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Engine Oil Filtration and its Effects on Wear in Internal-Combustion Engines.

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CHAIRMAN: J. CALDERWOOD, M.Sc., (Chairman of Council).

Synopsis.

This paper relates to a subject regarding which there are many controversial views, and the author has been at some pains to produce a logical discussion which may not only be of current interest but may lead to a better conception of the effects of oil filtration on engine wear.

The author demonstrates the effects of dynamic loading on a lubricated bearing. With a clear understanding of these effects it will be possible to understand how and when the wear of bearing surfaces occurs. The possibilities and practical limitations of engine oil filtration and the reasons why an inferior or unsuitable filter may be of no advantage are considered. It will be shown, too, that an inferior filter may actually cause an increase in the rate of wear.

Effects of Dynamic Loading on Lubricated Bearings.

We will first consider the action of a lubricated bearing when subjected to a dynamic loading. The action of a bearing under a constant load has been thoroughly investigated and theoretical formulae relating thereto have been well established by experiment. The effects of dynamic loading do not appear to have been so well investigated. These effects, however, have an important influence on wear in engines, especially when the lubricant is contaminated with abrasive particles.

Under a constant load the eccentricity of a shaft in its bearing may readily be determined in terms of the parameter ZN/P. Let us, however, determine the eccentricity of a shaft under constant loading in terms of the intensity of pressure P while the product ZN, of the viscosity and speed, is kept constant. We then obtain

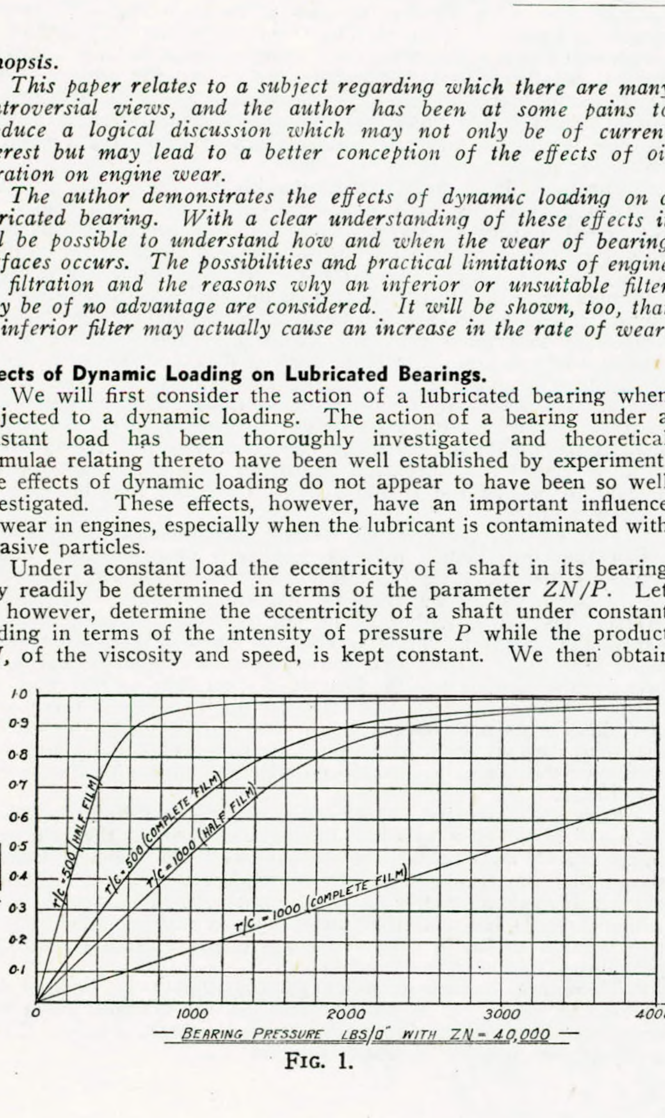
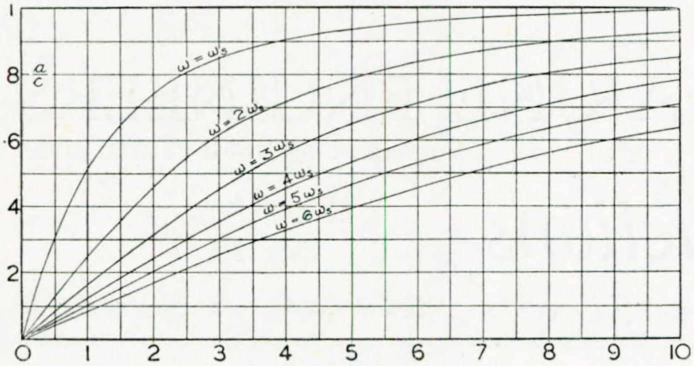


FIG. 1.

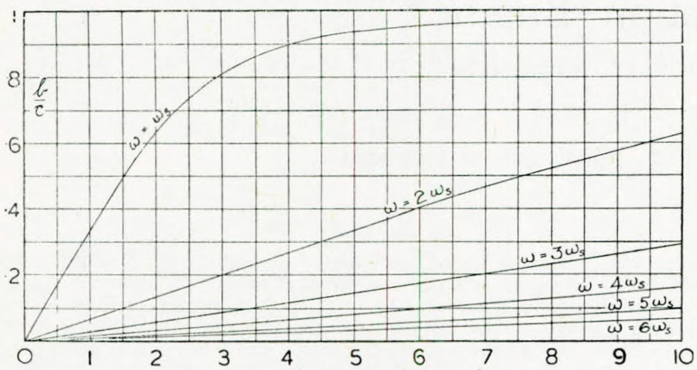
curves as shown in Fig. 1, in which the symbols r and c represent the radius of the journal and the radial clearance respectively. From these curves it will at once be noticed that the eccentricity of the journal is proportional to the load thereon for eccentricities up to more than half of the radial clearance. Thus, for small deflections of a rotating journal, the oil film reacts like a spring possessing a constant elasticity. It must be remembered, however, that the deflection of the journal is not in the same direction as the load. Nevertheless, the motion of a journal relative to its bearing can be regarded as that of a body provided with an elastic restraint and damping, and acted upon by a variable force. A periodic irregular force, such as that acting on a crankshaft bearing for example, can be represented by a series of harmonic components, and the response to the irregular force can be obtained by superposition of the responses to these harmonic components. Let us consider one such harmonic component, that is to say, a harmonic periodic force applied to a lubricated journal. If the periodic force has a small amplitude, the locus of the centre of the journal is an ellipse given by the equation $x^2(p^2-1)^2+y^2(p-1/p)^2=(F_o/k)^2$. Transient motion, if any, will rapidly subside. p is equal to $\beta\omega/k$ plus a negligible mass inertia function, ω being the harmonic frequency, β the viscous damping factor and k the elastic restraint factor. The methods of determining this and other formulae which have been devised for the purpose of this paper are included in a mathematical analysis presented in the Appendix to this paper.

The above equation is only true for small oscillations because both the damping and elastic restraint factors vary with the deflection. However, we can replace k by k_a, the effective average value of k up to the amplitude "a" of the deflection, where $k_a=\phi(a).k_o$, $\phi(a)$ being a function of "a" and k_o being the value of k for very small deflections. For a given bearing k_o is equal to a constant multiplied by $Z\omega_s$ where ω_s is the angular velocity of the shaft and Z is the viscosity of the oil. The effective average value of β up to the amplitude "a" is equal to a constant multiplied by $Z/(1-a^2/c^2)^{\frac{1}{2}}$, where c is the radial clearance of the bearing. If now we write $q_a=\left(\frac{2\omega}{\omega_s}\right)\Big/\phi\left(\frac{\omega_s}{2\omega}\cdot\frac{a}{c}\right)\sqrt{1-\left(\frac{a}{c}\right)^2}$ and let $f\left(\frac{a}{c}\right)$ equal (q_a-1/q_a), we are then able to demonstrate that $\frac{\Delta}{c}$ is equal both to $\frac{a}{c}\cdot\phi\left(\frac{\omega_s}{2\omega}\cdot\frac{a}{c}\right).f\left(\frac{a}{c}\right)$ and to $\frac{b}{c}\cdot\phi\left(\frac{2\omega}{\omega_s}\cdot\frac{b}{c}\right)q_b.f\left(\frac{b}{c}\right)$ where Δ is the deflection which would be obtained by a steady load equal to the amplitude of the force component and subjected to a uniform elastic restraint equal to k_o and directly opposing the steady load. *"a"* and *"b"* are the major and minor radii respectively of the elliptical locus previously referred to, the *"a"* axis being the direction



$$\frac{\Delta}{c} = \frac{a}{c}\cdot\phi\left(\frac{\omega_s}{2\omega}\cdot\frac{a}{c}\right)\cdot f\left(\frac{a}{c}\right)$$

FIG. 2.



$$\frac{\Delta}{c} = \frac{b}{c}\phi\left(\frac{2\omega}{\omega_s}\cdot\frac{b}{c}\right)q_b f\left(\frac{b}{c}\right)$$

FIG. 3.

of the periodic force considered. In Figs. 2 and 3 these functions have been plotted against a/c and b/c respectively, for a series of harmonic component frequencies ranging from ω_s to $6\omega_s$.

All that is now necessary is to calculate Δ for each harmonic component of the force curve by reference to the appropriate curve in Fig. 1 and read off the corresponding values of the amplitudes *"a"* and *"b"* from Figs. 2 and 3. The component deflection curves can then be plotted and combined to give a resultant deflection curve corresponding to the given variable force. In the case of a big-end bearing it will, of course, be necessary to take into account the lateral forces due to the obliquity and inertia of the connecting rod. The method of dealing with these lateral forces is included in the mathematical analysis previously referred to.

The curves A, D and E shown in Figs. 4 and 5 have been

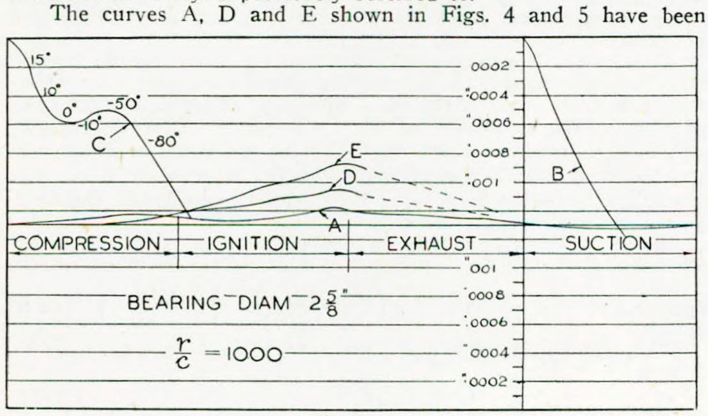


FIG. 4.

determined in this way, but the curves B and C were calculated by a different method because they relate to conditions under which the crankshaft is being accelerated. The curves in these figures apply to the big-end bearings of a typical 50 h.p. petrol engine having 2⅝in. diameter big-end bearings but, with the possible exception of curve A, they are equally applicable to any other engine having the same r/c ratios and in which the square of the cylinder bore multiplied by the m.e.p. is equal to 400 times the bearing area, provided ZN has the same value as that used for the curves. For different bearing diameters, the film thickness would be changed in proportion.

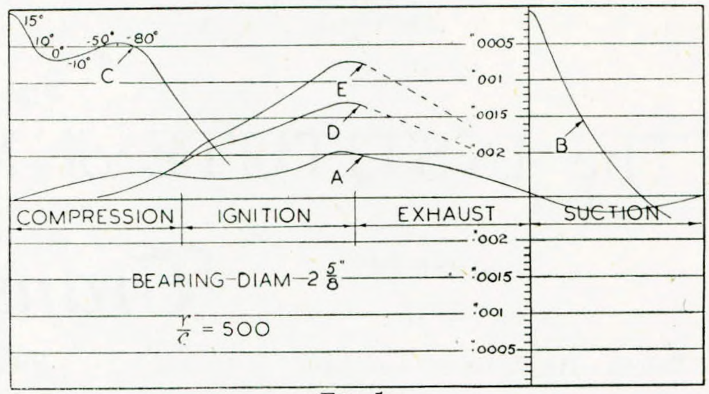


FIG. 5.

Time has not permitted the preparation of similar curves for the main bearings where, however, the conditions are not usually so severe.

Variation in Oil-film Thickness.

Curve A shows the variation in the oil-film thickness during a normal working cycle. Curve B shows the oil film variation in the case of a piston starting from rest at the beginning of the suction stroke, and curve C when starting from rest at the beginning of the compression stroke, the crankshaft being assumed to have a uniform angular acceleration of 12·6 radians per second per second. Curve D shows the variation in oil film thickness when the first ignition stroke occurs at 135 r.p.m., the indicator diagram showing a m.e.p. of 50lb. per sq. in., and curve E is a similar curve for a m.e.p. of 100lb. per sq. in.

Curves A, B, D and E are plotted so that the oil-film thickness on both sides of the big-end bearing can be seen at a glance, but the actual position of the minimum oil thickness is not indicated. In curve C the minimum oil film thickness is given by the ordinate as in the case of the other curves, but the position where this minimum value occurs is indicated by the angles written against the curve. These angles are measured from a point on the crankpin nearest to the crankshaft axis, and in the same direction as the direction of rotation. It will be noticed that, for curves B and C, it has been assumed that, when the engine is at rest, the big-end bearings are practically touching the journals. This, in practice, is justified because, if the piston rings provide sufficient friction to support the weight of the piston and connecting rod, the difference in linear contraction during cooling of an aluminium crankcase and a steel connecting rod is sufficient to force the bearings into the position assumed.

That side of the crankpin nearest the crankshaft axis will be called the inner side. Now it has been observed on frequent occasions that more wear sometimes occurs on the inner side than on the outer side of the crankpin and attempts have been made to explain this. The author is of the opinion, however, that the explanation is given, partly at least, by curve C. If we refer to curve C in Fig. 5 we see that while a considerable movement occurs between the journal and the bearing, the oil film on the inner side of the crankpin is only about 0·0005in. thick. Thus if there are a few abrasive particles larger than 0·0005in. diameter in this oil film, they may become embedded in the bearing metal and cause wear whenever the engine is started up with the piston in the position considered.

Curve A shows that under normal running conditions the oil-film thickness is sufficient to allow the passage through the bearing of comparatively large particles without injury to the bearing. Curve B shows that the crankpin rapidly approaches its central position in the bearing when starting at the beginning of the suction stroke, while curve D indicates that under normal starting conditions the first ignition stroke, in spite of the gas pressure at slow speed, still provides a fair margin of safety.

There are, of course, other working conditions besides those covered by the curves referred to, such as, for example, shock loads occasioned by rapid acceleration or stopping, detonation, etc. These, however, are usually of short duration.

Regarding the effect of engine speed, there is little change in the ignition peak value of curve A for speeds between 1,350 and 3,000 r.p.m. At speeds below 1,350, however, the ignition peak value increases. At about 500 r.p.m., with a m.e.p. of 100lb. per sq. in., it reaches that of the curve D unless Z is increased due to a fall of temperature. The position within the bearing at which the minimum film thickness occurs changes with the speed because of the change in the relationship between the inertia forces and the forces due to gas pressure.

In the case of the main bearings, misalignment or shaft distortion would have an important effect on the oil film thickness.

End leakage at the bearings has been neglected, but this has been partly compensated by assuming a rather high m.e.p.

The practical difficulties in determining the variation in oil-film thickness between the piston rings and cylinder wall are such as to render almost futile any attempt at calculation. We are mostly concerned with the minimum oil-film thicknesses which, in the case of the cylinder, occur during the change in direction of motion of the piston.

How and When Wear Occurs.

From curve A it would appear that there should be practically no wear in a big-end bearing during normal running conditions provided the oil is reasonably clean. This leads us to consider how and when does such wear occur. A little consideration will show us that wear must occur intermittently. If we were to assume the wear of a cylinder liner to be taking place uniformly throughout, say, 100 hours service of an engine, we should find that the thickness of metal removed per stroke would be about 6×10^{-10} cm. Since the size of an iron molecule is about 3×10^{-8} cm., it is obviously impossible to remove such a minute layer. Actually the iron particles due to cylinder wear appear to be from 4×10^{-5} to 6×10^{-4} cm. in length. In the case of a bearing wearing normally, for continuous wear to occur over the whole width of the bearing, the thickness removed would be about 2×10^{-10} cm., and it is equally impossible to remove a layer even 100 times as great as this. If we say that nothing less in thickness than 1×10^{-7} cm. could be removed from a bearing and if the wear occurred uniformly over the whole width of the bearing, then it could only take place during about 1/500th part of the running period. The only alternative to this is for the wear to occur over very narrow sections of the bearing at any given instant, and that is most probably what actually occurs in practice. Let us suppose that the lubricating oil in an engine is contaminated with abrasive particles some of which are greater in size than 0·0015in., and that the bearings have the dimensions assumed for the curves in Fig. 4. During normal running of the engine under consideration we have demonstrated that the oil-film thickness is never less than 0·001in., so that a particle of the size 0·0015in. is not likely to be embedded in the bearing metal, but it can be rolled between the metal surfaces and thus remove a very narrow section of the bearing metal. The amount of metal removed in this way by a single particle may form an almost invisible scratch but, if the oil is badly contaminated, thousands of such minute scratches may occur in a very short period. Having produced a scratch in one position, the chance of producing another in the same position is reduced until the surface on both sides of it are worn, in the same manner, down to the same level. Upon examining the big-end bearings of an engine which had been dismantled for overhaul, many fine circumferential scratches were found, some of which had evidently been produced just prior to stopping the engine.

Referring to curve E, Fig. 4, it will be evident that abrasive particles of the size considered may cause wear at the time the first ignition stroke occurs. The oil-film thickness in this case is reduced to about 0·0008in. and, whereas hard particles somewhat larger than this may cause scratches in the bearing surface in the manner already described, particles as large as 0·0015in. or over might become embedded in the bearing metal. Experimental evidence in support of this argument will be referred to later in this paper.

As already indicated, the worst condition appears to occur when starting an engine with the piston at the beginning of the compression stroke, when particles less in size than 0·0005in. may be effective in causing wear. Fortunately, however, this condition only lasts for a very brief interval of time—for about a quarter of a second in fact.

So far we have assumed that the oil film in the bearing is complete, even after the engine has been standing idle. This will usually be the case unless the bearings have become badly worn. With a badly-worn bearing the continuity of the oil film may be lost after stopping a hot engine, and the effects we have been considering would then be greatly worsened as will readily be understood upon reference to Fig. 1.

It has been established that, so long as satisfactory hydrodynamic conditions are maintained in the bearings, the coefficient of friction is independent of the chemical nature of the lubricant and of the composition of the bearing metal, but depends only upon the viscosity, speed of rotation and dimensions of the bearing. We have to consider whether a large number of small particles in the oil is likely to disturb the hydrodynamic conditions. If there is, say, one gramme per litre of solid contaminants uniformly distributed in an oil, and the average size of the particles is, say, 0·00025in., it can be shown that the average distance between adjacent particles would be about 0·010in., that is to say, the average distance apart of the particles is roughly 40 times the size of the particles for the concentration considered. Time has only permitted the author to make a cursory investigation regarding the effect of this on the flow of oil in a bearing, but it is believed that particles of such size and separation would not appreciably disturb the motion of the oil film. The matter is complicated by the fact that particles near the journal surface will have velocities different from those near the bearing surface. A simple experiment indicated that a particle centrally disposed in an oil film had a strong tendency to retain that position. In this position the speed of the particle is, of course, half of that of the surface speed of the journal if the bearing is stationary. With a sufficient number of particles to cause impacts between them, the movements of the particles were too confused to enable any conclusive observations to be made. It is evident however that, with a sufficient concentration of foreign matter in the oil, not only would the motion of the oil film be disturbed, but wear could occur when a hard particle near the journal meets another hard particle which is near the bearing and therefore travelling at a different speed.

Engine Oil Cleaners—Full Flow Type.

We will now turn our attention to engine oil cleaners or filters of the full flow type, and consider what are the possibilities and practical limitations in removing from the engine oil all particles above a specified size.

Exaggerated claims have been made for some filters and correspondingly impossible demands have been made by some engine makers. It is desirable, therefore, first to obtain an idea of the limitations in engine oil filtration. For this purpose, let us use, as a standard of reference, a filtering medium comprising a single layer of plain weave gauze made of fine hair 0·002in. thick. The resistance to the flow of oil through such a filtering medium can be calculated. This reference standard is only ideal in that it can, theoretically, be constructed to produce a specified filtration fineness with a known

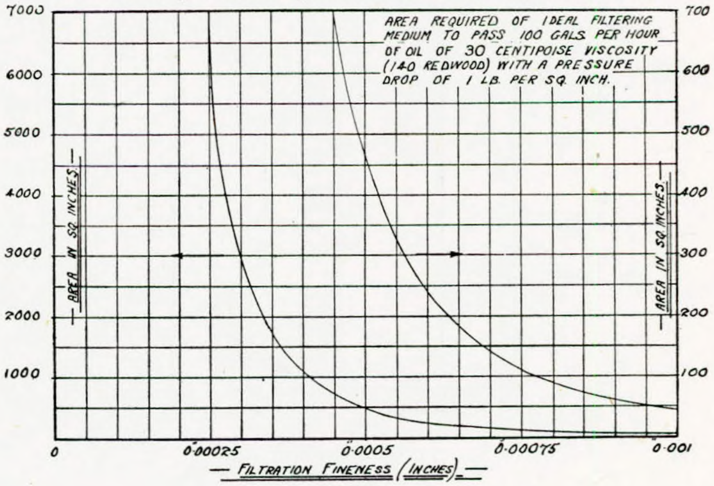


FIG. 6.

comparatively small resistance to flow. In Fig. 6 are shown curves giving the area of such a filtering medium required to provide a given degree of filtration fineness with a pressure drop of 1lb. per sq. in. and with a flow of 100 gallons per hour, the oil having a viscosity of 140″ Redwood. By filtration fineness we mean, of course, the size of the largest spherical particle which can pass through the filtering medium. In practice it will usually be found that from four to 10 times the area obtained from Fig. 6 will be required, but the curves will serve to illustrate the points which we now wish to consider. It will be understood that these curves only apply while the filtering medium is clean. The figures of 1lb, per sq. in. and 100 gallons per hour have been chosen because they facilitate the calculation of the area required for other rates of flow and other pressure drops. The area required will be approximately inversely proportional to the initial pressure drop and directly proportional to the rate of flow.

