely attacked in a circumferential band immediately below the outer edge of the discs. In other cases the top covers have been severely corroded. It is perhaps significant that where corrosion has appeared in the centrifuge the engine crankshaft does not appear to have been affected.

In all cases so far reported there has been a considerable amount of water present in the lubricating oil, a fact of considerable importance when investigating causes of the corrosion.

Causes of Corrosion

The most apparent cause of corrosion to machined parts in the crankcase is that residual fuels have a much higher sulphur content than marine Diesel fuels previously widely used. It is practically impossible to exclude all combustion gases from the crankcase, particularly in engines where the bottom end of the liners is open to the crankcase. Blowby is common and SO_2 and SO_3 are usually found in the crankcase to a greater or lesser degree. As previously mentioned, it is likely that sulphuric acid is formed by condensation of SO_3 on the cylinder walls and this can also occur in the much cooler crankcase. Whichever the cause, it is quite possible for sulphuric acid to find its way into the crankcase oil, even if in very small quantities. There is also the possibility of smaller amounts of hydrochloric and nitric acids finding their way into the oil.

Condensation inside the crankcase can be appreciable, particularly in cold weather, as blowby gases contain a considerable amount of moisture, but this can be reduced by efficient crankcase ventilation. It is virtually impossible to entirely prevent leakage from the glands of telescopic pipes, or swinging arms, used to convey cooling fluid from fixed points of the crankcase to the moving crossheads and pistons. Where lubricating oil is used for piston cooling, leaking glands are of no importance, but where water cooling is employed,

leakage allows water to enter the crankcase oil. The water content of the lubricating oil has been found to be as high as 12 per cent in some engines, but luckily such cases are rare. Unless leakage is excessive, it can be kept within reasonable limits, provided that an efficient centrifuge is installed and used regularly, and that it is working satisfactorily.

Salt water occasionally finds its way into the crankcase oil by direct leakage into the double bottom drain tank, through leaking oil coolers, by leakage past liner lower sealing rings, or from cracked liners or cylinder covers where salt water cooling is employed. Fortunately, such leakages are not common occurrences, particularly now that salt water cooling of pistons is virtually obsolete. The possibility of leakages of this kind must, however, always be guarded against.

A further factor which may influence corrosion is that most of the large engines still employing distilled or fresh water cooling use potassium dichromate as a corrosion inhibitor in the water system, the standard solution used containing 2lb. potassium dichromate per 1,000 gallons of water. There is a possibility that potassium dichromate solution which has found its way into the lubricating oil from leaking cooling water glands will be present in the oil mist in the crankcase and will come in contact with SO_2 and SO_3 gases, accelerating the formation of sulphuric acid. It is also possible that while potassium dichromate in correct concentration will act as an anodic corrosion inhibitor, when the solution becomes weakened by admixture with water present from other sources in the oil it will become an active cathodic corrosion accelerator and localized pitting will occur. It is known that where potassium dichromate solution has mixed with salt water in crankcase oil, severe corrosion has occurred.

The possibility of galvanic or bi-metallic corrosion cannot be ignored. Practically all marine Diesel engines employ tin base or lead base white metalled bearings, working on mild steel journals or cast iron guide slabs. An electrolyte is always present as the oil is never moisture-free and is acidic to a greater or lesser degree. Fortunately these metals are fairly close together in the electro-chemical series and the difference in potential is not great. It is unlikely, therefore, that bi-metallic corrosion plays any major part in crankshaft corrosion.

The possibility of stray electrical currents in a ship's hull must be taken into consideration as these can promote galvanic corrosion. If galvanic action is one of the causes of crankcase corrosion it would be expected that the most severe attack would occur where the current is most likely to be earthed, i.e. where the oil film is thinnest. Assuming that the current is travelling along the main shafting, the first place it is likely to be earthed is at the intermediate bearings. As, however, the load on these bearings is usually light, due to the shaft only transmitting torque, the oil film generally gives sufficient insulation. The next most likely earthing point is the crankshaft main bearings and following that the bottom-end bearings. Particularly at the crankpins, the oil film is thin and it is on crankpins and main journals that attack is most severe.

It has been proved that differences of potential can exist between crankshaft and bearings and in some earlier cases of shaft corrosion this was stopped by fitting earthing brushes to the shaft. It has yet to be proved, however, that the present cases of corrosion are due to electrolytic action, but this possibility certainly warrants close investigation.