If it were possible, it would be desirable to remove all particles exceeding 0·00025in. in size, but on reference to Fig. 6 it will be seen that we should require at least 7,000 sq. in. of filtering medium for the rate of flow and pressure drop assumed. On the other hand, to obtain a filtration fineness of 0·001in., only 40 sq. in. of the same filtering medium would be required. In other words, we require 175 times as much area for a filtration fineness of 0·00025 as we should require for 0·001. As far as full flow filters are concerned, therefore, we have the choice of using either (*a*) a super fine filtration and enormous areas, or (*b*) a moderately fine filtration with convenient areas. The former would be very difficult, if not impossible in many cases to accommodate, so that we must consider the disadvantages which accompany the latter. We have seen that, in general, provided the particles do not exceed about 0·001in. they will not cause wear excepting under certain conditions which only occur for very short intervals of time as, for example, when starting a stationary piston from the beginning of a compression stroke.

There are, however, other points to be considered. A filtering medium tends to choke up more or less rapidly according to its initial degree of filtration fineness. If a filter has a filtration fineness of, say, 0·001, this does not mean that all particles less than this size will pass through the filter. The deposit on the filter will, in fact, contain quite a large proportion of particles less in size than 0·001, some of which may subsequently be carried through and replaced by others. Experiments have shown that there is, in fact, a continuous interchange of the smaller particles in the deposit. With finer filtration, we not only obtain a heavier deposit from the same quantity of oil but a denser one. As the deposit builds up on a filter the pressure drop through it increases. The pressure drop increases almost proportionally with the weight of the deposit until a certain critical weight has accumulated. After this critical weight is exceeded, the pressure drop increases progressively more rapidly until ultimately, if the filter is left in service without attention, it becomes useless. The weight of the critical deposit depends upon the type of filtering medium, the initial filtration fineness and upon the rate of flow per unit area of the filter. As the deposit builds up, the filtration fineness of the combination is improved. In many cases it is improved to the extent that the filtration fineness is reduced to about a quarter of its original value by the time the critical deposit is reached.

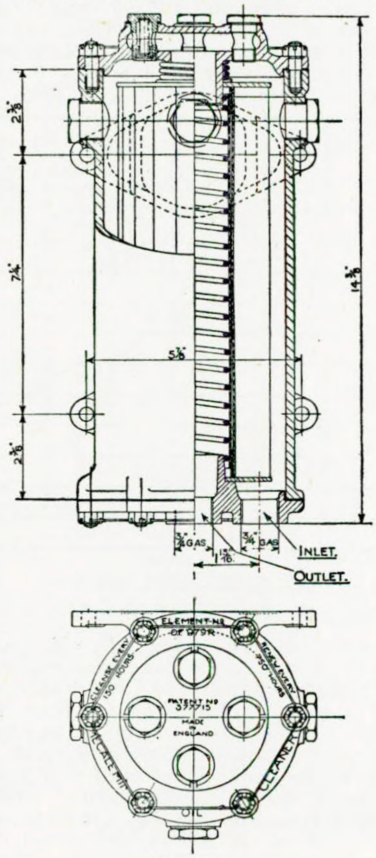


FIG. 7.

All things considered, it appears that an initial filtration fineness of 0·001in. is a very suitable figure for full flow filters. With this initial filtration fineness, if we double the area of a filter, other factors remaining unchanged, we increase the useful life of the filter roughly fourfold, so that where large areas are possible it might be economical slightly to improve the filtration fineness. Even so, only a small improvement in the filtration fineness could be introduced if an increase in the filter life is required to accompany the increase in size.

A brief description of a full flow oil cleaner suitable for use on marine I.C. engines might not be out of place at this stage. A Tecalemit oil cleaner is illustrated by Fig. 7, which is largely self-explanatory. As will be seen, it comprises a casing in which the filter element is mounted between two metal plates, the upper one of which is spring loaded. The cover can be removed for the purpose of replacing or cleaning the filtering element. The filtering medium comprises a special felt the density of which is adjusted to intercept particles exceeding 0·001in. in size. The felt is mounted on a wire gauze support, the cross-section of which is star shaped; the effective area of the felt is about 410 sq. in. In the removable cover are mounted four by-pass valves which, in the event that the filtering element becomes choked due to neglect or that the oil when starting the engine is too cold for all of it safely to pass through the filtering element, allow some of the oil to pass directly from the outside of the filtering element down through the centre thereof to the outlet at the bottom. One of these by-pass valves can be replaced by a device which gives electrical warning when the filtering element needs attention.

The length of time a filter of this type can be left in service without cleaning or removal depends upon the flow-area relationship and upon the condition of the engine. As already indicated, it is desirable to install a filter as large as can conveniently be accommodated but, in any case, the cleaning and replacement instructions provided by the makers should be adhered to in order to ensure a satisfactory performance.

It must not be expected that the oil will remain of good colour with a full flow filter only in use. A comparatively large proportion of the oil contamination consists of very small particles of carbonaceous matter, much of which is not removed by the filter, and this causes the oil to assume a very dark colour. The colour of the oil, however, is not an indication of its serviceableness. A clean-looking oil containing particles which may be invisible to the unaided eye but which may be greater in size than the minimum working oil-film thickness in the bearings, can cause far more wear than a comparatively dirty-looking oil in which all the particles are smaller than the minimum oil-film thickness.

By-pass Oil Cleaners.

Let us now turn our attention to the by-pass oil cleaner of the adsorbent type. As is well known, this type of oil cleaner is constructed to deal with only a small proportion of the oil flow, and a good design is capable of maintaining a good colour of the oil. Such an oil cleaner will remove a very high proportion of the solid contamination from the oil which passes through it. It must be remembered, however, that usually less than 10 per cent. of the oil in circulation passes through the cleaner so that, if some abrasive particles are transferred to the oil in any way, there are at least 10 chances to one that these abrasive particles will reach the bearings before they reach the by-pass cleaner. It is almost certain that if we add new oil to the sump of an engine or withdraw the dipstick for inspection, we shall introduce some dirt into the oil. A clean microscope slide left exposed for a few hours on a shelf in a comparatively clean room was found to have collected numerous particles of dust, some of which comprised hard particles up to 0·002in. in size. Such particles will take several hours to settle through 1in. of engine oil. It can be understood, therefore, that particles of dust and grit are readily transferred to the engine oil by oil measures in spite of all the care which may be exercised. Crankcase aspiration will also be responsible for the addition of dust and grit, and most of this dust and grit will ultimately reach the bearings if only a by-pass cleaner is used. A by-pass oil cleaner of the adsorbent type will, however, ultimately remove from the oil the very small particles which cannot be removed by a full flow filter, and its use in conjunction with a reliable full flow filter is desirable provided it is of a type which has no harmful effect on the oil itself.

Other types of cleaners are, however, arranged to work on the by-pass system. There is, for example :—

The Magnetic Filter.

This is, of course, only intended to remove ferreous particles, and a good design is effective in protecting the bearings of machinery in which the bulk of the solid contamination of the oil comprises iron or steel particles. The greater part of the wear in an I.C. engine is not, in the author's opinion, due to iron particles, so that the use of a magnetic filter alone is insufficient to protect the bearings in such an engine.

Centrifugal Oil Cleaners.

Centrifugal oil cleaners are frequently employed for continuous by-pass purification. The effectiveness of a centrifugal oil purifier depends upon the choice of the right capacity. Upon the capacity depends not only the degree of purification but the size of the limit particles, that is to say the smallest particles which can always be removed from the oil passing through the centrifuge. Whether particles smaller than the limit particles are removed or not depends upon their position in the oil entering the centrifuge. An excellent paper dealing with this subject has already been presented to this Institute by Professor E. Forsberg in February, 1940, in which is also explained a method for determining the optimum working conditions for a given centrifuge. The size of the limit particles for a centrifuge operating under optimum conditions will, for a given design, be governed by the actual size of the centrifuge. For a given substance the linear size of the limit particle is related to the difference between its density and that of the oil. For example, a limit particle of sand would be about twice the linear size of a similarly-shaped limit particle of iron. From the arguments set out in this paper, therefore, it appears to be important to ensure that the limit size of the lighter abrasive particles does not exceed 0·001in. The optimum working conditions previously referred to provide the greatest degree of purity reckoned on a percentage contamination basis. Such a basis does not necessarily ensure that a sufficiently small limit particle size is obtained and, if it is not, the use of a larger centrifuge is indicated. It appears to be doubtful whether some centrifuges in use are large enough from this point of view. A well-known advantage of the centrifuge is, of course, its ability to remove water from, and to reduce the acidity of, lubricating oil. If, as is usual, the centrifuge is used in a by-pass system, there is a probability, depending upon the proportion of the total flow passing through the centrifuge, that injurious particles entering the oil will reach the bearings before they reach the centrifuge. For this reason it would appear to be desirable, where possible, to use a reliable full flow filter in addition to the centrifuge. Such a combination may, in fact, give the best possible protection against wear that can be suggested at the present time. The centrifuge would prevent the accumulation of sludge and other substances which tend to choke the filter and the filter would trap injurious particles which by-passed the centrifuge.

Oil Additives.

So much has already been written regarding the formation of carbon, metallic oxides, asphalts, acids and sludge that reference to this subject appears to be unnecessary. Oxidation inhibitors are now added to some engine oils and, from the point of view of filtration, these are desirable because the prevention of sludge formation lengthens the useful life of a filter. Other additives are sometimes used for the purpose of increasing the load-carrying capacity of the oil under boundary working conditions. We have to consider the effect of filters on these oil additives. Unfortunately, the composition of these additives is usually a closely-guarded secret, but the author has been conducting experiments with a view to determining to what extent some of them are removed by filtering media. The experiments have so far been confined to those additives which affect the so-called "oiliness" of the oil. The author would prefer to use the term "olacity" for the property imparted to oils by polar constituents which cling to metallic and other surfaces and on which the minimum safe value of ZN/P depends.

Some filtering media reduce the olacity of an oil to a considerable extent. For example, it has been found that some activated or alkaline filters, usually not of the full flow type, will remove up to 70 per cent. of olacity constituents in the course of a few hours. Others, on the other hand, remove only a comparatively small proportion. During normal running conditions, these constituents may have no effect whatever on the wear of the bearings, but under severe load conditions they may be a very important factor in preventing wear. As an example, using a mineral oil of low olacity, seizure occurred when ZN/P was reduced to unity, whereas with castor oil, which has a high olacity, ZN/P could be reduced to nearly 0·02 before seizure occurred.

Sources of Wear Not Influenced By Oil Filtration.

There are, of course, sources of wear which will not be influenced by an oil cleaner. The dilution of oil by fuel, as is well known, causes a reduction in the viscosity of the oil with a resultant reduction in the oil-film thickness. In a petrol engine dilution usually reaches a more or less stable figure, depending upon the conditions, which is insufficient to cause a serious loss in viscosity. In the case of Diesel engines, however, dilution is cumulative and might lead to serious consequences unless proper attention is given to it. An oil cleaner cannot, of course, play any part in this connection, nor can it influence the wear caused by dirt entering the cylinders with the air or fuel.

Oil Filters Which Do Not Prevent Wear.

It may now be evident that some filters may have little or no effect in preventing wear. If a filter passes particles larger in size than 0·002in., it is obvious that all of those which are abrasive and reach the bearings can cause wear. If, in addition, the filter seriously reduces the olacity of the oil, its use may actually cause an increase in the rate of wear.

Practical Wear Tests.

If several engines of similar type and make are used without oil cleaners under identical conditions and for the same length of time, it will usually be found that far more wear will take place in some than in others. It has already been shown that dust and grit can readily be introduced into the oil, and it is probable that some of the engines will have been more fortunate than others in this respect. It has been stated in connection with some tests carried out in America on 12 similar aeroplane engines which were used over similar territory for approximately similar periods, that the cylinder wear in the most badly-worn engine was as much as seven times as great as that in the least badly-worn engine. Thus, it is very difficult to obtain truly comparative data regarding the effect of filtration on engine wear, but there is evidence to show that, when a suitable oil filter is fitted, engine wear is reduced.

A very instructive example of bearing wear has been brought to the author's notice. In this we have a clear indication of the action on a bearing of the larger and smaller particles carried by oil, the

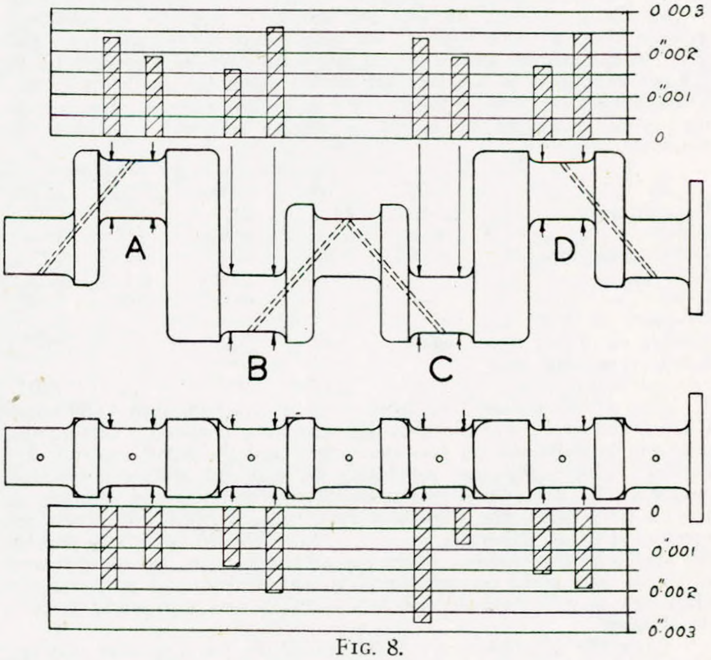


FIG. 8.

results being produced in the same engine. Referring to Fig. 8, this shows a chromium-plated 1 per cent. Ni steel crankshaft used in a 24·8 h.p. R.A.C. rating commercial vehicle petrol engine. On this is shown a diagrammatic representation of the wear on four diameters on each of the crankpins as obtained by accurate measurements after 34,000 miles. The shaded areas referred to the scales on the right of the drawing represent the amount of wear on the diameter indicated. The diagram illustrates the remarkable results of a separation by centrifugal effect of the larger particles which was brought about quite unintentionally. Oil was fed directly to the three main bearings and the wear of the main journals was practically uniform in an axial direction, but it will be observed that on each of the crankpins more wear occurred on one half than on the other. During the passage of the oil through the oil channel leading to the crankpin, the larger particles were apparently caused by centrifugal force to move to the outer side of the oil channel with the result that, in the case of the crankpins marked A and C, the larger particles were delivered to the left-hand half of the bearing and, in the case of the crankpins B and D, the larger particles were delivered to the right-hand half of the bearing. In each bearing particles were embedded in the bearing material on that half corresponding to the side of maximum wear, as shown in Fig. 9.

Thus most of the particles passing to the right of bearing A, for example, were too small to become embedded in the bearing material but were large enough to cause wear. On the other hand, many of the particles passing to the left of bearing A were sufficiently large to become embedded in the bearing material with the result indicated by the diagram. This not only supports the theory described earlier in this paper but indicates that, if the larger particles responsible for the wear shown by the longer shaded areas had been removed by an oil cleaner, we should at least have had the wear reduced to that represented by the shorter shaded areas.

The effect above referred to is not uncommon. Results far worse than this, but not accompanied by accurate wear measurements, have been brought to the author's notice.

Apparently the only useful practical figures at present available relating to the effect of filtration on engine wear comes from petrol engines. Practical tests on serviceable engines under normal running conditions require very long periods of time to obtain useful data and, as already indicated, it is difficult to obtain satisfactory comparisons because of the very different results frequently obtained with similar engines operating under apparently similar conditions. Engine owners who are sufficiently interested in the matter to co-operate in making the necessary observations are also sufficiently careful of their engines to insist on having a filter fitted. Careful observations over a long period have been made on a 2-litre M.G. car which is fitted with a Tecalemit oil cleaner, and the results obtained are included in this paper. In order fully to appreciate the significance of these results one should have some idea of the effect of omitting the filter, but the only information which the author has been able to find regarding engines not fitted with a filter has been obtained by searching through various publications. Figures so obtained indicate that, without a filter, we might expect oil used for 1,000 miles in a motorcar engine to contain solid contamination of 1·5 to 2·3 per cent. and to produce an ash of from 0·5 to 0·7 per cent. These figures can only be taken as a rough basis for comparison with the figures provided by the M.G. engine referred to which are tabulated below :—

	Total distance travelled in miles.				
	8,000	10.000	16,000	21,300	21,900
Fuel dilution ...	2·8%	4·0%	3·6%	2·1%	3·6%
Water	trace	trace	trace	nil	trace
Solid contamination...	0·18%	0·10%	0·15%	0·15%	0·21%
Ash	—	—	0·09%	0·10%	0·03%
Iron	—	—	0·04%	0·05%	0·03%
Viscosity at 140° F.					
Sample as drawn ...	200″	160″	166″	—	165″
After removing unburnt fuel	260″	220″	223″	—	210″

The mileages stated are those recorded after the first 1,500 miles when both the oil and the filter element were changed. During the next 21,300 miles the oil was unchanged and the filter was not disturbed. The engine was topped up as required with a compound engine oil of a grade to suit seasonal requirements and samples of the oil were taken for inspection from time to time. It is not recommended that either the oil or the filter should be left in service for such a long period but, at the end of 16,000 miles, the oil appeared to be in such good condition that it was decided not to drain the crankcase as it was considered that useful information might thereby be obtained.

Upon the completion of 21,300 miles, the filter element was replaced by a new one and the deposit on the old element was examined. The total deposit remaining on the old element after draining it at steam heat for 48 hours was found to weigh 35 grammes. This deposit was analysed and was found to comprise :—

Volatile matter	...	83·3%
Fixed carbon	...	8·9%
Metals	... ...	7·4%
Silica	... ...	0·4%

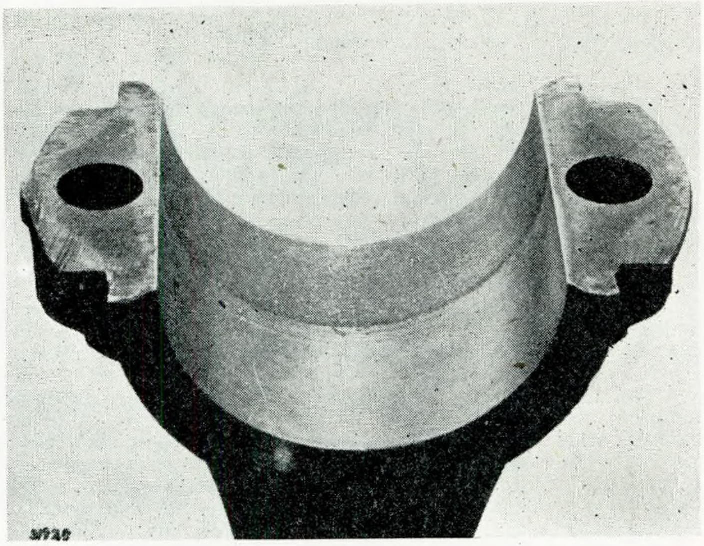


FIG. 9.

It was found that the total amount of iron removed by wear during the 21,300 miles was 6·1 grammes, from which it was estimated that the cylinder wear for this mileage was of the order of 1·5 thousandths of an inch. This is, of course, a remarkably good figure bearing in mind that it is not at all uncommon to find as much cylinder wear as this in about 2,000 miles. Evidence of the low wear in this engine was provided by the fact that even with the thinnest possible oil in use in the engine the original pressure and oil consumption figures were almost fully maintained.

After the new element above referred to had been in use for about 600 miles, a further sample of the oil was removed for examination. Twelve small samples of this oil were examined under a high-power measuring microscope. In all of these samples quite 80 per cent. of the visible particles were smaller than 0·00004in. in diameter. Isolated particles of larger size were found, the largest of which was 0·00032in. Only 13 of these larger particles were found, and the majority of these were less than 0·0002in. in size. Other tests of this sample appear in column 5 above.

The following is a summary of the points which the author has endeavoured to demonstrate :—

(1) During the normal running of a satisfactory engine, that is to say one not suffering from misalignment or excessive wear, the minimum oil-film thickness in the bearings is seldom, if ever, less than 0·001in. thick.

(2) A very much smaller oil-film thickness is probable at the moment of starting up an engine, and the effect of this may be most severe for such of the big ends as are connected to pistons on the point of beginning a compression stroke. These effects are, however, of very short duration.

(3) For running speeds as low as 500 r.p.m. the minimum oil film may be reduced to about half of that obtained during normal running speeds so that rapid accelerations from low speeds may be conducive to wear.

(4) Under normal running conditions very little wear is likely to occur provided there are no abrasive particles in the oil larger than 0·001in., but if the concentration of solid contamination becomes excessive, much smaller particles can cause wear even under normal running conditions.

(5) Wear is intermittent and is largely caused by abrasive particles passing between or being trapped by the bearing surfaces.

(6) Owing to considerations of space and weight, it is not practical to make full flow filters which will intercept all particles appreciably smaller than 0·001in. Other things being equal, a full flow filter having a filtration fineness of 0·00025 would require about 170 times the area of a filter having a filtration fineness of 0·001.

(7) Full flow filters cannot prevent the oil becoming discoloured, but the colour of the oil is not an indication of its serviceableness.

(8) A by-pass oil cleaner of the adsorbent type can remove a large percentage of the solid contamination from the oil which actually passes through it, and it can maintain a good colour of the oil as long as it remains effective. It is important to remember, however, that at least 90 per cent. of the abrasive matter which enters the oil will reach the bearings before it reaches the by-pass cleaner unless an effective full flow filter is fitted.

(9) The removal of ferreous particles by means of a magnetic filter is insufficient to protect the bearings of an I.C. engine.

(10) Centrifugal oil purifiers can remove solid and liquid contamination, but it is important to choose a size which will enable the limit particles of the lighter abrasive materials to be kept down to 0·001in. If the centrifuge is used in a by-pass system there is a probability, depending upon the proportion of the total flow passing through the centrifuge, that injurious particles entering the oil will reach the bearings before they reach the centrifuge. For this reason it would appear to be desirable to use a reliable full flow filter in addition to the centrifuge. Such a combination may, in fact, give the best possible protection against wear that can be suggested at the present time. The centrifuge would prevent the accumulation of sludge and other substances which tend to choke the filter and the filter would trap injurious particles which by-passed the centrifuge.

(11) The chances of transferring injurious particles to the oil during topping up or dipstick inspection are considerable.

(12) Some filters reduce the "oiliness" or "olacity" of the oil to a large extent and if, in addition, such filters permit abrasive particles larger than 0·0015in. to reach the bearings, their use may actually increase rather than decrease the rate of wear.

Appendix.

Effects of dynamic loading on a lubricated bearing.

Referring to Fig. 10, let us first consider a shaft rotating at ω_s radians per second in a bearing having its centre at the origin O, and let the axis of the shaft (at right angles to the paper) pass through the point m where the mass m of the shaft is assumed to be concentrated. Assume that the lateral movements of the shaft are only restricted by the hydraulic reactions of the oil film within the bearing.

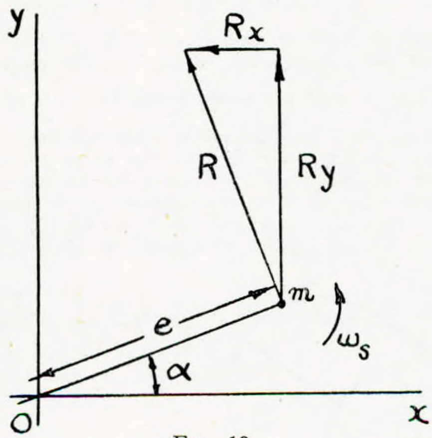


FIG. 10.

It has been shown by previous investigators that, for a fully-lubricated complete bearing, there will be a hydraulic reaction R at right angles to Om, where $Om=e$, the eccentricity of the shaft. As shown with reference to Fig. 1, for a constant force F', F'/e is equal to a constant k for small values of e. Thus we have $R=k.e.$ Therefore the horizontal and vertical components of R are respectively

$$R_x = -k.e.\ \sin\alpha = -k.y.$$
$$\text{and } R_y = \ k.e.\ \cos\alpha = \ k.x.$$

Let the viscous damping factor be β, and assume the shaft to be subjected simultaneously to a vertical periodic force and a horizontal periodic force respectively equal to

$$F_y = F_1 \sin \omega_1 t, \text{ and } F_x = F_2 \sin(\omega_2 t + \theta)$$

We then have the following simultaneous differential equations:

$$m\frac{d^2x}{dt^2} + \beta\frac{dx}{dt} + ky = F_2 \sin(\omega_2 t + \theta)$$
$$m\frac{d^2y}{dt^2} + \beta\frac{dy}{dt} - kx = F_1 \sin \omega_1 t.$$

The particular solutions to these equations are as follows:

$$y = \pm\frac{kF_2}{m^2\sqrt{e_2^2+f_2^2}}\sin(\omega_2 t+\theta-\psi_2) \mp \frac{\beta\omega_1\sqrt{1+\dfrac{\omega_1^2m^2}{\beta_1^2}}}{m^2\sqrt{e_1^2+f_1^2}} F_1\sin(\omega_1 t-\phi_1-\psi_1)$$

$$x = \pm\frac{kF_1}{m^2\sqrt{e_1^2+f_1^2}}\sin(\omega_1 t-\psi_1) \pm \frac{\beta\omega_2\sqrt{1+\dfrac{\omega_2^2m^2}{\beta_2^2}}}{m^2\sqrt{e_2^2+f_2^2}}\sin(\omega_2 t+\theta-\phi_2-\psi_2)$$

where

$$e = \frac{\beta^2\omega^2}{m^2} - \frac{k^2}{m^2} - \omega^4 \qquad \tan\phi = \frac{\beta}{m\omega}$$
$$f = \frac{2\beta\omega^3}{m} \qquad \tan\psi = \frac{f}{e}$$

$$e^2+f^2 = \frac{k^4}{m^4}\left[\left\{\frac{\omega^2}{k^2}(\beta^2-\omega^2m^2)-1\right\}^2 + \frac{4\beta^2m^2\omega^6}{k^4}\right]$$

It will be shown hereafter that, for the cases we are considering, ω^2m^2 is negligible compared with β^2, and that $4\beta^2m^2\omega^6/k^4$ can be neglected when compared with $\left\{\dfrac{\omega^2\beta^2}{k^2}-1\right\}$

Therefore

$$\frac{1}{\sqrt{e^2+f^2}} \simeq \frac{m^2}{k^2\left\{\dfrac{\omega^2\beta^2}{k^2}-1\right\}} \text{ and } \psi \simeq O$$

Now $\tan\phi = \dfrac{\beta}{m\omega}$, but β is sufficiently large compared with $m\omega$, as will be demonstrated, to enable us to put $\phi = \frac{1}{2}\pi$.

Let $\dfrac{\omega\beta}{k} = p$. We can then write the above equations in the following simple form:

$$y = \frac{F_2 \sin(\omega_2 t+\theta)}{k(p_2^2-1)} - \frac{F_1\cos\omega_1 t}{k(p_1-\dfrac{1}{p_1})}$$
$$x = \frac{F_1\sin\omega_1 t}{k(p_1^2-1)} + \frac{F_2\cos(\omega_2 t+\theta)}{k(p_2-\dfrac{1}{p_2})}$$

It is obvious, from these equations, that we can, if desirable, deal with the horizontal and vertical force components separately and then integrate the horizontal and vertical displacements. Putting $F_x = O$, we now obtain the following equations for the motion due to a single vertical periodic force:

$$y = -\frac{F_0\cos\omega_1 t}{k(p_1-\dfrac{1}{p_1})} \quad \text{.......... (I)}$$
$$x = \frac{F_0\sin\omega_1 t}{k(p_1^2-1)} \quad \text{.......... (II)}$$

So that we have

$$x^2(p^2-1)^2 + y^2\left(p-\frac{1}{p}\right)^2 = \left(\frac{F_0}{k}\right)^2 \quad \text{.......... (III)}$$

From equation III, therefore, it appears that the locus of the centre of the journal is an ellipse for a single periodic force. The major radius of the ellipse is

$$a = \frac{F_0}{k\left(p-\dfrac{1}{p}\right)}$$

and the minor radius is

$$b = \left(\frac{F_0}{k(p^2-1)}\right)$$

The complementary functions for x and y obtained by putting both F_y and F_x equal to zero in the original differential equations are equal to

$$Ae^{\lambda_1} + Be^{\lambda_2} + Ce + De^{\lambda_4}$$

$$\text{where } \lambda = -\frac{\beta}{2m} \pm \frac{\beta}{2m}\left(\frac{\sqrt{1+\left(\dfrac{4km}{\beta^2}\right)^2}+1}{2}\right)^{\frac{1}{2}} \pm i\frac{\beta}{2m}\left(\frac{\sqrt{1+\left(\dfrac{4km}{\beta^2}\right)^2}-1}{2}\right)^{\frac{1}{2}}$$

These expressions relate to transient motion which, for our purpose, can be neglected.

If we consider the shaft axis as fixed and the bearing and its support as oscillating, it will be understood that similar equations will be obtained. In the case of a crankpin, however, the effective value of m for the y axis will be different from that for the x axis. However, it will be shown that this is not of much importance.

In the above analysis, we have assumed the oscillations to be sufficiently small to regard both k and β as substantially constant.

Let us now consider the value of k and determine its effective average value for a given amplitude.

Fig. 11 is intended to represent the position of a shaft relative to the centre of a bearing when the shaft is subjected to a constant

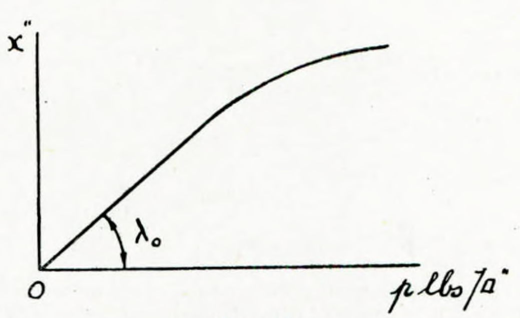


FIG. 11.

load $F_{,,,}$ the product ZN being also constant. Let A equal the area of the bearing and let

$$k_o = pA/x_{\to o} = A/\tan\lambda_o \quad \text{.................................} \quad \text{(IV)}$$

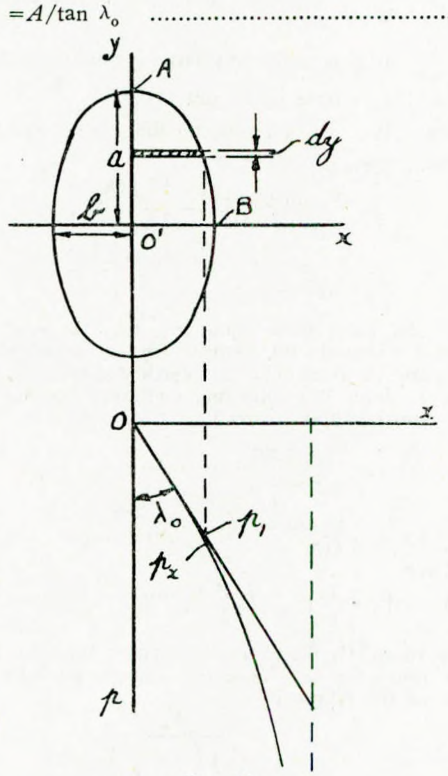


FIG. 12.

Referring now to Fig. 12, the ellipse is intended to represent the motion of the centre of a shaft relative to the centre O' of a bearing.

First consider the case where k is a constant equal to k_o. Then the reaction parallel to the axis Oy due to k is equal to $k_o x$, and the work done by a movement dy

$$= dW = k_o x.dy.$$

Therefore, in this case, the work done in the direction Oy when moving from B to A is equal to

$$Wy = k_o \times \text{the area of the quadrant } OAB.$$

Let k_x equal the actual value of k corresponding to x.

Now $k_x = p_x A/x$ and $k_o = p_1 A/x$ (see Fig. 12 for p_1 and p_x). Therefore k_x/k_o equals p_x/p_1. But $dW = k_x.x.dy$.

Therefore $\dfrac{dW}{k_o} = \dfrac{k_x}{k_o}.x.dy = \dfrac{p_r}{p_1}.x.dy.$

Now it has been established by other investigators that

$$p_r = \frac{b\left(\frac{x}{c}\right)}{\left(2+\frac{x^2}{c^2}\right)\sqrt{1-\frac{x^2}{c^2}}} \text{ where } b = 6\pi\mu\omega_s\left(\frac{r}{c}\right)^2$$

$$\therefore \frac{p_x}{p_1} = \frac{2}{\left(2+\frac{x^2}{c^2}\right)\sqrt{1-\left(\frac{x}{c}\right)^2}}$$

Therefore, if the work done in the direction Oy from B to A is W', then

$$\frac{W'}{k_o} = \int_o^{x=b} \frac{2x\,dy}{\left(2+\frac{x^2}{c^2}\right)\sqrt{1-\left(\frac{x}{c}\right)^2}}$$

Let k_a equal the effective average value of k, that is to say the value of k which will require the same work to be done in moving from A to B, and let E equal the area of the quadrant OAB. Then $W'/k_a = E$, so that

$$\frac{k_a E}{k_o} = \int_o^{x=b} \frac{2x\,dy}{\left(2+\frac{x^2}{c^2}\right)\sqrt{1-\left(\frac{x}{c}\right)^2}}$$

$$\text{or } k_a = \frac{8k_o}{\pi b^2} \times \int_o^b \frac{-x^2.dx}{\left(2+\frac{x^2}{c^2}\right)\sqrt{\left(1-\frac{x^2}{c^2}\right)(b^2-x^2)}}$$

$$\text{Let } \phi\left(\frac{b}{c}\right) = \frac{8}{\pi}\left(\frac{c}{b}\right)^2 \int_o^b \frac{-\left(\frac{x}{c}\right)^2 dx.}{\left(2+\frac{x^2}{c^2}\right)\sqrt{\left(1-\frac{x^2}{c^2}\right)(b^2-x^2)}}$$

$$\text{Then } k_a = \phi\left(\frac{b}{c}\right)k_o \quad \text{.........................} \quad \text{(V)}$$

where k_a is the effective average value of k which should be used for determining the value of the axis "a" of the ellipse. Similarly $k_b = \phi\left(\frac{a}{c}\right) k_o$ can be used for determining the value of the "b" axis. In due course we shall find the relationship between a and b.

It is now necessary to determine the value of β, the viscous damping factor, and its effective average value for a given amplitude.

To find the value of β we have to determine the force acting on the bearing due only to a given velocity of the axis B of the bearing relative to the axis O of the shaft, where $BO = x$.

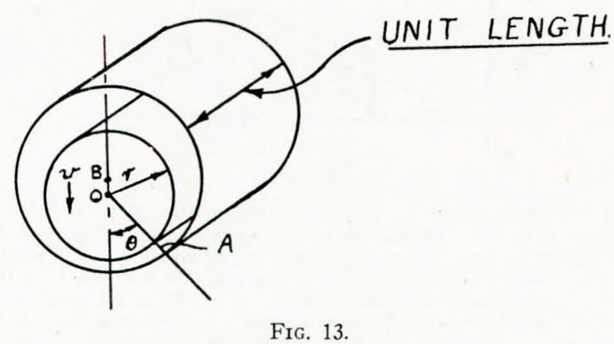


FIG. 13.

Referring to Fig. 13, the extra amount of oil passing through the gap A due to a downward movement dx of the shaft relative to the bearing is equal to

$$r \sin\theta dx$$

Therefore the extra rate of flow at this point $= Q = r \sin\theta \dfrac{dx}{dt}$

$$= r.\sin\theta v$$

Therefore, over a narrow element $r.d\theta$ at the gap A of the bearing, the change of pressure due to Q is equal to

$$dP = \frac{12Q\mu r d\theta}{(c-x\cos\theta)^3} = \frac{12\mu v r^2 \sin\theta d\theta}{(c-x\cos\theta)^3}$$

so that the pressure at A due to v is equal to

$$P_\theta = 12\mu v r^2 \int_o \frac{\sin\theta d\theta}{(c-x\cos\theta)^3}$$

$$= \frac{6\mu v r^2}{x}\left\{\frac{1}{(c-x)^2} - \frac{1}{(c-x\cos\theta)^2}\right\}$$

Therefore the vertical force over the narrow element at A is equal to $dF = P^\theta r d\theta \cos\theta$

$$\therefore dF = \frac{6\mu v r^3}{x}\left\{\frac{\cos\theta d\theta}{(c-x^2)} - \frac{\cos\theta d\theta}{(c-x.\cos\theta)^2}\right\}$$

and the total vertical force due to v (per unit length of bearing) is equal to

$$\frac{12\,\mu v r^3}{x}\left\{\int_0^\pi \frac{\cos\theta d\theta}{(c-x)^2} - \int_0 \frac{\cos\theta d\theta}{(c-x\cos\theta)^2}\right\}$$

$$= \frac{12\pi\mu v r^3}{(c^2-x^2)^{\frac{3}{2}}}$$

or, if the length of the bearing is l, the total force acting on the bearing due to v is

$$F = \frac{12\,\mu\pi v r^3 l}{(c^2-x^2)^{\frac{3}{2}}} \text{ dynes in c.g.s. units}$$

$$= 5{\cdot}46\times10^{-6}\, Zv'l'' \left(\frac{r}{\sqrt{c^2-x^2}}\right)^3 \text{lb.}$$

where Z is in centipoises, v' in inches/sec., and l'' is in inches. But $\beta = F/v$

$$\therefore \beta = 5{\cdot}46\times10^{-6}\, Zl\left(\frac{r}{\sqrt{c^2-x^2}}\right)^3 \text{lb. sec./inch} \;\ldots\ldots\ldots\ldots \text{(VI)}$$

Let $K = 5{\cdot}46\times10^{-6}\, Zl\left(\frac{r}{c}\right)^3$ in lb. in. sec. units.

$$\text{Then } \beta = K\Big/\left(1-\frac{x^2}{c^2}\right)^{\frac{3}{2}} \;\ldots\ldots\ldots\ldots\ldots\ldots \text{(VIA)}$$

To determine the effective average value of β, let the resistance due to $\beta dx/dt$ bring to rest at x_1 a mass m which starts with a velocity v_1 at $x=O$.

The K.E. of the mass $=E=\frac{1}{2}mv^2$, so that $dE/dv=mv$. But $dE=-\beta v.dx$. Therefore $dv/dx=-\beta/m$, and we have

$$-m dv = \beta dx = \frac{K dx}{\left(1-\frac{x^2}{c^2}\right)^{\frac{3}{2}}}$$

Therefore

$$m\int_0^{v_1} dv = K\int_0^{x_1}\frac{dx}{\left(1-\frac{x}{c^2}\right)^{\frac{3}{2}}}$$

$$\therefore \frac{mv_1}{K} = \frac{x_1}{\sqrt{1-\frac{x_1^2}{c^2}}} = x_1\left(\frac{\beta x_1}{K}\right)^{\frac{1}{3}}$$

Let βa = the effective average value of β up to x, then

$$\frac{mv_1}{K} = \frac{\beta a}{K}\int_0^{x_1} dx = \frac{\beta a x_1}{K}$$

$$\therefore \frac{\beta a}{K} = \frac{1}{\sqrt{1-\frac{x_1^2}{c^2}}}$$

$$\therefore \beta a = \frac{K}{\sqrt{1-\frac{x^2}{c^2}}} \;\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(VII)}$$

We can now find the relationship between β and k. According to an existing formula,

$$\frac{\mu\omega_s}{p}\left(\frac{r}{c}\right)^2 = \frac{\left\{2\left(\frac{c}{x}\right)+1\right\}\sqrt{\left(\frac{c}{x}\right)^2-1}}{6\pi\left(\frac{c}{x}\right)^2}$$

$$= \frac{(2c^2+x^2)\sqrt{c^2-x^2}}{6\pi c^2 x}$$

$$\therefore \frac{p}{x} = \frac{6\pi c^2\mu\omega_s\left(\frac{r}{c}\right)^2}{(2c^2+x^2)\sqrt{c^2-x^2}}$$

$$\text{therefore } k_o = \underset{\to o}{\frac{F}{x}} = \underset{\to o}{\frac{p(2rl)}{x}} = \left(\frac{12\pi c^2\mu\omega_s rl\left(\frac{r}{c}\right)^2}{(2c^2+x^2)\sqrt{c^2-x^2}}\right)_{x\to o}$$

$$= 6\pi\mu\omega_s l\left(\frac{r}{c}\right)^2$$

$$\therefore k_o = \tfrac{1}{2}K\omega_s \;\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(VIII)}$$

Thus, from equations VII and VIII, we see that, at $x\to o$, $\beta/k = 2/\omega_s$

Let us now consider the relationship between the values of k, m and β. It will be convenient to use inch/lb./sec. units. Referring to Fig. 1 it will be seen that $k_o = A/c.\tan\lambda_o = F/x$, where A is the area of the bearing, c is the radial clearance and λ_o is the slope of the chosen curve in this figure for $p=O$. Thus, for the two complete film curves shown, and using a $2\frac{5}{8}$in. diameter $\times 1\frac{9}{16}$in. big-end bearing, $k_o = 2\times10^7$ approximately when $r/c = 1{,}000$, and $=2{\cdot}4\times10^6$ when $r/c=500$. On the other hand m is of the order of 7lb./386, say 0·02 mass units. Under the same working conditions for small values of x, β will have twice the values given for k/ω_s.

Thus the earlier statement that the factors including m can be neglected is justified.