As all the possible causes of corrosion mentioned could be experienced in motorships operating on either marine Diesel fuel or residual fuel, it is difficult to see why serious corrosion should be limited to a handful of ships, unless a combination of causes prevail in these instances. It is obvious that further investigation must be made into this problem, as once the cause (or causes) is established it may not be too difficult to find the antidote. Even with our present knowledge, however, certain precautions can be taken to reduce the possibility or eliminate some of the potential causes. In the author's opinion the major cause is corrosive attack by mineral acids and if these can be eliminated it is a step in the right direction.

Possible Methods of Preventing Corrosion

As water has always been present in the oil, usually in appreciable amounts, where severe corrosion has occurred, the first step is to do everything possible to keep the water content to a minimum. The simplest, and probably the most effective way of doing this, is to fit an efficient centrifuge and operate it regularly and efficiently.

In engines fitted with a division plate between the lower end of the cylinder and the crankcase, the possibility of products of combustion entering the sump are greatly reduced provided the piston rod gland is maintained in good condition. Oil drainings from the cylinder should never be added to the crankcase oil.

Where pistons are water cooled, particular attention should be paid to keeping all glands in the crankcase as leak free as possible. Care should be taken to ensure that the water system is drained before breaking water joints during overhauls, otherwise a considerable amount of water can leak into the oil.

It is possible that improved types of additives may be developed which could neutralize mineral acidity but the amounts required might greatly increase the cost of the lubricants. If, however, increased corrosive wear of liners is linked with corrosion in crankcases, the increased costs would be well worth while.

If it can be proved that the presence of potassium dichromate is a major factor in crankshaft corrosion, then alternative cooling-system corrosion inhibitors will have to be found. Soluble oils, of the cutting oil type, have already been tried for this purpose but results were unsatisfactory. Corrosion occurred in the water spaces, particularly in the piston heads, and in at least one case natural rubber-base packing and rubber flexible piping were attacked and the softened rubber choked the cooling water pipes. The answer to the latter trouble is simple—fit synthetic rubber packing and piping. It is likely, however, that some conventional soluble oils do not provide the corrosion resistance required, but they can vary widely in this respect.

Diesel Crankcase Oil Maintenance

It is apparent that the maintenance of Diesel lubricating oil is an extremely important matter and this is especially true in vessels burning residual fuel. Practically all modern motorships are fitted with centrifuges, associated oil heaters and, usually, settling tanks. It is a matter of some importance that all ships' engineer officers should be fully acquainted with the oil cleaning equipment and how and when it should be used.

It is difficult to lay down hard-and-fast rules regarding the amount the centrifuge should be used, as this varies considerably with the condition of the oil, the amount of blowby, the type of engine and several other factors. For instance, if water, from whatever source, is known to be leaking into the oil in appreciable quantity, the centrifuge should be operated continuously until the origin of the leak can be traced and stopped.

As the purpose of the centrifuge is to remove all abrasive matter and water from the oil as soon as possible after they enter the system, the machine should be operated for at least four hours per day. An examination of the water discharge from time to time will soon indicate whether or not there is any serious water leakage, and also whether it is fresh or salt.

The question of whether or not hot water washing or steam washing should be applied when centrifuging is a rather difficult one. Where the oil is known to be in good condition and reasonably dry, there appears little point in deliberately adding water. Again, if the centrifuge is not working properly, and cases have been reported where the water outlets of the machine were completely choked, a considerable amount of water can return with the oil to the drain tank, thus creating conditions we are trying to avoid. As, however, the crankcase oil, after it has been in service for even a short period, will be oxidized to some degree and it is agreed that there is always a possibility of mineral acids being present, water washing

has the beneficial effect of dissolving water-soluble acidic compounds and these will be removed with the water discharge, leaving the oil much cleaner than it would be with dry centrifuging. With some of the binders removed, solids will also separate out more readily.

Where water washing is employed it is important that clean fresh water is used. Treated water can be undesirable, for instance, heavily chlorinated water might well cause corrosion in the centrifuge. The water temperature should be kept as high as convenient, 160-180 deg. F. being quite suitable. Tests of the discharged water should be carried out at intervals as these will indicate whether or not it is acidic. If tests cannot be conveniently carried out on board ship, periodic samples should be sent to the oil supplier, particularly in cases where corrosion is known or suspected. The rate of water wash should not exceed about 5 per cent of the centrifuge oil throughput.