Reverting now to equations (I) and (II), the value of p is defined as $\omega\beta/k$. Now for a single vertical periodic force it will be seen that a/b, the ratio of the axes of the ellipse, is equal to p which, as will be seen from equations (VII) and (VIII) is equal to $2\,\omega/\omega_s$. From equation (V), we can therefore write

$$k_a = \phi\left(\frac{\omega_s}{2.\omega}\cdot\frac{a}{c}\right)k_o \;\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(IXa)}$$

$$k_b = \phi\left(\frac{2\omega}{\omega_s}\cdot\frac{b}{c}\right)k_o \;\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(IXb)}$$

We can now substitute q for p in equations (I) and (II), where $q=\omega\beta_a/k_a$, using the effective average values of β and k.

$$\text{Thus } q_a = \frac{2\left(\frac{\omega}{\omega_s}\right)}{\phi\left(\frac{\omega_s}{2\omega}\cdot\frac{a}{c}\right)\sqrt{1-\left(\frac{a}{c}\right)^2}} \;\ldots\ldots\ldots\ldots\ldots\ldots \text{(Xa)}$$

$$q_b = \frac{2\left(\frac{\omega}{\omega_s}\right)}{\phi\left(\frac{2\omega}{\omega_s}\cdot\frac{b}{c}\right)\sqrt{1-\left(\frac{b}{c}\right)^2}} \;\ldots\ldots\ldots\ldots\ldots\ldots \text{(Xb)}$$

If now we let $\Delta = F_a/k_a$, the deflection which would be obtained by a steady load against a constant elastic restraint equal to k_a, and if $f(a/c) = (q_a - 1/q_a)$ and $f(b/c) = (q_b - 1/q_b)$, we can then write:

$$\frac{\Delta}{c} = \frac{a}{c}\phi\left(\frac{\omega_s}{2\omega}\cdot\frac{a}{c}\right)f\left(\frac{a}{c}\right) = \frac{b}{c}\phi\left(\frac{2\omega}{\omega_s}\cdot\frac{b}{c}\right)q_b f\left(\frac{b}{c}\right) \ldots\ldots\ldots \text{(XI)}$$

The functions (XI) have been plotted against a/c and b/c in Figs. 2 and 3, in which their method of application is described.

It is not always desirable to consider the vertical and horizontal forces separately. As far as the gas pressure forces are concerned it may be desirable, but the inertia forces can more conveniently be treated in the following manner.

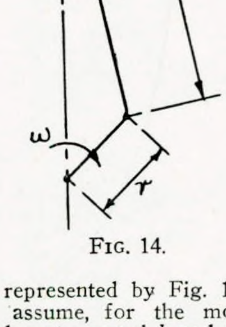


FIG. 14.

In the mechanism represented by Fig. 14, consider the inertia forces separately and assume, for the moment, that these are sufficiently small to enable us to regard k and β as constant. Neglecting the obliquity of the connecting rod, we can then write

$$F_y = m_1\omega^2 r\cos\omega t.$$

$$F_x = m_2\omega^2 r\sin\omega t.$$

Using equations (I) and (II), we then obtain

$$y=(a_1+b_2)\sin\omega t$$
$$x=(b_1+a_2)\cos\omega t$$

where

$$a_1=\frac{m_1\omega^2 r}{k\left(p-\dfrac{1}{p}\right)} \qquad a_2=\frac{m_2\omega^2 r}{k\left(p-\dfrac{1}{p}\right)}$$

$$b_1=\frac{m_1\omega^2 r}{k(p^2-1)} \qquad b_2=\frac{m_2\omega^2 r}{k(p^2-1)}$$

Now

$$a_1+b_2=\frac{m_1\omega^2 r}{k}\left\{\frac{p+\dfrac{m_2}{m_1}}{p^2-1}\right\}$$

$$b_1+a_2=\frac{m_1\omega^2 r}{k}\left\{\frac{1+\dfrac{m_2}{m_1}p}{p^2-1}\right\}$$

Let $m_2/m_1=\rho$. Since the force frequency equals the angular velocity of the shaft, $p=2$ approximately. m_1 is the mass of the piston and connecting rod taken together, while m_2 is a mass equal to I/L^2, where I is the moment of inertia of the connecting rod about the gudgeon pin.

Let $\Delta_1=m_1\omega^2 r/k$. We then obtain

$$y=\Delta_1.\frac{2+\rho}{3}\sin\omega t \text{ and } x=\Delta_1.\frac{1+2\rho}{3}\cos\omega t.$$

Thus we can treat the inertia forces and the gas pressure forces separately, using the above simple method for the inertia forces. Thus we only have to find a harmonic series for the gas pressure forces.

In dealing with the gas pressure forces, we have to consider the horizontal force components due to the obliquity of the connecting rod. We have assumed that F can be expressed by the series

$$\sum_{n=o}^{\infty} b_n\cos(n\omega_s t)+\sum_{n=1}^{\infty} a_n\sin(n\omega_s t)$$

If we let h equal the corresponding horizontal force acting at the crankpin due to the obliquity of the connecting rod, then $h=F.\tan\alpha$, where α is equal to the angular displacement of the connecting rod.

$$\sin\alpha=\frac{r}{L}\sin\theta$$

$$\therefore \tan\alpha=\frac{\sin\theta}{\sqrt{\left(\dfrac{L}{r}\right)-\sin^2\theta}}\simeq\frac{r}{L}\sin\omega_s t$$

$$\therefore h=\frac{r}{L}\left[\sum_{n=1}^{\infty} a_n\sin(n\omega_s t)\sin\omega_s t+\sum_{n=0}^{\infty} b_n\cos(n\omega_s t)\sin\omega_s t\right]$$

We could now plot this curve, find a new harmonic series corresponding thereto and proceed as before to find the displacement due to h. This, however, is a laborious procedure. It will be obvious that h is small compared with F. As an alternative, therefore, we can obtain a sufficiently close approximation by replacing F by $F_c=F.\sec\alpha$, the force acting along the connecting rod, and using a harmonic series corresponding to F_c for determining the total displacement due to F.

Discussion.

Mr. E. P. Paxman, M.A. (Member), opening the discussion, said: The theory put forward by the author that wear occurred in the bearings of engines due to particles of abrasive matter being rolled between the surfaces was very interesting. It explained the removal of very small sections of the bearing metal, but he did not think that many present felt that this was the only cause of wear. There must be other influences, such as imperfect oil films resulting in metallic contact, which no filter could prevent.

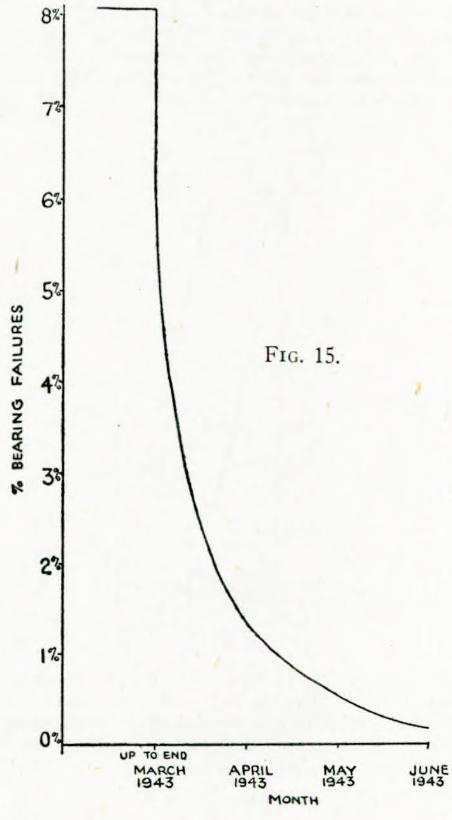


FIG. 15.

It would be interesting to know to what extent the size of the engine and bearings would modify the advice the author gave as to the degree of fineness desirable in a full flow filter. The engines with which Members of The Institute were mainly concerned were large, and the bearing sizes were many times larger than those to which the author had referred; the clearance were proportional.

The filters on engines for driving ships were coarse. Reliance was mainly placed on centrifuges to remove impurities, the filters serving as strainers to trap many of the things which get dropped into the system.

The effect of fitting very fine filters could negative the supposed advantages because the filters were bound to by-pass over too long a time.

There was a lot of evidence that dirt, especially when it was present when the engines were starting, could cause a great deal of trouble.

The accompanying graph was interesting as showing the comparative reduction in seizures on a test bed of new engines when the maximum effort was made to remove dirt from the system, and then to trap what remained before it reached the bearings.

Full flow fabric filters of very large size were fitted during all tests; it was found however that, when starting, they by-passed due to the high oil viscosity and thus allowed any dirt present to enter the bearings. Therefore in later tests even greater precautions were taken that the oil itself should be clean, and in addition relatively coarse non-by-passing filters were installed in series with the full flow fabric ones to intercept dirt when the by-pass was open. The oil in the system was continuously centrifuged and magnetic filters were also installed.

The cumulative result of these precautions, together with the greatest care in cleaning of engine components, had been virtually to eliminate seizures. The tests related to very many thousands of bearings and covered a prolonged period.

Formerly, filters were solid and would have burst if the concentration of dirt was allowed to get very high.

Probably one of the most common causes of wear in Diesel engines was water. His Company's experience was that the presence of water increased the rate of wear enormously.

They had also found that the use of magnetic filters was valuable where there were running gears or parts where metallic abrasion was likely to occur. These filters, by picking up the particles, prevented them from acting as further abrasives. The magnetic filter used in conjunction with filters of the full flow type, which had felt or merely fabric elements, protected them from being pierced by metallic slivers.

What were the author's views as to the optimum size of oil sump? Did he favour a small sump and rapid circulation and cleaning of the oil, or *vice versa?*

Very many points would arise in the discussion of the paper, and the author's presentation of the subject had set them all thinking. This was perhaps the most useful thing an author could do, and accordingly he would like to voice his great appreciation.

Dr. E. Giffen (Research Department, Institution of Automobile Engineers): The paper contained a large number of interesting points but, unfortunately, he (the speaker) had not had sufficient time to study in detail the mathematical treatment given in the Appendix. Something of this nature, however, was very much needed and would be valued by all concerned with the performance of engine bearings.

The paper contained many provocative suggestions and statements. Most of these would be dealt with by others better qualified than himself and he only wished to refer to one particular point. This, however, was a most important point, probably the most important in the paper. This was the conclusion reached by the author: "In general, provided the particles do not exceed about 0·001in., they will not cause wear excepting under certain conditions which only occur for very short intervals". In another paragraph, the author said it would be desirable to remove all particles exceeding 0·00025in. and this, presumably, was suggested to be on the safe side. Now the limiting size of 0·001in. was based on the results of the calculation, as shown in the diagram in Fig. 4, indicating that, for the bearing considered and under normal conditions of running, the oil film was never less than 0·001in. thick. This, however, was an ideal case and was based upon a number of assumptions which did not apply in practice. Of these, one of the most important was the flexibility of the bearing housing and of the shaft itself. In the big-end bearing of an automobile engine, such as was used for the author's calculation, rigidity under load was most certainly not to be expected and such deformation would completely upset the results as calculated and shown in the diagrams (Figs. 4 and 5).

At the I.A.E. Research Department, a series of bearing tests had recently been carried out and it was hoped to be able to measure the oil-film thickness by measuring the electrical resistance of the oil film between the shaft and the bearing. The test bearing was the big-end of a commercial-vehicle engine connecting rod, mounted on the test shaft and loaded by means of a lever and a steady load. In this respect, with a steady, instead of a fluctuating load, the conditions were not the same as in the author's calculation, but the film thickness could easily be calculated. When tested it was found that, while a high resistance was measured at no load and high speed, showing that the oil film was separating the shaft and bearing surfaces, as soon as a very small load was applied the resistance decreased to a very low value. The mounting of the connecting rod was such that it had freedom to align itself to the shaft and the result was taken to indicate that the bearing housing was sufficiently flexible to allow the bearing to distort under quite low loads and to make partial contact with the shaft surface over certain localized areas. In this respect a commercial bearing, as manufactured for a high-speed engine, would be very different from a bearing of the stationary type in which rigidity was more easily attained. When it was pointed out that this bearing was mounted on a stiffly-supported shaft, and in such a way that it had special provision for aligning itself to the shaft, it was clear that even more distortion would be expected in practice, since the shaft's flexibility and the lack of alignment, always present in some degree, would tend to make matters worse than in the case just quoted.

For these reasons, the speaker would suggest that, while the method of calculation was useful as showing the effect of different variables on the oil-film thickness in an ideal bearing, the results could not be directly applied to a commercial type and could not be used as evidence in support of the statement that particles smaller than 0·001in. would not cause bearing wear. The limiting size of particles below which no bearing wear would be produced was, of course, one of the most important factors in determining the degree to which oil cleaning should be carried. This limiting size would vary no doubt with the type of bearing and the duty it performed, but there appeared to be no real evidence as to the limiting size of foreign particles which could be tolerated in the lubricating oil of automobile engines. It was hoped that the tests referred to above at the I.A.E. Research Department would throw some light on this matter, but at present they were only in the preliminary stage.

Mr. A. Beale (Visitor): The speaker's reaction to the opening half of the author's thought-provoking paper was expressed half-a-century ago by W. S. Gilbert:

"If this young man expresses himself in terms too deep for me,
Why, what a very singularly deep young man, this deep young man must be!"

But although the author's mathematical terms might be too deep for the speaker, some of the depths could be plumbed. On page 111 were given curves relating to the oil-film thicknesses in "the big-end bearings of a *typical* 50 h.p. petrol engine having 2⅝in. diameter big-end bearings". The author went on to suggest that they had equal application "to any other engine having *the same r/c* ratios, *and* in which the square of the cylinder bore multiplied by the m.e.p. was equal to 400 times the bearing area, *provided ZN* had the same value".

This suggested a willingness to argue from the particular to the general, and when after an impressive analysis of factors governing filtration rate the author arrived at the conclusion that "all things considered, it appears that an initial filtration fineness of 0·001in is a very suitable figure for full flow filters" one was well prepared for the discovery that he knew the name of just such a filter!

More seriously, the author was to be congratulated on some careful analytical work which would be helpful to all engaged in the assessment of factors governing the choice of oil filters to suit specific engine requirements. In applying his conclusions engine users should bear in mind the following points:

(1) Except for a short period after cleaning or renewal the output of a filter depended on the resistance of the accumulated solids rather than that of the filtering medium itself. Not too much importance should be attached to a flying start.

(2) It had long been realized by engine makers that for ideal protection an engine needed a full-flow strainer *and* an efficient by-pass filter. There should be no sense of competition between the suppliers of the two types of equipment, which were in fact complementary.

(3) In choosing the full-flow strainer the object should be to secure the finest straining action which could reasonably be obtained without premature clogging. It was in providing some guidance on this point that Mr. Langley had proved most helpful.

(4) In choosing the by-pass filter no compromise was possible. In order to prevent the cumulative action of fine impurities, the by-pass filter must be capable of removing *all* of them. The size of the filter should be limited only by considerations of weight and space: the bigger the better from the point of view of good engine performance.

(5) The author mentioned three types of by-pass oil cleaner: "adsorbent", "magnetic" and "centrifugal". It might be permissible to add the type in which the speaker was personally interested, "the Stream-Line filter". This would remove the smallest particles, yet was free from the objections which have been mentioned with regard to the "adsorbent" type of medium. There were other by-pass filters which did not fall within the three groups either, such as those made by The Metafiltration Company and Messrs. Vokes.

(6) The suggestion that particles below a certain size were not damaging must not be allowed to pass unchallenged, although time would not permit going into details. A paper before this Institute in 1934 by Messrs. Le Mesurier and Stansfield included some interesting test results suggesting that particles small enough to "swim" into the oil film had a noteworthy effect on cylinder wear.

There were two other minor points which might be mentioned. The author referred to the ability of the centrifuge "to reduce the acidity of lubricating oil". This was contrary to the speaker's experience, although of course a centrifuge might be used as an adjunct of equipment for reducing acidity by chemical means. Perhaps the author or one of the makers of centrifuges could make the actual possibilities clear.

The author expressed a preference for referring to the property of "oiliness" by the term "olacity". This seemed a pity because engineers were just beginning to have a vague glimmering as to what "oiliness" meant, and life was going to be further complicated if we now had to catch up with "olacity", which was presumably similar but not quite the same! But whatever we call the property, we must hold it as a serious disadvantage of the "adsorptive" type of medium that it should remove anything which the oil experts had deliberately added to oil in order to help it to do its job, and he would like to make it clear that no such adsorptive action was associated with the practical use of the filter with which he was concerned.

Mr. T. R. Stuart (Member): The paper contained much information on a very important subject, particularly in establishing the thickness of the oil film, the size of the particles of dirt which caused wear, the rate of flow and pressure loss through fine mesh filters and the advantages obtained by having the filter as large as possible.

Although the author dealt with a small engine and a flow of only 100 gallons per hour, he, in common with all others, soon found that perfect filtration was impossible on account of the large filter required and he immediately had to compromise. This was the keynote of all engine lubricating-oil filtration. It was necessary to provide an engine with filters which would keep the oil in a condition which would allow it to function satisfactorily. It was an axiom that for any mesh, the larger the filter the better would be the results obtained, and the longer it would function, but there were limits to the size that could be accommodated.

For instance, on the author's basis of 10 times the area obtained from Fig. 6, the filter illustrated would be a large article to accommodate under the bonnet of a small car circulating 100 gallons per hour. If the same ratio of filtering area to flow were adopted, say, for the "Queen Mary", which circulated 288,000 gallons per hour, the filtering area required would be so large that there would be insufficient room for it in the engine room even if the weight of the necessary casing and stowage of spares would be permitted. The same applied to large Diesel-engined ships, but the record of these vessels was proof that the present filtration was satisfactory for their highly-loaded bearings and thrusts.

While not challenging conclusions in the author's paper, it should be emphasized that they represented unattainable ideals at least for commercial practice in respect of large engines, and that there was no guide except that of experience over long periods, as to what was the degree of filtration required. It was interesting to note that to retain similar pressure losses and maintenance requirements, a 0·001in. mesh filter would need to be about 1,890 times larger than existing installations.

It was not sufficient to determine the best mesh, there must be due regard to pressure loss which had probably more effect in efficient filtration than any other factor. It must be kept so low that particles of dirt, fluff, fibres, etc., would lie on the filter surface and increase the filtering efficiency as the author had described. Some industrial filtration problems owed their solution to rigid and automatic control of the pressure loss by means of a differential pressure governor.

The author seemed to infer that wear was caused only by "hard" particles, but it could also be caused by relatively soft material such as fibres from cotton waste, as marine engineers would appreciate from observations of the wear produced by cotton packing.

Another point in the paper which called for comment was that which referred to the by-pass valve coming into action when starting up from cold. The true function of a by-pass valve was to maintain a flow of oil if the filter became choked. It was an emergency fitting and in a well-designed filter should not open at any other time. The filter itself should be capable of dealing with cold oil conditions.

By-pass filters were a very valuable adjunct to main flow filters, especially for small engines and other installations where a Streamline filter of a centrifuge could not be provided because of expense or through lack of space or personnel for their successful operation. In the case of large engines, centrifuges were more efficient and practical and they were light and took up less room.

Fabrics were efficient filtering media if they were kept in good condition, but maintenance and replacement charges were often surprisingly high. Experiments made some years ago with various felts and fabric indicated that the by-pass valve functioned more frequently than was considered desirable, and also emphasized a tendency for dangerous concentrations of dirt to be forced through the fragile element under the influence of a rising pressure loss as the element became choked. For these reasons, robust, all metal filters were preferred and these were available as removable cartridge types, duplicated so that a dirty element could be removed for cleaning while the engine was running, or as self-cleaning types with hand-operated or automatic cleaning mechanism. These showed to great advantage in a crowded engine room by reducing labour and obviating a great deal of mess and use of cleaning materials.

Filtration of lubricating oils and Diesel fuel oils and also the filtration of other liquids such as machine tool coolants, was an important matter which had received little attention and about which there was very little authoritative literature, and engineers were greatly indebted to the author for the facts presented in his paper. It was a matter of considerable importance which could well be the subject of prolonged investigation under working conditions to determine what was practical as compared with what was ideal or was necessary only to comply with arithmetical formula.

Mr. S. R. Joyce (Visitor): It had been noticed that no direct mention was made in the paper to the use of centrifuges in the lubricating circuit of engines during their brake test or "running-in". In view of the fact that at this stage in an engine's life, all the parts were moving together for the first time and that clearance limits were at their finest, it seemed opportune to point out that if the lubricating oil was centrifuged during this initial period, abrasive particles would be removed and the conditions mentioned as being contributory to wear, reduced or obviated.

Where this was done and centrifuges of sufficient capacity to deal with the whole of the oil circulating were employed, it was probable that only particles smaller than the clearance obtained would be left in the oil and these would have free passage round or through the bearings.

The firm with which the speaker was associated had supplied numbers of centrifuges in this connection, and it seemed that, although during such testing of engines, shock loads set up by repeated accelerations, etc., did occur, it was a fact that pistons, crankshaft bearings, etc., when dismantled, were found to have smooth glass-like surfaces free from score marks, this being contrary to the state of these surfaces if the engine was "run-in" without adequate centrifuging.

Did the author agree that providing the initial content of foreign matter in the oil was removed, wear and the occurrence of abrasive matter would be reduced, or did he consider that iron particles of such a size that they might cause serious wear, in spite of a concurrent loosening up and slight increase of clearances, were being continuously produced?

With regard to the views expressed under the heading "By-pass Oil Cleaners", the speaker observed that the introduction of adventitious grit, etc., by adding new oil to the engine sump was usually prevented by first passing such additions into the system *via* the centrifuge.