If an oil is known to be in a badly oxidized condition, and it has not previously been water washed, care must be taken when it is first applied. In extreme cases stable emulsions could be formed which could not be broken down in the separator and would be carried through to the drain tank. In such cases the entire oil charge should be removed as soon as possible.

The question of whether steam should be used in place of hot water is a rather controversial one. Some engineers have used it for several years and claim that they are able to get the oil very much cleaner than when employing water washing. On the other hand, complaints have been made that the emulsions formed using steam have been so stable that the centrifuge could not break them. It has also been suggested that steam washing is liable to "crack" the oil.

It is considered that blowing a jet of high velocity steam into the oil will be much more effective as a washing agent than hot water, due to the very thorough mixing of steam and oil and the higher temperature involved. The emulsions formed will, however, be more stable, rendering the task of the centrifuge more difficult. In many cases the steam has been injected too near to the centrifuge, allowing no time for settling. It might be worth while experimenting with a closed mixing chamber holding two or three gallons, located about six feet from the centrifuge. The steam jet could be located in the top of the chamber and the washing could be thoroughly carried out, while the oil leaving on its way to the centrifuge should have settled down.

There is little danger of cracking the oil with the steam available on motorships. The temperature rarely exceeds 400 deg. F. at the boiler and, of course, low pressure steam should always be used for steam washing. Unless a very poor quality oil is in use, cracking will not occur until temperatures are well above 600 deg. F.

One of the strongest objections to either steam or water washing is that several tons of valuable fresh water may be used and not recovered every day, an important item, particularly on long runs. The cost of using steam is, of course, much higher than for an equivalent weight of water. As a compromise, dry centrifuging could be carried out at sea, and batch filtration carried out in port. The most effective method of batch filtration is to pump the oil into the dirty oil settling tank and allow it to settle for 12-24 hours with the steam coils in use. Free water and solids would settle out and should be drained off at intervals, and the oil then passed through the centrifuge heater to raise the temperature to about 150-160 deg. F. The oil should then be passed through the centrifuge, using a steam or water wash and after cleaning returned to the engine drain tank.

As previously mentioned, even if it takes some time to establish the causes of crankshaft corrosion, if we can entirely remove water from the oil, it will act as an insulator instead of an electrolyte and the possibility of corrosion will be greatly reduced. A single centrifuge, although efficient by normal standards, cannot entirely remove all moisture. With this in

mind one shipowner has installed a second duplicate centrifuge working in series with the first. In this case it has been proved that oil leaving the first centrifuge contains traces of sulphuric acid but that the oil leaving the second machine is completely free of mineral acids and free water. This is believed to be a major step forward in the prevention of corrosion and further tests will be closely followed. In this case it is worth noting that the normal mineral acidity as determined by the standard Institute of Petroleum test IP.(1/46) was nil. It is obvious that during research into this problem of corrosion more sensitive methods of determining mineral acid contents will have to be used.

Maintenance of Detergent Oils

As detergent oils will undoubtedly be used more and more in marine Diesel engines in the future, particularly for small high- and medium-speed engines, it may not be out of place to devote a few words to the question of their maintenance. An essential difference between straight and detergent oils is that the latter type will rapidly become black and opaque, in fact if they do not they are not doing their job. This fact is not always appreciated and some engine operators do their best to make the oil clear and bright again. If they succeed in doing this, it means that not only have they removed the carbon which has darkened the oil, but they will also have removed some of the detergent additive intimately associated with it and thereby reduced the oil's effectiveness. A filter which removes carbon and thereby additive is not suitable for use with this type of oil.

It is, of course, just as important to filter effectively detergent oils as to filter straight oils; that is, all harmful abrasive matter and water should be removed. Suspended carbon particles in a detergent oil seldom exceed 1 or 2 microns in size (1 micron = approximately 0·00004 inch) and as it is generally accepted that particles below 5 microns cannot cause abrasive wear, there is little point in fitting filters which give superfine filtration. None of the normal full flow filters will cause any difficulty with removal of additive but certain very fine bypass strainers using edge type paper packs, or cartridges containing activated alumina and similar materials, are known to remove carbon plus additive. The majority of fine filters, however, used essentially with auxiliary engines, are quite suitable for use with detergent oils.