It had been found by experiments that centrifugal oil purifiers would remove carbon particles down to 0·003 of an inch in size. As carbon in such fine division had an "apparent" gravity behaving as though it were lighter than oil, it was fair to assume that still smaller particles of abrasive matter, which were usually heavier, *i.e.* iron and sand, would be centrifuged out, from which it followed that still larger and therefore heavier particles—which could do most damage—would also be removed. It would be interesting to know if any experience had been obtained concerning the wear of parts caused by the presence of fine carbon alone which might be left in the oil?

It was stated that with a centrifuge in a by-pass system, injurious particles may reach the bearings before having passed through the centrifuge. This, of course, was dependent on the rate and degree of contamination and the size of the centrifuge employed. Could the author give an example of a rate of contamination such that a large centrifuge capacity could not cope with it?

The very extensive preference for centrifuges in the marine field showed that they were most effective and that wear due to oil contamination was very low. The working of a suggested combination of a full flow filter and centrifuge was not fully appreciated by the speaker. Assuming the centrifuge handled only a proportion of the oil in a given period by the by-pass system, and the filter was handling the oil at the maximum circulating rate, would not the latter tend to collect particles of matter (and possibly choke) which it was intended should be removed by the centrifuge?

Considering filters as distinct from the centrifugal oil purifier, it was claimed for some types that all suspended solids were removed, including colloidal carbon. Would the author say if this was true and, if so, what relation filter area bore to rate of handling—which was admittedly low with filters—and the rate of contamination?

The information given regarding the removal of oil additives by some filters led one to the query: in what form did they exist in the oil? Were they in solution or colloidal dispersion, and would they be removed by centrifugal force as a semi-immiscible liquid much in the same way as castor and mineral oils used in aero engines could be separated?

Mr. G. A. Frampton (Visitor): Had the author any idea what was the limit particle size which would apply to large Diesel engines such as marine engines? His limit size for the small petrol engine, *viz.* 25 microns, was rather larger than one would have expected, and presumably it would be larger still for big marine engines which had greater bearing clearances. If an accurate value for limiting particle size could be arrived at for such engines it would be most useful in solving the vexed question of the proper operating capacity for a centrifugal oil purifier which governed the size of the purifier installation for a given lubricating oil system.

The functions of the full flow and by-pass filters were very definitely complementary. The former was intended to deal with every drop of oil immediately before it passed to the bearings and therefore had the limitation of requiring a very large capacity. It must ensure a flow of clean oil equal to the full circulation rate, whereas the function of the by-pass filter or centrifuge was to eliminate impurities from the system and thus enable the full flow filter to continue working for a reasonable period without cleaning. The logical thing was therefore to install the by-pass filter so that it was fed with oil from the dirtiest point in the system, and to feed the full flow filter from the cleanest possible point.

The speaker recommended the use of an additional pump to feed the centrifuge independently from the lowest point instead of taking its feed as a bleed-off from the delivery of the main circulating pump. This would give a great advantage in ensuring foreign matter removal for the expense of the extra pump.

He subscribed to the view expressed by Mr. Joyce that centrifuges did remove acidity, purely as a consequence of their ability to separate water from the oil. Thus, their usefulness in this connection was confined to the removal of water-soluble acidity. For turbines the acidity which developed in the lubricating oil was organic and only a part of it was soluble in water so that centrifuging could not eliminate all the acidity. In the case of Diesel engines, however, where considerable mineral acidity could develop in the lubricating oil, the centrifuge definitely could remove such acidity as it was water-soluble.

Mr. J. W. G. Brooker (Visitor): The author had shown that impurities in the oil had a considerable effect on wear and tear of engine parts, and this was irrespective of size and type of engine, but a previous speaker's reference to the "Queen Mary" was rather outside the scope of the paper, which dealt with internal-combustion engines.

On one point two of the critics cancelled each other out. One maintained that the clearances in large marine engines were so big that the figure of 0·001in. put forward by the author seemed very small, while the other speaker mentioned running tests that showed that clearances were likely to be less than 0·001in. even in large engines and that, therefore, impurities of this size in the oil would have a wear and tear effect.

Assuming an oil which contained no impurities, engines could still suffer wear and tear. Certain engines had been run and had worn out the bearings in a time so brief that it was patently not due to solid impurities in the oil, and it could be said that a great deal of wear occurred which could in no way be attributed to the presence of dirt in the oil. Traces of water could be equally destructive.

He would be glad to have the author's views on the effect water would have on the use of the filter mentioned in the paper.

The speaker considered that a great deal of wear and tear in both the cylinders and bearings of engines arose also under the conditions of starting. It was generally realized that when an engine stood for a few hours, many of the parts came into metallic contact, and in the movements of starting a good deal of wear was inevitable. In large engines, therefore, it was frequently the practice to float the bearings by oil under pressure before any movement was made. It was a fact that engines had been run for the equivalent of 2,000-hours with hardly any wear at all, but this was under constant load, speed and temperature and practically no starting and stopping. This was entirely apart from any question of filtering the oil or presence of water.

Coming to the subject of acidity, in a centrifuge much of the acid was removable by the use of hot water. All engineers would endorse the use of a feed of almost boiling water for this purpose, particularly for removing mineral acids. A trace of hydrochloric acid is almost inevitable in any engine used at sea, due to the presence of salt and this is a very destructive acid.

The table of figures on page 114 was rather interesting as showing that the use of a filter element could not be continued beyond a certain point without running considerable risk. The figures showed that the total solid contamination remained practically unchanged from 16,000 to 21,900 miles, indicating that very small particles of carbon, a product of incomplete combustion of the fuel, passed through the filter whether in a much used or in new condition.

In the case of ash and iron on the other hand, a change to a new element at 21,300 miles effected a marked improvement in the purity of the oil. The ash which was the most abrasive form of impurity was reduced by the greater efficiency of the new element to less than one-third the percentage.

Mr. W. S. Burn, M.Sc. (Member of Council): The subject matter of the paper was of great interest and of national importance at the present moment when not only was it necessary to reduce wear of existing and new engines of normal design, but because the nearer we were to the elimination of wear of bearing surfaces the nearer we were to the practical utilisation of high-speed engines with consequent vast reductions in man-hours for production, weight and size. Not only this, but it permitted of the greater use of a greater number of non-adjustable bearings which were simpler, much easier to make and free of the stress-susceptible bolt and nut; the radial aero engine with articulated connecting rods was a good illustration of the possibilities in this direction.

Within wide limits the I.C. engine was almost as happy thermally at high speeds as at low speeds and it was chiefly the increase in frequency of alternations of loading which affected fatigue susceptibilities and the increased number of momentary squeezings of the oil film and consequent possibilities of wear by abrasion which limited the wider application of high rotational engine speeds. Engine balance or dynamic loadings did not present fundamental difficulties and accumulated experience with high grade aero engine design was reducing fatigue effects and permitting substantial increases in the endurance range of alternations, and freedom from wear even permitted longer fatigue endurance. There were, of course, other problems of lubrication such as dilution, carbonisation and corrosion which required solution to the extent of ensuring long life of wearing parts without stuck rings or dirty ports, but in general the chief bugbear was common or garden abrasive wear.

Generally speaking the filtration arrangements of marine engine installations was good except immediately after engine installation or after overhauls, as fine marine filtration invariably worked on the by-pass system with centrifugal separators and streamline filters. Full flow filtration almost invariably was limited to relatively coarse wire mesh strainers or autoclean filters which removed the "small coal" and debris, but not the fine particles of a few thousandths of an inch in diameter. In some cases full flow magnetic filters were adequate in design and removed the dangerous iron scale particles which tended to increase with the increased use of steel plate fabricated engine parts, but the problem of the certain removal of small silicous particles remained, and it was here that the use of fabric filters recommended by the author had a definite appeal and utility and the writer would like to see the general adoption of this filter type *in addition* to all the present devices.

The use of fabric filters of the broad general type shown in Fig. 7 had been almost universally used on aero engines with success. What was now needed was a battery of the elements of such units arranged in one or more fabricated cases. The writer had used such a battery, with as many as eighteen Vokes elements, for several years with complete success, for fuel oil filtration in cargo liners. A single very large fabric element was used at one time by the writer for the lubricating oil cleaning of 4,000 b.h.p. units during shop tests, but choking caused its eventual collapse due to an inadequate relief valve, and from the experience gained the writer believed that the future of fabric filters was essentially with a battery of multiple elements with a common, well-designed, relief by-pass valve, *not* incorporated with the actual elements, and arranged in such a way that dirt from the filter could not get back into the system.

In the case of large marine installations the function of the fabric filter might well be chiefly for the first few hours of running after assembly or overhaul of the engines—a sort of permanent adaptation of the old Admiralty practice of putting mutton cloths round the gauze filters during initial trials—and the use of centrifuges, etc., depended on for normal running. The use of fabric filter elements necessitated renewal when choked, any attempt at cleaning the elements being strongly deprecated, and the supply of numerous spare elements would be a worth-while investment.

Whilst the marine engine could well learn from the aero engine the benefits of the fabric filter, it would seem that if the aero type engine could be fitted with large oil sumps and centrifugal separators and streamline filters, as in marine installations, the increased longevity would be most marked, as full flow centrifugals would be practical, in fact there was no reason why the use of high speed engines should not bring along literally perfect filtration as the residual particles could be always made less than the minimum oil film if it could be established that this was normally not less than 1/1,000in.

The author had endeavoured to prove that the oil film of automotive engines under working conditions was not less than 1/1,000in. and that therefore provided abrasive particles have dimensions less than this, virtually no abrasive action would result, and generally the speaker believed this to be confirmed in practice. Nevertheless the speaker would have liked Mr. Langley to have carried out tests on, say, four crank systems driven from a common shaft, but with separate crankcases and with separate lubrication supplies each with hard sharp abrasive particles, say carborundum, a different maximum size, say ½/1,000, 1/1,000, 1½/1,000 and 2/1,000in., and with controlled loadings from dynamic forces and the compression of a neutral gas equivalent to engine practice. This would prove the effect of particle size on engine wear and directly prove the truth of Mr. Langley's extremely interesting deductive conclusions of film thickness.

One would expect the film thickness to depend largely on the oil groove design, and in this respect the common circular groove was to be deprecated; the relation of oil grooves to bearing wear was well known, but it would appear that the reason for a substantial oil film was related more to the limitations of accommodation of particles than any metal to metal contacts. It would be interesting also to have Mr. Langley's idea of the minimum oil film in the normal operation of large direct-drive oil engine with a crankshaft diameter of 19in. and cylinder dimensions and weights as given on pages 142 and 143 of the speaker's paper, published in part 10 of Volume LIV of the TRANSACTIONS of the Institute of Marine Engineers. In this case the double-acting loading permitted particularly good bearing-oiling conditions, and one might expect a relatively thick minimum oil film. The author's comments on the effect of double action on oil films in general would be appreciated by those interested in this type of engine, as it was almost certain that larger particles were permissible in this type—which accounted for the reduced wear compared with single-acting types. The value of the paper to the Institute would be increased if Mr. Langley could give us his ideas of the design of a fabric filter to pass, say, 50 tons and 150 tons of oil per hour respectively, in which the latter case assumed piston cooling and bearing lubrication in one system such as in the case of a number of modern installations. It was suggested that incorporated with the filter relief valve there should be an indicator to show the degree of choking and the amount of oil by-passed.

Investigations by the writer into the lubrication of aero engines have not only established the effectiveness of the fabric filter in removing particles below 1/1,000in., but have indicated the striking effect of hardened steel gear type oil pumps in breaking up sand particles, especially in the case of the sump pump before the fabric filter. In the speaker's large double-acting engines a gear pump had been also used but with unhardened teeth, and the abrasion of the teeth was equally striking. Careful microscopic examination and measurement of the sand particle size in aero engines had established the efficiency of an aero engine sump pump (as well as other gearing) as a pulveriser of adventitious matter, and the speaker believed that such a pump, if specially designed, could well act as a further type of lubricating oil filter provided the larger particles were first removed by magnetic and wire mesh strainers. On the other hand it might be thought that the gear types of pumps were a menace in breaking up "hefty" sand particles of say 5 to 15 thousandths of an inch in diameter which could be easily removed by fabric filtration to something under one thousandth of an inch, which would be much more difficult.

The working thickness of the oil film would have a bearing on the matter and would suggest that in the case of most engines, and aero type engines in particular, the sump pump should *not* break up the particles *before* the fabric filter. Once the fabric filter had taken its full toll of particles there seemed much to recommend making the main lubricating oil pump of the gear type which would crush any remaining particles to well below the film size, but which eventually could be removed readily by a streamline type of filter working on a by-pass system.

Th speaker considered the paper to be most valuable in bringing the finer points of lubrication to our notice.

Mr. H. Crowther (Visitor): In offering the result of his investigations regarding engine oil filtration and its effect on wear in I.C. engines, the author had brought to light some exceedingly interesting data regarding the effects of dynamic loading, and the corresponding influence of wear in engines. It was unfortunate that the author had limited his investigations to small high-speed petrol engines, although undoubtedly much of the calculus presented could be applied to the compression-ignition engine.

The author appeared to have had complete disregard of oil pressure, which in the speaker's opinion was of great influence, not only as a source of supply, but in additional scavenging of impurities which could be considered supplementary to any of the design features that might be incorporated in the engine. Pressure also determined the type of filter one intended to use, and wear was minimized at low operating speeds if suitable measures were taken.

When studying engine filtration two features had to be borne in mind:

(*a*) The normal filtration recommended for engine in performance.

(*b*) The engine manufacturer's concern regarding filtration where large-scale construction of engines was concerned.

In the former case the speaker saw no reason to depart from the findings referred to in the paper, or that the type of filter could not be successfully operated. In the latter case, however, engines being submitted to test incorporated many solid contaminants accrued during phases of engine construction, and unless extraordinary care was taken by manufacturers, amounting to air conditioning and microscopic inspection of engine assemblies, these solid contaminants could not be avoided. Many engines could have seizures within a few moments from starting and the ordinary filtration system was therefore inadequate. The speaker's practical experience proved that the most satisfactory method was first to incorporate a magnetic filter to remove the ferreous particles, followed by a metallic filter, and subsequently the felt-element type.

The latter would be quite inadequate in trapping the various particles which in many cases could be up to 0·010in. in size, as with the pressure built up, these isolated particles would penetrate the felt pad and return to the lubrication passages. Pressure therefore played a very predominant part, and could be utilized to insure positive flow at low speeds by incorporating a pressure-reducing valve before the element filter so that half the maximum speed supply was diverted to crankcase. In the case of an engine developing 600 h.p. at 1,500 r.p.m., the maximum capacity of pump would be 1,400 gallons per hour at 75lb. sq. in. pressure, and by-pass valve set at 60lb. per sq. in., which would allow therefore for 800 gallons per hour to be circulated through filter, with remainder being returned to crankcase. At lower speeds the by-pass would not be operative as maximum pressure would not have been reached, assuring sufficient supply during the lower range of speeds.

The speaker felt that scavenging of impurities by crankshaft design must be taken into consideration where filtration was concerned, as many manufacturers still persisted in solid crankshafts, and probably under-estimated the advantages of the hollow-pin type. A hollow-pin crankshaft recently inspected after 2,400 hours performance showed crankpin wear of only 0·0015in., whereas a solid crankshaft in the same engine after 1,400 hours performance showed a corresponding wear of 0·004in.

It was to be hoped therefore that the author would not rest on his laurels, but would carry his investigations further, bearing these significant points in mind.

BY CORRESPONDENCE.

Mr. C. Lawrie (Member): The paper unfortunately covers principally the problems of filtration on small high-speed automotive engines which are considered to present essential differences in the practice of lubrication as compared to slow-speed marine Diesel engines.

So far as the actual filtration is concerned, marine practice incorporates an appreciable volume of oil being circulated over a fairly long interval of time. Standard practice is for the complete cycle of lubrication being attained in 3½/5 minutes. On automotive work, such as the paper embraces, the cycle of lubrication is possibly once or twice every minute. This has a marked and direct influence on the chemical change and resultant deposits derived from the oil itself which largely determine the suitability of the lubricant for further service.

Experience on marine work covering a number of years has shown that with the standard type of centrifuge of possibly 1½ tons throughput per hour, charges of crankcase oil can be maintained in splendid condition for very prolonged service, if certain facilities in addition to the fitting of the centrifuge are provided.

It has been found good practice to provide a cleaning tank of sufficient capacity to accommodate the whole lubricating oil charge fitted with a steam coil sufficient to maintain the oil at a temperature of 180/200° F. A steam jet or hot-water drip at the centrifuge is of inestimable value, for experience shows that with the introduction of low-pressure steam the effective weight of the carbon particle is increased by absorption or being surrounded by a water envelope or alternatively flocculation, these effects assisting precipitation at the centrifuge proper. The introduction of condensate, either in the form of a steam jet or water drip, when treating circulation oil also provides a very definite washing action which assists the precipitation of acidic bodies derived from the products of combustion and constantly changes the water forming the seal in the bowl proper, thereby avoiding the concentration of acidic bodies which corrode the centrifuge internal parts.

As a result of reviewing a number of such cases it is the exception rather than the rule to find any appreciable bearing wear in marine engines, while the condition of the lubricating oil on the completion of 15/18 years' service would indicate that the intrusion of abrasives is not one of prime importance.

The author gives some very interesting analyses on page 115. It is to be noted that the fuel dilution, as one would expect, is not constant. A possible explanation of this would be the amount of high-speed running over the period covered. No reference is made to the soluble material which is miscible in the oil itself, and therefore there is no indication as to the true value of the lubricant on the completion of 21,900 miles.

The analysis of the deposit confirms general experience in this direction, but no mention is made of the volatile matter soluble in petroleum, ether or chloroform. It would also be of interest to know the moisture fraction present.

In conclusion, the views of the author are sought on the following points:—

Has any observation been noted on the rate of chokage between petrol engines and Diesel engines?

The water fraction allowable before filtering efficiency is impaired?

Mr. T. L. Kendall (Member): The author has quite obviously gone very closely into his subject, and has given some extremely interesting facts regarding thickness of oil film under varying load conditions, and the relation of size of foreign abrasive particles to this film.

The necessity for very thorough filtration, and for ensuring that the degree of filtration takes into account all the foreign matter likely to cause wear, is made amply clear by the author. The writer, from his own experience, can emphatically endorse the author's recommendations in this respect, but is inclined to disagree with him on the matter of the full flow type of filter. This type of filter may be practicable on a motor-car engine such as the one upon which the author's experiments were carried out, but on any other type of engine, industrial or marine, such a filter has to be so coarse as to be only a first line of defence; if fine enough for complete filtration, its size puts it out of bounds, or alternatively, the period between cleaning is so short as to become impracticable. In this respect it must be borne in mind a motor-car engine covering 21,000 miles, as the one in question did, has only been operating for 700 hours, and this is spread over quite a considerable time, whereas a normal engine in every-day industrial or marine service, can run the same period in one to two months.

Reverting to the size of a full flow filter, the author gives for a filtration fineness of 0·001in., approximately 50 square inches of filtering medium for 100 gallons per hour, but for a relatively small marine engine of, say, 300 b.h.p. this would involve a filter of about 150 to 200 sq. inches through area multiplied by the safety factor given by the author of 4 to 10, so that the size is approaching the unwieldy. Taking these factors into account, the writer is in favour of the combination of a coarse full flow filter of the edge type, cleanable while running, and an absorption filter of the by-pass type. Granted, with this combination there is a 10 to 1 chance, to take the author's figure, of any abrasive matter introduced with new oil causing bearing wear before it is removed by the by-pass filter, but this is a risk which is, the writer thinks, justifiable.

The author states that colour of lubricating oil is no indication of its service ability, but in this the writer thinks he is only partly right, as the man responsible for running an engine has not the facilities for measuring the wear producing particles, and his only guide is the colour of the oil to show him whether the filter is functioning or not. This is a point which should be strongly stressed, particularly in view of the broadcast statements of certain authorities that colloidal carbon, which is principally responsible for discolouring oil, is of no detriment, and should not be removed. This in itself is quite correct, but the discoloration can be a cloak for much more damaging constituents.

Mr. H. Mackegg (Member): With regard to bearing surface wear, the practical life of an engine commences with the first revolution of its crank shaft, irrespective of whether the engine is "motored" or operative. If at that point in its life the bearing surfaces are not clean and correctly lubricated, and the lubricating oil system free from all solid impurities, wear will take place forthwith.

Thus, it is incumbent upon engine manufacturers to see that their production methods are efficient, and that their test bed lubricating oil cleaning systems operate effectively, so that during shop tests the lubricating oil is absolutely clean.

Many manufacturers realize this point, as is evidenced by the number of centrifugal separators which have been installed on test beds dealing with all types of internal-combustion engines.

When an engine is subsequently installed and the lubricating oil system and the lubricating oil storage plant connected up, the whole of the lubricating oil should be passed through the oil cleaners several times, so that the whole system is cleared of impurities which collect in the interconnecting pipework, tanks, etc.—*before the engine is started up.*

In dealing with centrifugal oil purifiers, the author refers to the ability of the centrifuge to remove water from and reduce the acidity of lubricating oil, and the writer would enlarge upon that point. In the majority of instances it is advantageous to introduce water washing during purification, and this system can be adopted by introducing an independent stream of fresh water at 180-200° F.; this water is fed to the purifier together with the oil and in its passage to the purifier bowl the water is intermediately mixed with the oil prior to complete separation. The object of this procedure is as follows:—

(1) To wash out of the oil the water soluble oxidation products, which if left long in the oil may oxidize further and form oil soluble bodies which could not be removed.

(2) To cause those sludges which are insoluble in water and which give rise to oxidation products to form solids which would be removed in the purifier.

(3) To replace frequently the water seal in the bowl and thus prevent the accumulation of acid materials in it. This prevents the corrosion of the bowl parts by these acid materials.

(4) To effect the coagulation of the finely divided particles, thereby assisting their removal during centrifuging.

Some oxidation products formed in the deterioration of oil are not soluble in water and do not form solids. These are not removed and appear to do no harm. They tend, however, to raise the neutralization number of the oil, but being insoluble in water they do not cause corrosion. The wash water should preferably be condensate which has the best solvent properties.

It should be clearly understood that wash water should be used from the commencement with a new engine and with new oil, otherwise where oil had become wholly contaminated most of the oxidation products would have already become soluble in oil and the amount removed by water washing would be low.

It is an advantage to have a permanent wash water connection to a centrifugal oil purifier as it is then a simple matter to water seal the bowl when starting, and a further point of very definite importance is that the use of wash water has the effect of continuously replacing the water seal. If no wash water is used and if the water seal is comparatively stagnant (except for any water normally separated from the oil) the water seal becomes highly contaminated by the absorption of acids picked up from the oil in its passage through the separator, thus serious corrosion troubles arise.

Mr. J. A. Jaffrey, M.Sc. (Member): There was no doubt that wear occurred due to interruption of the oil film from one cause or another. The author had stated that the minimum oil film thickness was seldom if ever less than 0·001in. during normal running provided there was no misalignment or excessive wear, and the argument had been based upon the petrol engine. Unfortunately the results stated could not be applied to normal marine I.C. engines, but whatever the thickness of oil film the basic argument regarding wear was still applicable.

It was not physically possible to manufacture the journals, crankpins, etc., of a crankshaft, particularly for the larger sizes of marine engines, dead in line and round to 0·001in., apart from any further variations which would occur in service due to temperature and distortion. Consequently the minimum oil film must be considerably less than 0·001in. thick, and of a limited area varying in position throughout one revolution. In this connection it was possible, for example, to take measurements by micrometer of a journal about 10in. in diameter, which appeared to indicate the maximum error was 0·0005in. on the diameter, and it could be demonstrated that the journal was actually as much as 0·002in. out of round. This phenomenon occurred when a three point 120° steady was used during manufacture. It appeared therefore that the lubricating oil for marine engines required better filtration than for petrol engines.

Wear in I.C. engines could be divided into three main classes:—

(*a*) crankshaft and bearings,

(*b*) cylinders,
(*c*) other parts.

Relatively speaking, parts under (*b*) and (*c*) were easily changed, but the interchange of a crankshaft was a major operation, and so it was considered that advantage would accrue from any steps that would prevent wear. Wear occurred generally in several stages:—

(1) when the engine was new or had been dismantled and was first started up,
(2) when the engine was started normally,
(3) normal running wear.

Unfortunately (1) was the maximum in a large number of cases, due to foreign matter in the engine. The writer saw a new engine which for four hours had been flushed through with hot oil by a special oil pump and very large oil filter before starting up. Every big end and small end was scored at the end of the trials. It was common practice to provide a "wash away" in the bearing liners on the joint centre line, and foreign matter collected at the bottom of the "wash away", and all the flushing possible would not move it. Depending upon the arrangement of the oil pipes, when an engine was stopped there was generally a gravity head of oil on the bearings. As a result all the foreign matter in the oil gradually settled either at a bend in the pipe or in the "wash away" of the bearings.

During normal running crank pins wore more than the journals. Moreover, the crankpins always showed the maximum wear at an axis which made a small angle with that through top and bottom dead centres. The position of the oil hole in the crankpin was almost invariably at the top, or slightly off centre on top dead centre and at the bottom on bottom dead centre, so as to avoid any stoppage or accumulation of foreign matter in the oil ways. From the lubrication point of view it could not be said that the positions were the best.

It appeared therefore than when the engine was new or overhauled it was of absolute importance thoroughly to clean the bearings before the engine was run, and if necessary disconnect the oil supply pipes from the bearings and thoroughly flush them out. The second point was that the oil pipes to the bearings should be so arranged as to give no head of oil. Moreover the suction side of the oil pump should be in such a position that any settlement that took place in the sump was not sucked up into the pump on starting.

Particularly in the larger sizes of marine engines and where a separately driven lubricating oil pump was used, it was possible that the lubricating oil hole in the crankpin might with safety be placed in the best lubrication position, which was off centre on the inside of the pin in bottom dead centre. Large crankshafts were often provided with holes drilled through the centre of the pin, and if the holes were provided with removable plugs then on account of centrifugal force the particles of foreign matter would be returned in this large centre hole and would not have to pass to the big end. The centre hole could be cleaned out periodically.

In the larger marine engines quite a large proportion of the oil was fed to parts other than the crankshaft, and it would seem worth while to provide a special auxiliary filter for the oil which actually passed to the crankshaft bearings.

In connection with magnetic filters, the author was obviously referring to foreign particles of metal resulting from wear, but it was well to note that on large marine engines it was not always possible to remove all the scale and metallic particles from castings, and with welded structures it was very possible that welding splash as well as mill scale from the rolled plates might become detached in service.

The Author's Reply to the Discussion.

General Remarks.

The discussion was both interesting and gratifying to the author as it disclosed some points of view which, together with the following deliberations thereon, should add to the usefulness and scope of the paper. That some sections of the paper were not sufficiently explicit is now evident. For example, on page 110 it is stated that: "For different bearing diameters, the film thickness would be changed in proportion". This was intended to mean that, for the same r/c and ZN/P values, the film thicknesses obtained from the diagrams would be multiplied by the ratio of the actual bearing diameter to the bearing diameter assumed for the diagrams. It appears that this point was not sufficiently emphasized, and it will be referred to in the replies which follow.

Some doubt was expressed during the discussion regarding the agreement between the oil film thicknesses obtained in practice and those indicated by the calculations. It will be agreed that the accuracy of the results obtained by a properly conducted mathematical analysis depends only upon the accuracy and completeness of the data on which the analysis is based. The author believes that the data used were substantially accurate. Under some working conditions the data would not always be complete. For example, the presence of water in the oil might cause discontinuity in the oil film which would certainly lead to greater eccentricities of the shaft in its bearing as will be understood upon reference to Fig. 1. In the example used for the calculations it was assumed that the impurities in the oil were insufficient to affect the results, and some experimental evidence in support of the general conclusions will be referred to in the replies to the discussion.

The effect of manufacturing inaccuracies has been considered, and this matter is also referred to in the following replies.

As pointed out by the Chairman, the author omitted to explain the meaning of the symbol "x" shown on Fig. 1. This symbol represents the eccentricity of the shaft in its bearing. It would have been better to have used the symbol "e" for this figure because "x" is used in a different sense in the mathematical appendix.

Mr. E. P. Paxman, M.A. Replying to Mr. Paxman's very interesting contribution to the discussion, it was not the author's intention to imply that the causes of wear referred to in the paper were the only ones encountered. Undoubtedly other causes of wear exist but these were considered to be rather outside the scope of the paper. Imperfect oil films could have a very important effect. Fig. 1 indicates that the result of an imperfect oil film may be to increase the eccentricity of the shaft as much as threefold under certain conditions.

The statement on page 110 referred to in the preceding general remarks was of such brevity that its significance might easily be overlooked. It was intended to mean that the minimum oil film thicknesses would, for the same r/c and ZN/P values, be proportional to the bearing diameter. The author also intended to convey the recommendation that the filtration fineness of a full flow filter should, if possible, be somewhat less than the minimum oil film thickness obtainable under normal working conditions, and he hopes that the method given in the paper for determining the oil film thicknesses may assist in deciding the filtration fineness desirable for any particular engine, whether large or small. Unfortunately, in spite of the author's efforts to simplify the method in every way possible, it still involves somewhat laborious calculations unless the r/c and ZN/P values agree with those used in the example.

The use of a by-pass valve in a full flow filter is desirable, if not always essential, but rather too much importance is sometimes attached to the operation of this valve during cold starting. If the valve is closed during normal running and the filter is doing its job properly, then the oil should be sufficiently clean when the engine is stopped and there should be very little risk in the temporary opening of the by-pass valve during a subsequent cold starting of the engine.

In testing new engines this argument does not apply, of course, and the opening of the by-pass valve may be dangerous in this case. The graph provided by the speaker is most interesting in this respect. At the same time it clearly indicates the great advantage to be gained by efficiently cleaning the lubricating oil. The somewhat elaborate method described for cleaning oil on new engine test beds was amply justified as evinced by the graph referred to.

With reference to the effect of water on wear, it has been shown and confirmed by tests using sugar syrup as a lubricant that, provided hydrodynamic conditions are fulfilled, the nature of the lubricant has no effect on the action of the oil film, which action only depends upon the variation of ZN/P. This apparently is only true provided the lubricant is homogenous. Water in oil may, indeed, cause discontinuity in the oil film and this may, in turn, have an important effect on wear. It is a subject, however, which the author has had no opportunity of investigating.

Regarding the speaker's enquiry relating to the optimum size of an oil sump, apart from the question of cost and space, it would appear that the larger the sump the better. A larger volume of oil in circulation will, of course, remain cleaner and cooler, so that less oil contamination is carried to the bearings and more heat can be removed therefrom. Furthermore, the oil will need replacement less frequently. From this point of view, it would appear that the optimum size can only be arrived at by a judicious comparison of these advantages with cost and space considerations.

Dr. E. Giffen. The author is glad to have Dr. Giffen's general agreement that some such treatment as that provided in the paper for determining the working oil film thicknesses is needed. The speaker appeared to have a serious doubt, however, regarding the possible agreement between working conditions and those assumed for the theory, particularly because of the flexibility of the bearing housing and of the shaft itself. As far as the big ends are concerned, shaft and bearing distortion can have no appreciable effect on the oil film unless they cause an angular displacement of the crankpin axis. Now the angular displacements for the crankshaft shown in Fig. 8 have been approximately determined. For the crankpins the maximum angular displacement is of the order of 9×10^{-5} radians and for the outer main bearings roughly about three times this figure, assuming the main bearings to be rigid. These angular displacements would only entail a reduction in the oil film thicknesses of from 0·00009 to ·00027in. It would certainly be difficult to determine the main bearing deflection but, when it is considered that the maximum deflection of the shaft between the bearings is only of the order of 0·0005in., the author feels that the deflection of the bearings themselves must be very small indeed. Some experimental evidence of the author's figures was obtained in the following way. Again referring to Fig. 8, it is a comparatively simple matter to determine the maximum particle size passing to the right of the bearing A for example. Taking into account the pump pressure, the pressure due to centrifugal force and the end leakage at the bearings, the maximum sizes of the particles which would pass to the right of this bearing were calculated. These were found to be as follows:—

Bearing Clearance (radial).	*Max. size of particle passing to right half of bearing at:*							
	500 r.p.m.		1,000 r.p.m.		2,000 r.p.m.		3,000 r.p.m.	
Inch.	Sand.	Iron.	Sand.	Iron.	Sand.	Iron.	Sand.	Iron.
·002	·001	·0005	·0008	·0004	·0007	·0003	·0006	·0003
·0025	·0014	·0007	·0011	·0006	·0009	·0005	·0008	·0004
·003	·0018	·0009	·0014	·0007	·0011	·0006	·001	·0005

It is not claimed that these figures are precise, but in view of the tenableness of the assumptions made in their calculation, the author believes they are substantially correct.

It will be understood that any size of particle which can pass between the bearing surfaces can move to the left of the bearing considered. Now the condition of the big end bearing as shown in Fig. 9 indicates, in the author's view, that some of the particles passing to the left of the bearing were appreciably larger than the oil film thickness, but that none of those passing to the right could have been appreciably larger. Some passing to the right could have been somewhat larger than 0·001in., which appears to indicate that the minimum oil film in the bearing could not have been less than 0·001in. during the major part of its service.

Mr. A. Beale. Undoubtedly the speaker's witty comments were appreciated as much by the whole assembly as by the author.

The author is substantially in agreement with the speaker's points (1) to (4), but at the moment has no experimental evidence in support or otherwise of the statement (5). Regarding point (6), the author is indebted to the speaker for drawing attention to the paper by Messrs. Le Mesurier & Stansfield which, in fact, supports his (the author's) views. Thus on page 141 will be found the following statements: "This petrol suspended dust may, however, be too fine to span the oil film between the piston rings and the cylinder walls and, if so, will be harmless". Again, on the same page: "This fact supports the view that abrasive wear on cylinder liners occurs when the particles of abrasive span and rupture the oil film between cylinder walls and piston rings". On page 144 will be found the statement: "Abrasion is probably the most serious factor and can be attributed in part to the fuel storage conditions and in part to the inlet air".

With reference to the speaker's statement that the reduction of acidity by a centrifuge is contrary to his experience, it may be stated that there is experimental evidence in support of the author's statement. Acidity can be reduced by a centrifuge without chemical treatment. Water soluble acids are removed by allowing hot water to pass with the oil into the centrifuge. During the early stages of formation of organic acids, that is to say before the acidic compounds acquire a long chain formation, they are water soluble, and that is one reason for recommending the continuous operation of the centrifuge. When the acids have reached the water insoluble stage they cannot be removed without chemical means. As indicated by other speakers, the inorganic acids are the most harmful and these are, of course, water soluble and easily removed.

"Oiliness" has been the subject of much controversy probably because it had never, as far as the author is aware, been satisfactorily defined. "Olacity" is intended to have a definite meaning, such as that given in the paper. Tests, which are fairly well advanced, but not yet complete, appear to indicate that, not only may a clear definition be provided for this property but, what is equally important, a well defined unit of Olacity may be formulated.

Mr. T. R. Stuart. It is true that we have to compromise in the matter of lubricating oil filtration as in many other matters connected with engineering. However, it is important that, in doing so, we do not allow injurious substances to reach the bearings.

The author does not agree that it is so very difficult to incorporate full flow filters or strainers of sufficient fineness and capacity for large engines provided the fineness is chosen to suit the normal working oil film thickness. That a satisfactory full flow filter can be found for a small engine has been shown in connection with the report on the M.G. car test referred to in the paper. The filter in this case was not inconveniently large and if, for a large engine, a satisfactory full flow filter only occupied a proportionally larger space, there could be no objection to it on the score of bulk. Now the h.p. of an engine and the capacity of a full flow filter or strainer of given fineness are both approximately proportional to the square of the linear dimensions. Thus if QZ/f is proportional to the h.p., Q being the rate of oil flow to the bearings, Z its absolute viscosity and f the filtration fineness of the filter, then the linear size of the filter would be substantially proportional to the linear size of the engine. Referring to the notes in the author's general remarks, it will be seen that "f" may be increased for larger engines and if advantage is taken of this there should not be much difficulty in keeping full flow filters for large engines down to reasonable sizes.

Regarding the effect of particle hardness on wear, the author is in agreement that the wear is not directly proportional to hardness, but he finds it difficult to believe that cotton fibres in oil cause wear unless they first pick up abrasive material. The effect of cotton packing is not comparable.

Regarding the question of the opening of a by-pass valve on a filter when the oil was cold, it must be remembered that the viscosity of cold oil may be several hundred times as great as the viscosity under normal running conditions, and the pressure drop through the filter, if it were compelled to pass the same quantity of oil, would be increased in approximately the same proportion. The opening of the by-pass valve during cold starting is not so important as some would have us believe. The oil usually gets cold after it has been made hot, that is to say, after the filter has had ample opportunity to remove all abrasive matter from the oil, so that there is very little risk in the temporary opening of the by-pass valve during cold starting.

Mr. S. R. Joyce. The author is glad Mr. Joyce raised the question of oil cleaning during running in. This is, of course, a very important matter. Undoubtedly a centrifuge constructed to deal with the full flow, has much to recommend it. A rather large centrifuge would be required in some cases and the author has been informed that 1,200 gals. per hour is about the maximum capacity available at the present time so that, in many cases, more than one such machine would be required to deal with the full flow. Full flow filters in combination with a by-pass centrifuge have been used quite successfully for this duty, and such a combination may be cheaper.

Replying to the enquiry regarding wear by carbon, it is believed that fine carbon alone does not cause wear.

Regarding the use of a centrifuge in a by-pass system, the author believes it will be conceded that there will always be a risk of injurious matter reaching the bearings, whatever the size of the centrifuge, unless a full flow filter is also fitted. On the other hand, a centrifuge can always be chosen to suit the rate of contamination when operating in a by-pass capacity. The speaker's suggestion that a full flow filter would tend to collect particles which it was intended should be removed by the centrifuge (and possibly choke) seems to the author to be an implication that the centrifuge is not doing its job because the full flow filter would only collect matter which would otherwise reach the bearings.

The removal of colloidal carbon by a filter depends, according to one authority, on whether the filtering material is hydrophobic or hydrophilic. Colloidal carbon is hydrophobic and is not removed by hydrophobic filtering media such as felt for example, but it can be removed by hydrophilic material such as cotton. Another investigator claims that its removal also depends upon the pH value of the filtering medium, but his experiments were conducted with water suspensions and may not apply to oil suspensions.

Regarding the removal of additives, the author has been informed that experiments have already been conducted which indicate that these are not removed by a centrifuge.

Mr. C. A. Frampton. Replying to the speaker's enquiry regarding the limit particle size for large Diesel engines, the author believes that this can be successfully determined. As indicated in the author's general remarks, for the same r/c and ZN/P values, the minimum oil film thickness would be proportional to the bearing diameter. The calculation of the deflection curves for any particular case is admittedly somewhat laborious and the author has not had the opportunity of applying his method of calculation to more than the one example referred to in the paper. There does not appear to be any short cut to the results unless the ratios above referred to happen to be the same.

The author is in agreement with the speaker's statement on the complementary nature of full flow and by-pass filters, and with his recommendations relating to the feed to a centrifuge.

Mr. J. W. C. Brooker. The speaker first cleared some of the fog and then introduced a little more haze. The latter was well worth removing.

The author is doubtful whether it will ever be possible to have in a working engine an oil "which contains no impurities". If such perfection could be obtained, however, the author believes that wear and tear of bearings, properly designed, made and run in, would be negligible. The cylinder is a source of trouble and may probably always remain so. In some test bed experiments it was noticed that, if an engine was started up just before lunch time and stopped for lunch, the engine was very difficult to turn over after the luncheon interval. It was found that, in that brief interval, sufficient cylinder corrosion had occurred to account for the trouble. Corrosion appears to be a very controversial subject and is rather outside the scope of the paper. Apart from corrosion, however, the author fails to see any reason for appreciable wear for the hypothetical case considered.

The speaker considered that traces of water could be destructive. Now it has been shown by some investigators that, so long as hydrodynamic conditions are fulfilled, neither the lubricant film nor the coefficient of friction is affected by the nature of the lubricant, provided the value of ZN/P is unchanged. In some tests sugar syrup of known viscosity was used in place of oil, and the coefficient of friction curves were found to be the same as for oil of the same viscosity. It is difficult to understand, therefore, how traces of water as such, excepting in so far as they may reduce the effective viscosity or cause discontinuity of the oil film, can have any appreciable effect on wear. Water in combination with some acid formations will, of course, be responsible for corrosion, and emulsification can readily occur with water in the presence of soot and/or lead salts.

Coming now to the question of the effect of water on a full flow filter of the type mentioned in the paper. Felt, of which the filtering medium is composed, readily absorbs oil but tends to repel water. The filter will, to an extent depending upon the conditions, separate water from oil. Such filters did, in fact, successfully remove water from fuel oil, but in this case the rate of flow of oil through the felt was comparatively small. If the water was allowed to accumulate to the extent that it remained in contact with the felt then it could be carried through, especially if the oil flow was fairly rapid as in the case of full flow lubricant filtration. Unfortunately water under certain conditions, especially in winter, assists in the formation of sludge which is one of the worst enemies of the full flow filter. Ample evidence is to be found in support of the speaker's view that frequent starting and stopping is conducive to wear.

The author is in full agreement with the other points raised by the speaker.

Mr. W. S. Burn, M.Sc. Mr. Burn's contribution is both interesting and thought stimulating. His approbatory remarks are much appreciated by the author.

Dealing first with the question of engine speed, high speed in itself is not necessarily conducive to bearing wear. For the same temperature and loading, a higher speed may be accompanied by a lower rate of wear. Within certain limits the peak loading on the bearings is reduced by an increase in speed because the piston inertia forces, which are, of course, proportional to the square of the speed, are opposed to the gas pressure forces during the first half of the working stroke. On the other hand, however, the temperature rise which usually accompanies an increase in speed may more or less counterbalance any advantage derived from the reduction in peak loading as far as bearing wear is concerned. With badly contaminated oil in a forced feed system higher speed is usually accompanied by a higher rate of wear because of the higher rate of feed of injurious material to the bearings.

The author is in full agreement with the writer's recommendation to fit full flow fabric type filters in addition to other existing oil cleaning devices. For the larger marine engines, as indicated in the replies to Mr. Stuart and Mr. Kendall, such fine filtration as that provided by the filter shown in Fig. 7 is not essential. If the filtration fineness of the full flow filter is made to suit the normal working oil film thickness for the engine for which it is intended, the author believes the resulting filter will not be considered unduly large, and there is no doubt that fabric filter manufacturers will most readily co-operate in producing the necessary range of filtration fineness as soon as the demand is established. One of the difficulties in the past has been due to an endeavour to use on large engines the fine fabric filtering media required for small engines because the demand for the latter has been so much greater. The author is in full agreement with the suggestion that the relief valve should be so positioned that dirt from the filter could not return to the oil system.