Efficient centrifuges can filter down to about 5-micron particle size and as there is no chemical or physical adsorption on the discs or bowl, they are perfectly suitable for use with detergent oils Centrifuges have the advantage over static filters in that they can also effectively remove comparatively large amounts of water.

Detergent additives vary considerably in their water solubility and also in demulsibility properties. It is not advisable, therefore, to use a water wash with detergent oils unless the oil supplier agrees that this practice is acceptable. This can, of course be advanced as an argument against the use of detergent oils in the crankcases of engines operating on heavy fuel, where water washing the oil is standard practice as a safeguard against corrosion. This does not, of course, preclude their use in auxiliary Diesel engines.

Laboratory Tests on Used Diesel Oils

In the early part of this paper reference was made to the more common tests carried out on lubricating oils, some of which are used for evaluating the condition of used Diesel engine crankcase oils, and in addition special tests are also frequently carried out. It is rather difficult to be precise regarding how often samples should be drawn for testing as there are so many variables concerned. There is little point in taking samples too frequently as this just involves the oil company's chemist in unnecessary work, but as a rough guide a sample taken every 1,000 hours' running is usually sufficient. Where special investigations are being carried out, however, samples should be drawn more frequently and from several different points.

Routine test samples should be drawn off at some suitable point in the engine where the oil is circulating. Where they are drawn from a drain cock or valve it is advisable to draw off about a gallon of oil before filling the clean sample bottle or can. If this is not done, it is possible that sludge or solids, trapped in the drain pipe and cock, will be flushed out into the sample container and will give a completely false picture of the condition of the bulk of the oil in the system. A small, but important point, is to see that all sample containers are clearly marked with the name of the ship, type of engine, point from which sample is drawn, grade of oil concerned, hours in service and any comments which will be of assistance to the examining chemist. It is surprising how often an oil sample arrives at the laboratories in containers such as sauce bottles and with no indications from whence the sample has come!

Some superintendent engineers prefer two samples to be tested at intervals, the first a sample at the purifier inlet which can be taken as typical of the oil in the system, and the second from the purifier outlet. The difference between the two gives a check on whether or not the centrifuge is operating correctly.

Routine tests should comprise the following:—

Redwood No. 1 viscosity at 140 deg. F. (or kinematic viscosity at 100 deg. F.).

Closed flash point (P.M.).

Neutralization value, mgms. KOH/gm.

Water content, per cent.

Diluent content, per cent, for engines operating on marine Diesel fuel).

Insolubles (I.P. spirit), per cent.

The significance of viscosity, neutralization value, flash point and water content have already been covered earlier in this paper. Crankcase oil dilution by fuel is important, as even a relatively small amount of marine Diesel fuel dilution will
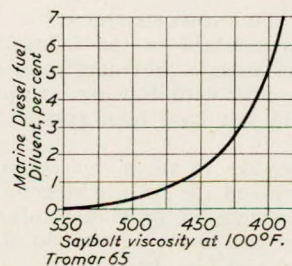

FIG. 26—*Effect of fuel dilution on viscosity of typical crankcase oil*

greatly reduce the viscosity of the lubricant. This is not, of course, true where residual fuels are concerned, as a 1,500 seconds fuel is as viscous as a heavy crankcase oil. In addition to reducing the viscosity, the flash point is also considerably lowered by fuel dilution.

Fig. 26 illustrates the effect of fuel dilution on an S.A.E. 30 crankcase oil. In this case all viscosities were taken in Saybolt Universal Seconds at 100 deg. F. on an unused oil, predetermined percentages of fuel being mixed with the oil
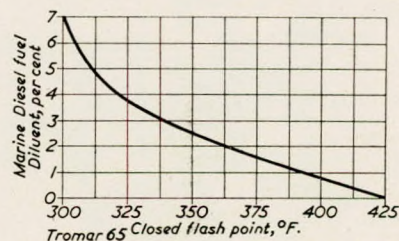

FIG. 27—*Effect of fuel dilution on flash point*

in the laboratory and the viscosity plotted against percentage diluent. It will be seen that the greatest relative decrease in viscosity occurs between 0 and 1 per cent diluent after which the curve gradually flattens out. It is considered that a graph of this kind plotted from laboratory results is more accurate than carrying out an actual dilution test, as the test is unreliable below about 5 per cent dilution. The effect of fuel dilution on closed flash point of the same oil is shown in Fig. 27. Here again the results were plotted from accurately controlled laboratory tests. Although different results can be obtained with other lubricants the general pattern is very similar.

The insolubles content of a used lubricating oil should be known as it gives the amount of solids in suspension in the oil. The solids may be carbonaceous or foreign matter but both are undesirable in straight oils. With detergent oils, of course, the position is different as the oil is designed to carry finely-divided carbon in suspension but there is a limit to the amount of insoluble material which can be carried and the test provides valuable information as to whether the oil is approaching the end of its useful life.

Where crankshaft corrosion is known, or suspected, samples of water from the purifier discharge should be taken. These samples can provide valuable information as to the likely cause of the trouble.

With detergent oils, the sulphated ash content is frequently taken, as it gives a guide to the amount of additive remaining in the oil. The value of the test is debatable, however, as it has not yet been found possible to differentiate between exhausted additive and active additive. Although little is known as to how detergent additives work, it is generally assumed that the fine carbon deposits are coated with a layer or envelope of additive which thereby prevents coagulation. In time, therefore, the additive gradually becomes exhausted even though it is being frequently replenished in new make-up oil.

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INSTITUTE ACTIVITIES

Autumn Golf Meeting

On a day of brilliant sunshine and high winds twenty-two players took part in the Autumn Golf Meeting at Hadley Wood Golf Club on Wednesday, 29th September 1954.

The morning Medal Competition was won by Mr. J. H. F. Edmiston with a net score of 68 and he received a brief case. Mr. R. M. Wallace was second with a score of 76.

In the Bogey Greensome Competition in the afternoon, Messrs. R. M. Wallace and R. B. Pinkney took first place with 4 down and received sets of onyx ash trays. Messrs. W.

Mr. J. H. F. Edmiston

Donaldson and H. P. Jones were second with a score of 5 down; they received silver ash trays.

As no member could take more than one prize, Mr. Wallace elected to forego the morning prize, so Mr. A. Bartholomew, with a score of 77, received a walnut cigarette box.

Mr. Alfred Robertson, Convener of the Social Events Committee, presented the prizes during the tea interval. He then proposed a vote of thanks to the committee of Hadley Wood Golf Club for the excellent arrangements which had been made for the day and to the catering staff for the enjoyable lunch and tea. This was carried unanimously. Thanks were also expressed to Messrs. A. Bartholomew, J. G. Edmiston, S. J. Jones, J. C. Lowrie, J. M. Mees, R. B. Pinkney, A. Robertson, R. M. Wallace and Commander J. White for providing the prizes.

Mr. Robertson, on behalf of all the members present, said how pleased he was to see Mr. J. Rhynas after his recent illness and hoped it would not be long before his recovery was complete. A telegram was received from Messrs. R. K. Craig and W. J. Borrowman in Switzerland expressing their best wishes for a successful day.

The meeting terminated with a vote of thanks to the Convener and Committee of Social Events.

Local Sections

Merseyside and North Western

Mr. G. Kenworthy-Neale, A.M.I.Mar.E., has taken over the duties of Honorary Secretary for the Merseyside and North Western Section, and communications should be addressed to him at Riversdale Technical College, Riversdale Road, Aigburth, Liverpool, 19.

Scottish Section

The first General Meeting of the session was held in the

Mr. R. M. Wallace

premises of the Institution of Engineers and Shipbuilders in Glasgow on 22nd September 1954.

A lecture on "The Condensing Gas Turbine" was given by J. F. Field, Controller of the S.E. Scotland Division of the British Electricity Authority, and thoroughly enjoyed by the 125 members and guests present.

This was followed by a pleasant and interesting discussion and, on the proposal of Mr. Dunshea, the speaker was accorded a hearty vote of thanks.

Sydney Section

A meeting of the Sydney Section was held at the Shell Theatrette, 2, Carrington Street, Sydney, on Thursday, 30th September 1954, at 8 p.m. Seventy-seven members and guests were present and Mr. W. G. C. Butcher was in the Chair. A lecture on "The Fundamentals of Petroleum Refining" was given by Mr. W. R. Evans and then two films were shown, "Prospecting for Petroleum" and "Petroleum Refining". Messrs. J. A. Pollock, H. W. Lees and D. N. Findlay took part in the discussion that followed.