Regarding the cleaning of fabric filters, a satisfactory device has been developed for cleaning fabric filtering elements, which device eliminates any danger of transferring dirt from the outside to the inside of the filter. Here again, however, the demand has not justified the production of this device. Until such a device could be used, the author is in agreement with the writer's deprecation of any attempt at cleaning these elements, and agrees that a store of spare elements would be a worth while investment.

The author would very much like to have the opportunity of carrying out tests such as those suggested by Mr. Burn. There is no doubt that these would be of great value but, as far as the author is concerned, the chance of starting such tests appears to be remote at the present time.

Regarding oil grooves, it is true that these may seriously affect the working oil film thickness, especially if they are badly positioned. Grooving in high speed bearings is usually unnecessary apart from the circumferential grooving required for force feed systems. For low speed bearings, grooving should be kept away from the high pressure areas.

With reference to the large oil engine referred to in Mr. Burn's paper published in Vol. LIV of the TRANSACTIONS of The Institute of Marine Engineers, the author will endeavour to find an opportunity of investigating the points suggested and will communicate the results at a later date. In such an engine there is no doubt that the normal working oil film thickness in the bearings would be very much larger than in the particular cases discussed in the paper. The dependence of oil film thicknesses on the size of the bearing has been referred to in the author's general remarks and in replies to other contributors, and it would be expected that a further enlargement of the minimum film thickness would be obtained with the double acting engine.

With reference to the design of fabric filters to pass respectively 50 tons and 150 tons of oil per hour, the construction of such filters would, of course, depend upon the filtration fineness required and this in turn depends upon the design of the engine. From rough estimations, it appears that an engine requiring 50 tons per hour should not need a filtration fineness less than 0·0035in. and, while the author has no test figures for fabric media of this particular fineness, it is estimated that, for a fabric filter of 0·0035in. fineness, an area of 4,000 sq. in. would be ample. Messrs. Tecalemit Ltd. make star shaped elements having 1,040 sq. in. effective area, and these are 6in. diameter by 13·6in. long, but the standard element is too fine for the case under consideration. The author would suggest two banks of four such elements, but of the required fineness, with a change over valve constructed so that either set could be put into service while the other set was being cleaned or renewed. Each set of four would, of course, operate in parallel. For 150 tons per hour, three times the above area would be required if the same fineness was used, but for an engine requiring a flow of 150 tons per hour of oil it would appear that a filtration fineness of ·006in. should be sufficient and, using this figure, an effective filtering area of 7,000 sq. in. would be desirable. An element of a similar construction to that previously referred to and having an area of 1,750 sq. in. would be about 7½in. diameter and 18in. long. Two sets of four of these could be arranged as already described. In each case one or more relief valves should be included. If a differential pressure gauge is fitted to indicate the pressure drop across the filter, the readings obtained from this under normal working conditions would be a good indication of the degree of choking of the filter, and also of the amount of oil being by-passed if the relief valve is suitably designed and calibrated. In fact, a chart could be provided showing the relationship between the pressure drop reading and the amount being by-passed, provided the readings were always taken under substantially similar running conditions.

Mr. Burn's account of his experience with fabric filters and gear pumps on aero engines and on his large double acting engines is very interesting. The author does not think, however, that a gear pump of the ordinary type could act as a reliable pulveriser because comparatively large particles could pass through the pump unharmed. However, the idea of pulverisation suggested by Mr. Burn, if it can be effected in a simple and efficient manner, is one which deserves careful investigation.

Mr. H. Crowther. The author is glad to have the views of the speaker who had, undoubtedly, had much experience in the matter of filtration and its effects on engine wear.

The author has already dealt with the effect of engine size in replying to a previous speaker.

As pointed out by the speaker, the effect of oil pressure had not been dealt with in the paper. It will be understood that the pressure drop through the filter will be dependent upon the rate of oil flow through the filter but not upon the actual pressure of the oil unless the oil pressure causes a compression of the filtering medium and the deposits thereon. While it is necessary to consider the oil pressure for the design of the casing, as far as the oil is concerned it is only its rate of flow and viscosity which determines the design of the filtering element.

While the speaker was no doubt correct in his statement regarding the scavenging of impurities due to oil pressure, the author is not sure that this would reduce wear. For a given degree of contamination, the author would expect a greater oil pressure to increase the rate of feed of contamination to the bearings by the same amount as it increased the scavenging. Furthermore, the greater speed of flow through the filter might impair the degree of filtration.

The author is in substantial agreement with the speaker's remarks regarding the testing of new engines. This subject has been discussed by, and in replies to, previous speakers, but it is very useful to know that the speaker has found, by practical experience, that the combination of a magnetic filter, a metallic filter and a felt filter has proved to be the most satisfactory for this work. The author is, however, surprised to hear that particles 0·01in. in size could penetrate the felt type filter—at any rate the felt type with which he has been acquainted. As already pointed out, the pressure drop through the filter is almost independent of the actual oil pressure, but it is the pressure drop which determines the force which could cause penetration. With a pressure drop through the filtering element of as much as 10lb./sq. inch, the force to cause penetration by a 0·01in. particle would only be about 0·012oz.

The interesting figures given by the speaker regarding the relative wear of hollow pin and solid crankshafts are well worth noting. The author trusts that there will be an opportunity of carrying his investigations further as suggested.

Mr. C. Lawrie. The subjects of the speaker's earlier remarks have been mostly dealt with in the replies to previous speakers. The author agrees that the centrifuge, with its ability to remove carbon and some forms of acidity in addition to abrasive materials, has much to recommend it, and the experience of the speaker has been noted with interest. When the speaker refers to lubricating oil giving such good service for 15/18 years he is, no doubt, referring to the lubricating oil of a turbine engine. With such an engine the speaker is probably right in saying that the "intrusion of abrasives is not one of prime importance" because, in this case, the working oil film thicknesses are so much greater than in the cases considered in the paper.

With reference to the analysis figures on page 115, the fluctuation of the fuel dilution is not so much a matter of high speed running as of varying climatic conditions. The samples which show low fuel dilution were taken during the summer months and those of high dilution during the winter. No doubt, since the average running conditions were fairly constant, the fuel dilution would also have been approximately constant if the atmospheric temperature had been uniform. Moisture was consistently low, and by a trace is meant a figure not exceeding 0·3 per cent. When "nil" was recorded, the figure was certainly less than this. Regarding impurities soluble in the oil, these were not estimated, but the oil showed so little evidence of oxidation that any appreciable proportion of oxidised impurity in solution was not anticipated. The oil used, being a mixture of Motorine "C" and Motorine "D" de Luxe, contained a considerable proportion of fatty oil which, as will be understood, is a very good detergent from this point of view.

With reference to the enquiry regarding the relative rate of chokage of filters on petrol and Diesel engines, this, of course, is not constant because it depends so much upon the condition and running of the engines and upon the type of oil used. Usually, however, the rate of chokage is somewhat higher on Diesel engines than on petrol engines, and may be as much as twice as rapid in bad cases. This does not mean, however, that the filter need be larger in direct proportion because a larger filter not only receives a thinner deposit in a given time but also provides a slower rate of flow through the filter. For Diesel engines it is desirable that the filter should have 50 per cent. greater area than for a petrol engine having the same rate of oil flow but, as stated elsewhere, the greater the area which can be accommodated the better.

Regarding the question of the water fraction allowable before the filtering efficiency is impaired, the author has not sufficient reliable data on which to base a considered opinion. Many samples of used oils and filters have been tested, but the author has not observed any relationship between the water content and the filtering efficiency. Some tests have shown that water increases the resistance to flow through felt to some extent and, if the water leads to the formation of sludge this will, of course, accelerate the choking of the filter.

Mr. T. L. Kendall. The author is glad to have the writer's approbation of the recommendations set out in the paper. If in the paper space had permitted, it may have been possible to deal with the relationship between the size of the engine and that of the filter. While emphasizing the fact that for small engines the filtration should be fine, the author is afraid that he may have unwittingly obscured the fact that larger engines did not require such fine filtration. As shown in replying to Mr. Stuart, the relationship between the linear size of the engine and that of the filter can be regarded as approximately proportional to the expression QZ/f, where Q is the rate of oil flow to the bearings, Z is the absolute viscosity and f the filtration fineness which, according to the author's view should, if possible, be somewhat less than the normal working minimum oil film thickness in the bearings. The casing of the full flow filter described in the paper is approximately 5¼in. diameter by 14in. long. "f" for this filter is 0·001in., and its area is 410 sq. in. This could be used for a Diesel engine of up to 250 b.h.p. Taking into account the protection provided, the author does not think that a filter of this size would be considered unduly large for such an engine, and if "f" could safely be increased the filter could be used for a somewhat larger engine. The danger of dust and grit carried by the atmosphere is less for the larger engines in which the working oil film thicknesses are larger and this, no doubt, is the reason why large engines have been worked successfully with coarse full flow filters. The problem is to determine the maximum safe value for "f", and the author believes that the method given in the paper will assist in this determination. Having decided upon this factor, a suitable size of full flow filter can be determined.

The author agrees that the combination of a full flow filter and a suitable type of by-pass cleaning unit is desirable. While in many cases the colour of the oil may be taken as a useful indicator of the condition of a by-pass cleaner, the author considers that the colour is a most unreliable indicator of the serviceableness of the oil. However, since carbon under certain conditions is conducive to the formation of sludge, its removal is desirable and means to that end are to be recommended.

Mr. H. Mackegg. The contribution by Mr. Mackegg is interesting and instructive. The latter part of this contribution deals exclusively with centrifugal oil purifiers and their operation, a subject on which the contributor can speak with such authority that further comment thereon by the author would serve no useful purpose.

The recommendations regarding the testing of new engines deserves careful attention by engine manufacturers. However, unless it is possible to install a centrifugal oil purifier which has a sufficient capacity effectively to deal with the whole of the oil being circulated, the author would repeat that it is very desirable to include a suitable full flow filter or strainer in the oil cleaning equipment. As shown in Mr. Paxman's contribution, a remarkably low percentage of new engine bearing failures was achieved by the use of a combination of a centrifuge, full flow filters and a magnetic filter.

Some of the particles of solid impurities in oil are so very minute that the author does not believe it is possible to ensure that the oil can be kept entirely free from solid impurities. Nor does he consider it essential that it should be provided that the oil cleaning equipment is capable of ensuring that the maximum size of the solid particles carried to the bearings is kept below a safe value. One object of the paper was to provide a means by which that safe value could be determined.

Mr. J. A. Jaffrey, M.Sc. The writer's instructive contribution draws attention to a point the effect of which was perhaps insufficiently considered in the preparation of the paper, namely, manufacturing inaccuracies. While the results of the author's investigations would undoubtedly be affected to some extent by such inaccuracies, the author believes that the method of determining the working oil film thicknesses given in the paper can advantageously be applied to marine I.C. engines. A knowledge of these oil film thicknesses is essential before we can properly decide upon the fineness of filtration necessary to protect the bearings.

The effect of bearing size on the results has already been referred to in replies to previous contributors. While it must be admitted that cylindrical errors in journals must have an effect on the oil film, the author does not agree that the extent of this would be such as to require better filtration for large marine engines than for comparatively small petrol engines. The cylindrical error of the 10in. journal referred to by the writer may not be so serious as it appears at first sight. The bearing clearance would be correspondingly larger than in the particular cases referred to in the paper and, without investigating the matter, the author would not expect the cylindrical error mentioned to cause any serious difference in the working oil film thicknesses which, of course, would also be larger.

Regarding the effect of misalignment, to cause a journal to rotate about an eccentric axis in a fully lubricated bearing sets up hydraulic forces sufficient to flex the shaft if the eccentricity is appreciable. Assuming, however, that the shaft is perfectly rigid then, by the "principle of least work", the shaft without lateral loading would tend to rotate about an axis such that the eccentricities in the various bearings would be a minimum.

In order to obtain some idea of the effect of lack of alignment, the author has investigated the following example. A 10in. diameter shaft was assumed to be running in a series of bearings having a radial clearance of 0·005in., the bearings being equally spaced at 50in., and all in perfect alignment. It was assumed, however, that the shaft was slightly bent near one end so that this end journal was eccentric by 0·0025in. when the rest of the shafting was concentric in the remaining bearings. The shaft was assumed to be rotating at 120 r.p.m. and subjected to a single harmonic load at the bent end 260lb. per sq. in. amplitude. A similar shaft but perfectly straight and similarly loaded was investigated for comparison. The calculated minimum oil film for the first case was 0·0022in. and in the other case 0·0024 approx. This perhaps will not appear to be so extraordinary when it is realised that, to deflect the end section of the shaft to a concentric position only requires a force of 883lb. The flexibility of this shaft would not be greatly different from that of a crankshaft of equal diameter, so that it would appear that a reasonable error in alignment should not seriously affect the oil film thickness under normal running conditions. Lack of alignment, however, would only have effect in the main bearings and since, as stated by the writer, the crank pins wear more than the journals, the author considers it to be sufficient to apply the calculations to the big end bearings on the assumptions that, if we can arrive at the conditions necessary for a minimum degree of wear at these bearings, these conditions would also satisfy the main bearings.

The question of testing new engines has been dealt with by, and in replies to, earlier contributors. In connection with the case of the new engine trials described by the writer, it is not possible, of course, to reach any conclusion regarding the scoring of the big and small ends without more detailed information. If no trouble was experienced with the main bearings, it is possible that the trouble with the big and small ends may have been more to do with faulty fitting than foreign matter in the oil.

The author is in complete agreement with the writer in stressing the need for thoroughly cleaning bearings and supply channels before testing new or overhauled engines. While agreeing with his second point relating to the arrangement of oil pipes to bearings, the author considers that this would be of less importance if the oil cleaning arrangements provided could be entirely relied upon.

With reference to the writer's suggestion to use a hollow crankpin to retain particles of foreign matter which could be caused to collect therein by centrifugal force, an equivalent arrangement is to be found in some aero engine crankshafts in which are formed over-drilled crankwebs providing pockets for a similar purpose. In this case, however, the pockets are only cleaned when the engine is dismantled, the pockets being large enough not to become filled during the period between overhauls.

Where, as in the case of the larger marine engines, a big proportion of the oil is fed to parts other than the crankshaft, the author agrees with the writer's suggestion that a special auxiliary filter for the oil passing to the crankshaft bearings would be desirable.

In connection with magnetic filters, the author's remarks in the paper did refer more particularly to metal particles resulting from wear. The author agrees that, in those cases where welding splash, mill scale and the like may be caught up in the oil stream, the magnetic filter can serve a very useful purpose.

Effect of Alignment Inaccuracies.

Since writing the above, the author has given further consideration to the effect of alignment inaccuracies on the oil film thicknesses in the main bearings. For this investigation a shaft was assumed to be supported in a series of bearings, one of which was out of alignment by an amount e_B. The journal in this bearing was also also assumed to be out of alignment with the rest of the shaft e_J. This journal was assumed to be (*a*) unloaded, (*b*) subjected to a S.H.M. load having a frequency equal to the R.P.S. of the shaft. The sum of the eccentricities was assumed to be less than half the radial clearance of the bearing.

By adding the necessary terms to the differential equations in the appendix, page 118, the results illustrated by Fig. 16 were obtained.

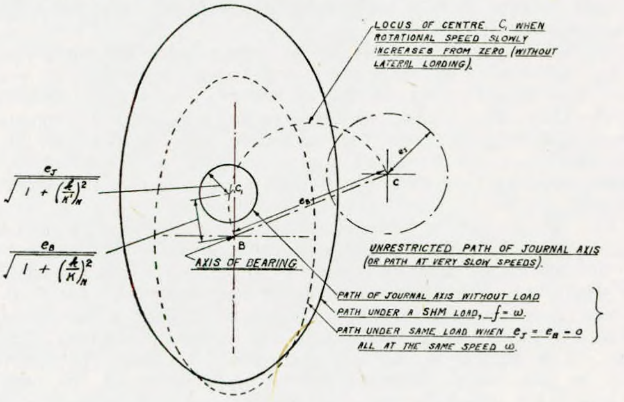


FIG. 16.

From this figure it will be seen that, at very slow speeds, the journal axis in the bearing considered, when not loaded, rotates in a circle of radius e_J about the centre C. As the speed is increased (slowly) the centre C of this circle travels along the semi-circle CC_1B while its radius gets progressively smaller. In the formulæ given for the radius of this circle and the distance of its centre from B, k has the same meaning as in the appendix to the paper and $K'=K\left(1-\frac{0{\cdot}36N^2L^4}{10^{12}d^2}\right)$ for an end bearing, and $=K\left(1-\frac{0{\cdot}124N^2L^4}{10^{12}d^2}\right)$ for an intermediate bearing, N being the R.P.M., L the distance in inches between the bearings (assumed equi-spaced), d the diameter in inches of the shaft or an equivalent equally stiff shaft, and K the static elastic constant in lb. per inch for the shaft at the bearing considered but with the bearing removed.

The two ellipses show the path traced by the journal axis (1) with the eccentricities stated and (2) with the eccentricities reduced to zero. The results illustrated only apply when the maximum working eccentricity of the journal is small, but they may be considered as sufficiently accurate when the maximum working eccentricity of the journal is less than half the radial clearance. For greater eccentricities these curves would all be distorted, but they give a fairly approximate idea of the effect which would be obtained.

For the case illustrated, the oil film is reduced by 0·3 times the sum of the bearing and journal eccentricities. In practice the reduction would be considerably less than this under normal working conditions. In order to make the diagram clear $\left(\frac{k}{K}\right)_N$ was taken to equal 4, whereas in practice this ratio would usually be over 200 for a marine engine shaft under normal running conditions.

As will be understood, the effect on the oil film depends not only upon the eccentricities of the bearing and journal, but also upon the angle, A say, between the direction of the load and the line of centres BC. Thus, when working at a given speed with eccentricities e_J and e_B at one bearing, we obtain a maximum reduction in the film thickness equal to $\frac{(e_J + e_B)}{(1+r^2)^{\frac{1}{2}}}$ when tan A is equal to $(-r)$, and a minimum reduction equal to $\frac{(e_J - e_B)}{(1+r^2)^{\frac{1}{2}}}$ when cot A is equal to r, r being equal to $\left(\frac{k}{K}\right)_N$, (K′ and K being practically equal). Thus, for a given eccentricity of a journal, if we could make e_B equal to e_J and cot A equal to $\frac{k}{K}$ then, at the R.P.M. corresponding to $\frac{k}{K}$ the minimum oil film thickness would be the same as if the bearing and journal were both concentric. The oil films in the adjacent bearings would, however, be influenced to some extent.

It will thus be seen that the oil film thickness under normal running conditions is not so seriously affected by alignment inaccuracies as might be expected. Under starting conditions, however, when the value of $\frac{k}{K}$ becomes equal to zero, the effect of misalignment may be far more serious, especially if the sum of the eccentricities considered should equal or exceed the radial clearance of the bearing.

MEMBERSHIP ELECTIONS.

Date of Election, June 8th, 1943.

Members.
Louis James Jackson.
William Greener Jackson.
John Portanier.

Associate Members.
Alexander Muir.
William White.

Associates.
Victor V. M. Chatfield, B.Sc.
Stanley Newman Clayton.
Kenneth Ernest Frost.
Edward Sydney Clement Osborn.
Thomas Johnstone Stewart.
David Whitton.

Date of Election, July 20th, 1943.

Members.
William Stafford Blenkinsop.
John Harwood Clarke.
John Scott Crawford, Major General
William Dalrymple.
David Shaw Edgar.
Francis Vincent Everard.
Robert William Grant.
Thomas Sidney Hill.
Henry John James Redwood.
Folkert Albert Willem Roorda.
David Scott.
Douglas Ralph Weaver.

Associates.
James Cummings.
Reginald William Groom.
John Harle.
James Lawrence Kinmond.
John Swanson Livingstone.
John James Marshall.
Donald Eric Millar.
David Easson Murray.
John Scott, Sub-Lieut.(E.), R.N.
George Wyllie Steele.
George Vitt.

Graduate.
John Beresford.

Student.
Peter Emerson Melly, Sub-Lieut.(E.), R.N.

Transfer from Associate Member to Member.
Ernest Alfred Lanyon Nicholas.
George Sidney Selman.

Transfer from Associate to Member.
John McDonald Paterson.
Henry Carmichael Williamson.
Leslie Williamson.

Transfer from Student to Graduate.
Arthur Edwin Richards.

Abstracts of the Technical Press

Report on 1,200-lb. Reheat Marine Installation.