A vote of thanks to the author was proposed by Mr. C. McLachlan and carried by acclamation.

West Midlands

A General Meeting of the West Midlands Section was held at the Imperial Hotel, Birmingham, at 7.0 p.m. on Thursday, 14th October 1954. Mr. H. E. Upton (Chairman) was in the Chair and over one hundred members and visitors were present.

Commander(E) G. F. A. Trewby, R.N. (Member) presented his paper entitled "British Naval Gas Turbines". The paper gave a comprehensive and factual report of the various gas turbine projects undertaken by engine builders on behalf of the Admiralty. It commenced with a report of the small main propulsion units installed in small naval craft, and continued by giving a detailed description of the more ambitious units which were built by Midlands firms. The paper concluded with a description of gas turbines designed for auxiliary services.

Eleven members took part in the ensuing discussion. The Chairman thanked Commander Trewby for his extremely interesting paper, and the meeting closed at 8.45 p.m.

Membership Elections

Elected 4th October 1954

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James Grant Holburn
Owen Henry Hughes
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Thomas Bertie Robinson, Lieut.-Cdr.(E), C.D., R.C.N.
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Martin Robert Ieuan Evans
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Ian Tyreman
Roger Giles Webb
Barry Edward Wilson
Keith Thomas Wooler
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TRANSFER FROM ASSOCIATE TO MEMBER
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Thomas Chapman
James Cowley
Colin Stewart Reed
Bertram Thomas Revans
William Gerald Rhodes
Peter Brace Thomas
Frans Gerard van Asperen

TRANSFER FROM ASSOCIATE TO ASSOCIATE MEMBER
Gordon Lindsay Cunningham
Thomas Jefferson Davis
Alan Moore

TRANSFER FROM GRADUATE TO MEMBER
William John Prowse, Lieut.-Cdr.(E), R.N.

TRANSFER FROM GRADUATE TO ASSOCIATE MEMBER
Richard Brown

TRANSFER FROM GRADUATE TO ASSOCIATE
Wallace James Ayers
Anthony Andrew Pollock

TRANSFER FROM STUDENT TO ASSOCIATE MEMBER
Roger James Gates, B.A.(Cantab.)

TRANSFER FROM STUDENT TO GRADUATE
Neil Welbourn Jephcott, B.A.(Cantab.)

OBITUARY

JOSEPH CHARLES BEAUCHAMP (Associate Member 6842) was born in 1885. He was apprenticed to Armadale Motors, Ltd., from 1910-19 he was first assistant and then manager of Wolseley Motors, Ltd., and from 1919-30 manager of Jones, Burton and Co., Ltd. In 1930 he became manager for Gaston E. Marbaix, Ltd., until in 1942 he transferred to Industrial Exports, Ltd., returning to the former company in 1944 for a further three years as manager. From 1947-50 he was manager of engineering exports for the Rom River Co., Ltd., when he became seriously ill with a complaint from which he died on 30th August 1954.

Mr. Beauchamp had been a member of the Institute since 1930.

PETER ERNEST BEGG (Associate 11693) was born in 1925. He served a five-year apprenticeship with Grayson, Rollo and Clover Docks, Ltd., Liverpool, and before joining the Anglo-Saxon Petroleum Co., Ltd., in 1946 he had been awarded a Higher National Certificate. He obtained a First Class Ministry of Transport Motor Certificate in 1950, a First Class Steam Endorsement in June 1951 and an Extra First Class Certificate in November 1951. In 1953 he joined the staff of the Admiralty Research Department of Yarrow and Co., Ltd., as a technical engineer and continued in this appointment until he failed to return from a climbing expedition on the mountains near Glencoe in the Western Highlands of Scotland on 4th September 1954. Mr. Begg was an experienced mountaineer, a keen motorist, and a powerful swimmer. He applied himself with the utmost zeal and energy to all his undertakings and his loss is much regretted by his colleagues as well as by the management of the company.

He had been an Associate of the Institute since 1948.

JOHN THOMAS BUSH (Associate Member 5733) was born in 1893. He served an apprenticeship with the Western Electric Co., Ltd., from 1909-11, after attending the L.C.C. School of Engineering and Navigation, Poplar, as one of their first pupils from 1907-09. He was technical assistant to Edward Wans and Co., of Chancery Lane, from 1911-14, and throughout the first World War, from 1914-19, he was a staff sergeant mechanist in the Royal Engineers. In 1919 he joined W. and L. Cole, Ltd., London, E.3, with whom the rest of his professional life was to be spent, as chief draughtsman and designer, and at the time of his death on 16th May 1954 he was one of their directors.

Mr. Bush continued his connexion with the Poplar Technical College throughout his life, being chairman of the governors when he died; he was also vice-chairman of the managers of a group of Poplar schools.

He was an Associate Member of the Institution of Mechanical Engineers and had been a member of the Institute since 1927.

ALAN LESLIE GATLAND (Member 10509) was born in Auckland, New Zealand, in 1886. He served an apprenticeship with the Clyde Iron Works, Onehunga, from 1901-06, and then spent a year in the United Kingdom as a fitter with Denny and Company, Dumbarton. He sailed as third and second engineer with Bell Brothers and Company, of Glasgow, obtaining his First Class Board of Trade Certificate in 1910. In 1912 he joined the Union Steam Ship Company of New Zealand, Ltd., as a junior engineer in the *Niagara*, then under

construction at Clydebank. He was third engineer of this vessel in 1915 when he proceeded to the United Kingdom to take a similar position in the *Aotearoa*, which was being built at the Fairfield Shipbuilding and Engineering Company, Govan. This vessel was taken over by the Admiralty and commissioned as H.M.S. *Avenger*, Mr. Gatland serving as Lieutenant(E) until the ship was torpedoed in 1917. He was transferred to the Royal Navy until 1919 when he rejoined the Union Company as second engineer in the *Maunganui*. He served later in the *Tahiti* and *Makura* to supervise the conversion to oil fuel burning of these vessels and left the latter to join the m.v. *Aorangi* in 1923, building at Fairfields. He remained in this ship as second and chief engineer until 1930 when he went to Barrow in Burness as chief engineer to supervise the construction at Vickers-Armstrongs, Ltd., of t.e.v. *Rangatira*.

In 1934 he was appointed assistant superintending engineer at the head office in Wellington and succeeded in April 1946 to the senior position, which he held until his retirement in 1952. During this period he revisited the United Kingdom on two occasions in connexion with post-war new tonnage and again after his retirement when he undertook supervisory duties in the United Kingdom, culminating in the delivery of the t.e.v. *Maori* in October 1953.

Mr. Gatland retired finally in December 1953 but he did not enjoy good health; after a long illness he died in Auckland Hospital on 4th October 1954.

He was a member of the New Zealand Institute of Marine and Power Engineers and had been a Member of the Institute since 1945.

THOMAS JAMIESON MORGAN (Member 2965) served an apprenticeship with N. G. Abercrombie, Broad Street Engine Works, Alloa. He was a seagoing engineer for about twenty years and obtained a First Class Board of Trade Certificate. In 1921 he was appointed to the staff of the Norwich Union Fire Insurance Society as an engineer surveyor and served in that capacity until 1928, when he was transferred to the staff of Insurance Engineers, Ltd., as district superintendent, Glasgow District Office, a position he held until he retired in December 1940. Mr. Morgan died in his seventy-ninth year on 20th August 1954 after a long illness.

He had been a Member of the Institute since 1915.

DAVID TAYLOR (Member 4406) was born at Ayr in 1886. He served an apprenticeship from 1900-05 with J. and A. Taylor, Ltd., Ayr, then went to sea in ships of the Ayr Steam Shipping Company, now the Burns Laird Line. He obtained a First Class Board of Trade Steam Certificate in 1910 and sailed in various foreign-going ships thereafter. He joined the Royal Navy in the first World War, was wounded in action at Zeebrugge and attained the rank of engineer lieutenant commander. In 1921 he was appointed engineering manager of Forster's Dry Dock, Ltd., and later he was representative in Glasgow for J. and E. Hall, Ltd. He rejoined the Royal Navy on the outbreak of the second World War, being stationed as engineer officer at Scapa Flow in the Orkney Islands, with the rank of commander. Owing to poor health, however, he had to resign before the end of the war. From 1945 until he died on 6th September 1954 Mr. Taylor was assistant manager for Heenan and Froude, and Fielding and Platt, at their Glasgow offices.

Mr. Taylor was elected a Member in 1921.