A brief report on the operation of the high-pressure reheat propelling-machinery installation of the American s.s. "Examiner", was recently given in a paper presented at a New York meeting of the Society of Naval Architects and Marine Engineers by Messrs. B. Fox and R. H. Tingey, who were the authors of the paper entitled "A 1,200-lb. Reheat Marine Installation" presented to the Society in 1941 (*see abstract on p. 166 of* TRANSACTIONS, *January, 1942*). The "Examiner" went into service early in 1942 and spent seven months in foreign waters before returning to a U.S. port. During this time she steamed over 30,000 nautical miles at an average speed of 15·85 knots, the mean O.F. consumption for all purposes being 0·521lb./s.h.p.-hr. On trials it was found necessary to reduce the reheat to about 680° F. to maintain a practicable division of load between the superheat and reheat furnaces; under these conditions the superheat was only 685° F. After the trials the heating surface of the primary superheater was increased, and baffles were fitted to prevent by-passing, whilst the water-screen tubes were respaced to make the primary superheater and the reheater more effective. The improvements which resulted from these changes proved inadequate and it is therefore proposed to increase the secondary superheater surface as soon as the ship becomes available. This is expected to raise the superheat and reheat temperatures to about 740° F. and to produce a satisfactory division of load between the two furnaces. The most serious difficulty encountered with the boilers was leakage through the superheater handhole gaskets. It proved necessary to reseat all the handhole faces during the first voyage, and great care had to be taken when connecting up the boilers, the handhole doors having to be tightened up at frequent intervals as the pressure was raised. When the new secondary superheater is installed in order to correct the unbalanced load, all the handholes in it will be eliminated and blind nipples will be rolled in instead. There was one boiler tube failure in a fire row and two other tubes were badly blistered in service. These failures were due to low water level. The economiser consists of hairpin tubes having individual return headers bolted against the tube ends with soft iron gaskets in the joints. Many of these joints leaked and it was found necessary to reface the tube ends and remake the joints before the first voyage. During the voyage a faulty gasket at the economiser distributing manifold developed a leak and had to be renewed, but no trouble has been experienced with these joints since that time. During the first two trials the pressures recorded in the H.P. turbine proved incorrect, whilst the water consumption was somewhat excessive. The H.P. turbine was therefore opened up and some alterations were made in the nozzle areas and the clearances. These changes produced some improvement and reduced the water rate by 3 per cent. After steaming 23,000 miles, mostly at normal power, the shaft revolutions suddenly dropped from 85 to 75 r.p.m. without any outward manifestation of trouble. It was then noticed that the steam pressure at the inlet to the H.P. second stage had suddenly gone up to boiler pressure, whilst the I.P. turbine inlet pressure had dropped very low. Circumstances made it necessary for the ship to continue on her course without stopping, but on the following day a slight noise developed in the H.P. turbine and its thrust bearing began to heat up, so the engines were stopped to allow the H.P. and I.P. turbines to be disconnected. The emergency piping was fitted and the ship proceeded to port using the L.P. turbine only. When the H.P. turbine casing was lifted it was found that the blading of the fourth and fifth stages was seriously damaged and that the corresponding nozzles were practically closed. The damaged fourth, fifth and sixth diaphragms were removed and the blades of the fourth and fifth stages were machined off flush with their roots in the ship's lathe. The work was carried out by the ship's staff. The H.P. turbine was then closed up and the vessel proceeded homeward taking steam through all three turbines in the usual manner. The original cause of the trouble was the breaking off of a piece of shrouding in the fourth stage. New blading and diaphragms have since been fitted; the weight of shrouding and number of blades per segment have been slightly altered and additional precautions were taken in welding the shroud to the blade tips. The I.P. and L.P. turbines were opened up for examination after 12 months' running and were found to be in good condition except that some blade roots in the I.P. turbine were loose and had to be recaulked. Whereas the original emergency connections provided for running on the L.P. turbine alone with the H.P. and I.P. disconnected or on the H.P. and I.P. turbines alone with the L.P. disconnected, additional piping has now been provided to enable the ship to steam on the I.P. and L.P. turbines with the H.P. disconnected. This arrangement provides for about 75 per cent. of full power to be available in the event of damage to the H.P. turbine. The feed control system for the reciprocating pumps consists of a pressure regulator which varies the stroke of the pump to maintain a constant discharge pressure, and a boiler level regulator which throttles the flow to the boiler. As the action of the level regulator is much faster than that of the pressure regulator, an automatic by-pass valve is fitted on the pump discharge to take away the excess feed water for the short time that the pressure regulator takes to adjust the pump stroke to the new feed requirements. In addition to this by-pass valve, there are relief valves on the pump discharge. The automatic by-pass valve proved unsatisfactory in service and had to be shut off the feed line, its functions being taken over by the relief valves. These, however, were not designed for such frequent service, and their seats were destroyed in a short time, so the ship's staff put the feed control devices out of action and regulated the feed-pump stroke by hand. This was not altogether satisfactory as it was impossible to prevent excessive pressures from building up momentarily. These defects have now been made good. The reheat system operated satisfactorily at all times and the drainage arrangements provided for the reheat piping proved adequate. No trouble was experienced with the high-pressure steam piping and joints, but two joints—originally defective—in the feed system had to be remade on the first voyage. The glands of the reciprocating feed pumps had to be repacked at frequent intervals owing to the excessive pressures built up in the feed system. On these occasions the turbine-driven pump was used and it performed very satisfactorily. At sea, the turbo-generators were run on bled steam, and the automatic changeover devices for putting the generators on live steam when the main turbines are slowed down, worked admirably at all times (*for description see abstract on p. 137 of* TRANSACTIONS, *November, 1941*). The 1,200-lb. pressure-reducing valves, which are of the air-dome type, have been very satisfactory. No trouble was experienced with any of the combustion-control gear, and the compressed-air soot blowers worked very well. The fuel consumption of the "Examiner" was, on the average, 10·3 per cent. less than that of the "Exchange", a sister ship having geared turbines operating with superheated steam at 425lb./in.² pressure and 740° F. temperature, but it is anticipated that the O.F. consumption of the former vessel will, in due course, improve to an extent corresponding to the 13 per cent. economy originally aimed at. The high-pressure reheat cycle would probably show up to better advantage with powers greater than that of the "Examiner" (8,000 s.h.p.), whilst with lower powers the economy to be gained from its use will be proportionately lower. Steam reheat is slightly less efficient than gas reheat, but the machinery installation required is somewhat simpler and lighter, and therefore less expensive. The slight increase in congestion in the "Examiner's" machinery spaces caused by the reheat installation constituted no inconvenience in service, neither did the reheater pipes impose any additional burden on the E.R. personnel as regards maintenance. Although the E.R. complement of the "Examiner" includes three additional firemen, it is unlikely that these extra men will be needed when the experimental stage with the new machinery has been completed.—*"The Marine Engineer", Vol. 66, No. 790, May, 1943, pp. 102-104.*

A New Lentz Poppet Valve Arrangement.

In a paper read before the Lower Rhine section of the Association of German Engineers by Dipl. Ing. B. Bleicken, and which he subsequently published in *Werft*Reederei*Hafen*, the author pointed out that maximum economy in machinery installations comprising a reciprocating steam engine and exhaust turbine is only attainable if both the H.P. and L.P. components are correctly designed in the light of present-day knowledge. If due regard is paid to these considerations, however, it is possible to have a steam consumption of the order of 10lb./b.h.p.-hr. in engines of about 2,000 b.h.p. The Lentz double-compound steam engine is particularly suitable for such installations on account of its simple and robust poppet valve gear. In the earlier type of Lentz double-compound engine the upper and lower exhaust valves of the H.P. cylinder served as the steam inlet valves to the corresponding ends of the L.P. cylinder, whilst the engine was of the usual Woolf compound design with the H.P. and L.P. cranks 180° apart and no intermediate steam receiver. The latest type of Lentz engine, however, differs from that which has been familiar for the past 20 years in that the steam inlet and exhaust double-beat poppet valves of the H.P. cylinder are superimposed instead of being arranged alongside each other in a fore-and-aft line with radial steam passages to the cylinder.—*"The Marine Engineer", Vol. 66, No. 790, May, 1943, pp. 115-116.*

Explosion Doors for Boilers.

In an article in the German periodical *Glückauf*, by A. Sauermann, it is stated that the purpose of explosion doors or relief openings in boiler flues or casings is to reduce the risk of damage to the casing, C.I. economiser or other vulnerable components in the event of the explosion of unburnt gases or pulverised coal. The doors must be located at or near the places where explosive mixtures are most likely to accumulate, and in order that they may serve their purpose they must, of course, be designed to open as the result of a small excess pressure and afford an adequate aperture for the escape of the large volume of hot gases produced by the explosion. It is often difficult to find room for such doors in suitable positions, as well as in preventing them from warping at high temperatures, thereby producing leakage and permitting cold air to infiltrate through the leaks, besides causing loss of draught. The number of factors involved is so great as to make it impracticable to calculate suitable door dimensions, and all that can be done is to limit to the lowest attainable value the pressures developed and maintained inside the boiler casing and flues. The author presents data from tests undertaken to record the pressure inside a cubical enclosure, of about 6½ft. inside dimensions, during and after the ignition of various explosive mixtures inside the chamber, the latter being successively fitted with different types of explosion relief doors. The maximum pressure attained was about 17·5lb./in.^{2}, which was more than twice the pressure at which the brickwork of the experimental enclosure was damaged. This pressure was developed when using a relief door weighing 18¾lb. and providing an opening of about 8in. by 12in. With two larger doors, each about 18in. square, the maximum pressure developed did not exceed 6lb./in.^{2}, notwithstanding the much greater weight of the doors, each of which weighed about 61lb., nearly 26lb. of which was normal to the inclined seating of the door. The lowest pressure recorded was 1·7lb./in.^{2} when using a "Jola" relief panel, the construction of which resembles that of a double window, the inner frame with "panes" of asbestos or similar material which is easily blown out by an explosion. The outer frame is a loose fit with a retaining chain to restrict its flight, and the space between the two frames is filled with ash which serves as heat insulation and helps to smother the flame of the explosion. The free area of this relief panel was about 3ft. 1in. by 5ft. 9in. All the tests confirm the conclusion that while lightness of the door is an advantage, it is much more important that the area of the relief opening should be as large as possible.—*"The Power and Works Engineer", Vol. XXXVIII, No. 442, April, 1943, p. 90.*

Superheated or Saturated Steam for Triple-expansion Machinery.

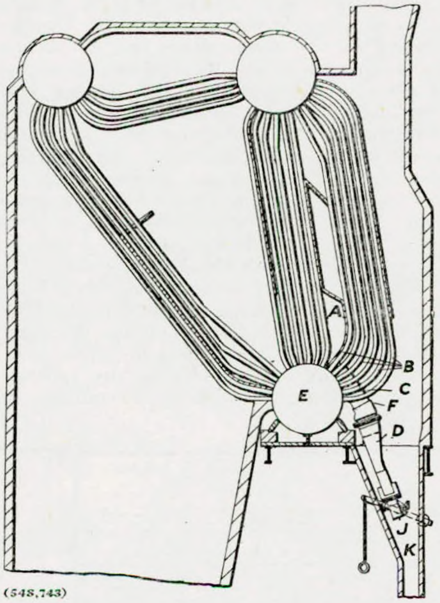
The performance of five sister ships (two employing superheated and three using saturated steam) of about 9,000 tons d.w., with triple-expansion machinery, was analysed over a period of six years. The results showed the superheated ships to be 7 per cent. more economical than the saturated vessels. One of the superheated ships made a few voyages with the superheaters removed, and the result was a 5 per cent. loss in economy as compared with her earlier results. The lagging in the two superheated ships was more efficient than that of the saturated ships, and the effect of this superior lagging probably accounted for 2 per cent. of the gain in economy, so that the net gain derived from the use of superheated steam (190° F.) was 5 per cent.—*Thos. Malone, "The Journal of Commerce" (Shipbuilding and Engineering Edition), No. 35,931, 8th April, 1943, p. 7.*

Mechanically-fired Vessels in War-time.

Experience with cargo ships equipped with mechanical stokers and Scotch boilers during the present war, would seem to indicate that the misgivings expressed in some quarters concerning the ability of mechanical stokers to handle widely differing types of coal are unfounded. Stoker-fired ships have, in fact, been compelled to use fuel ranging from American New River coal with a calorific value of 14,245 B.Th.U./lb., 13·52 per cent. volatile matter and 4·38 per cent. ash to low-grade English slack of 10,548 B.Th.U./lb. calorific value, 29·6 per cent. volatile matter and 19·26 per cent. ash, and in spite of this have successfully met the difficult conditions experienced when steaming in convoy. Thus, at one moment the steam demand from three Scotch boilers in one such vessel would be 53,000lb./hr. and a few minutes later it would fall to 16,000lb./hr. On one occasion, it is stated, a ship having a normal speed of 13½ knots was shadowed by a U-boat for several hours and although low-grade slack was being burned, generated sufficient steam to maintain a speed of 15¼ knots for six hours until the U-boat was forced to give up the chase.—*"Shipbuilding and Shipping Record", Vol. LXI, Nos. 15/16, 22nd April, 1943, p. 344.*

Removal of Fly Ash.

An arrangement for facilitating the removal from watertube boilers of the fly ash which is carried by the gases and which collects in the lower bend of the gas pass where a baffle meets the lower water drum, has been developed and patented by a well-known British firm of mechanical-stoker manufacturers. Referring to the accompanying diagram, the lower edge of the baffle (*A*) at the rear of the gas pass is provided with a number of apertures where it meets the lower water drum (*E*). The openings are located between the rows of tubes (*B*) which form the bank behind the baffle (*A*) and are connected by pipes (*C*) which pass between, say, every fourth pair of rows of tubes to an ash chute (*D*). The latter is placed at the back of the drum (*E*) and, to allow for expansion and contraction, each pipe (*C*) is connected to its chute (*D*) by a flexible connection (*F*) consisting of a sleeve of asbestos cloth inside a protecting metal shroud. The outlets from these chutes (*D*) are closed by valves (*J*) which are opened momentarily from time to time to discharge the ash collected in the chutes into an ash hopper (*K*). The ash pipes are divided into groups, each group being provided with an ash chute and ash hopper, so that one group can be operated at a time. With this arrangement the fly ash is removed without leakage of the furnace gases or entrance of cold air, except for that which passes through the ash-chute outlet when the valve is momentarily opened.—*"Engineering", Vol. 155, No. 4,032, 23rd April, 1943, p. 340.*



"Blowing Back to Evaporator".

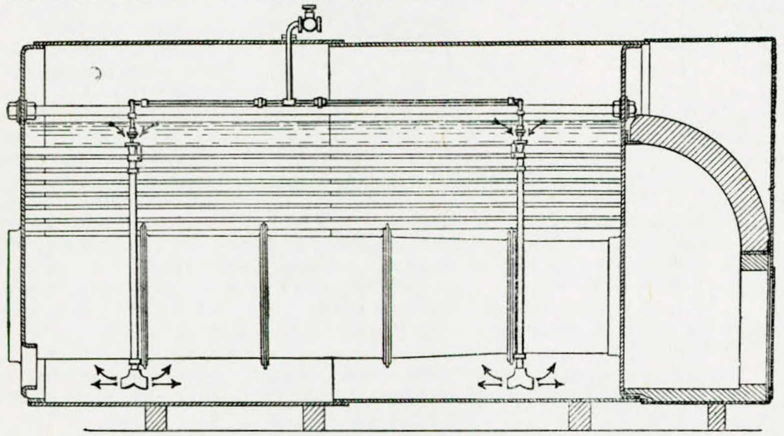
In many American steamships of recent construction the boiler blow-downs are connected by a small pipe line to the shell of the evaporator. This arrangement makes it possible to blow the water from the lower parts of the boilers direct into the evaporator, and the boilers can therefore be given a slow blow-down for a considerable period of time without losing all the heat of the water blown out. The dirt and sludge blown out of the boilers passes into the evaporator shell, whence it is removed when the latter is cleaned out. This method of blowing down the boilers is known as "blowing back to the evaporator".—*"Marine Engineer and Shipping Review", Vol. XLVIII, No. 5, May, 1943, p. 236.*

Thermometer in Boiler Uptake.

A thermometer for recording the temperature of the gases in the boiler uptake provides a ready means for checking the efficiency of the boilers. The temperature of the funnel gases should, of course be as low as practicable, since as much heat as possible should be extracted and given to the boiler water. For ordinary marine boilers a 40° F. reduction in the temperature of the gases passing from the uptake to the funnel means an increase of approximately 1 per cent. in the efficiency of the boilers. As the boiler tubes become fouled with soot during steaming they become less efficient for transmitting heat and the uptake temperature will be increased. This rise in the temperature of the gases may be used as a guide for using the soot blowers. If the uptake temperature is brought down very low (below dew point), corrosion may occur due to the formation of water vapour in the gases, but this is unlikely to take place unless the temperature is below 260° F.—*"Marine Engineering and Shipping Review", Vol. XLVIII, No. 5, May, 1943, p. 238.*

Boiler Temperature Equaliser.

An engineering firm in the North Country have developed a device known as Ward's Patent Boiler Temperature Equaliser for improving the internal circulation of cylindrical boilers when starting up from cold. For this purpose steam taken from the nearest auxiliary steam range is utilised to produce an injector-like action, the heat of the steam being likewise employed to the best effect. The steam is admitted through a special non-return stop valve at the top of the boiler to internal pipes connected to two (or more) equaliser injection units located just below the normal water level. The jets of steam issuing from the injector nozzles at high velocity impart energy to the columns of water in the mixing cones, thereby producing a rapid displacement of the water down the circulating pipes to the bottom of the boiler, where the warmed water is dispersed close to the shell plates. The equaliser can be put into operation before the fires are lit in the boiler furnace, thereby setting up circulation and gradually raising the temperature of the water itself. It is also advantageous to turn on the equaliser when shutting down a boiler, in order to ensure the maintenance of even temperatures; moreover, the water is thus kept in movement and solid particles remain in suspension instead of settling on the tube or plate surfaces to form hard scale. Finally, the heating of the water by the steam during starting up, combined with the forced circulation, helps to raise the steam pressure in the boiler to working level without any risk of straining the boiler by firing up too rapidly. The equipment takes up very little space within the boiler and does not obstruct access for internal examination and cleaning. The temperature equaliser can be fitted easily and rapidly to any type of cylindrical boiler. The accompanying sketch shows in outline an



Ward's Temperature Equaliser fitted in an Economic Boiler.

application to a boiler of the Economic type, but the arrangement indicated is equally suitable for any marine-type or waste-heat boiler. —*"The Power and Works Engineer", Vol. XXXVIII, No. 443, May, 1943, p. 119.*

Diesel Engines in Torpedoed Ship Keep Running.

An American cargo vessel in the West Indies was recently struck by a U-boat's torpedo forward and caught fire. The vessel developed a heavy list, whilst the bow was submerged about 15ft. Under these apparently hopeless conditions, the crew were ordered to abandon the ship and reached port safely. Shortly afterwards an aircraft on patrol duty reported the ship still afloat, whereupon a salvage crew was sent out accompanied by the chief engineer, second engineer, second mate and some of the crew. The salvage crew succeeded in extinguishing the fire in several hours. The second engineer, who was the first to board the ship, found the two 8-cylr. Cooper-Bessemer auxiliary Diesel engines still running, and the lubrication system of both units functioning perfectly after 40 hours, even though the engines were tilted at a steep angle due to the ship's list. The main and auxiliary engines in this particular vessel were run on Bunker-C oil, a relatively cheap, low-grade fuel containing considerable carbon and other impurities.—*"Motorship", Vol. XXVIII, No. 3, March, 1943, pp. 224-225.*

Pressure-charged Auxiliary Diesels.

Over a period of about 10 years Harland & Wolff, Ltd., have built many 6-cylr. four-stroke auxiliary Diesel engines of 500 b.h.p. at 300 r.p.m. for driving ships' generators. The firm have now developed a modified type with pressure charging. The new engine has a somewhat shorter stroke and develops its rated power of 500 b.h.p. at 500 r.p.m. with normal aspiration, but produces about 45 per cent. more power when pressure-charged on the Büchi system. The weight and space occupied per b.h.p. show a considerable advance on previous practice, while the specific fuel consumption is low at all loads.—*"The Marine Engineer", Vol. 66, No. 789, April, 1943, p. 94.*

Diesel and Steam Propulsion of Tugs.

It has been suggested that steam tugs can become serious competitors to Diesel-engined tugs on the River Thames, but it should be remembered that the average steam tug must coal at least once or twice each week, whereas the Diesel tug can take in sufficient fuel at one time to cover up to seven or eight weeks' towage. There is also the fact that the hull of the Diesel tug is not subjected to the destructive corrosion found in the bunkers of coal-fired vessels. The steam tug has to be out of service for at least an hour each week for boiler-tube cleaning, while the jackets of a Diesel engine only require to be washed through every few months. If spare fuel valves and air starting valves are carried these can be changed and overhauled at leisure, thus maintaining the engine in an efficient condition. The time and fuel lost while raising steam and lying with banked fires are eliminated in the Diesel tug, which, moreover, does not have to maintain steam while awaiting orders or craft from docks, etc. Finally, the relatively uneconomical burning of coal is to be deplored, especially in war-time, bearing in mind the valuable by-products that are wasted. Closer co-operation between the builders of engines for Thames tugs and the engineers responsible for their operation appears to be desirable, and propeller design could also be improved. As many tugs now operate on an overload during heavy tows, the writer suggests that a tug should be designed for a given pull at a given towing speed and not for a free-running speed, which latter is not of interest to towage contractors. An ideal design would consist of twin engines driving a single shaft through hydraulic couplings and reduction gearing. When running free or with a light tow, such an arrangement would enable one unit to be shut down and, by providing suitable pipe connections, the discharge water from the working unit could be passed through the jackets of the idle engine, thereby keeping the latter at the correct temperature for starting up and placing on load immediately. The propeller might, with advantage, be enclosed in a Kort nozzle. The two-stroke Diesel engine, with a minimum of working parts, is a suitable power unit for river tugs, as it can be operated successfully by semi-skilled labour under the supervision of a shore engineer. There is also room for further improvement in the design and construction of the present type of Thames barge. The building costs of such craft could probably be reduced and their construction simplified by a combination of welding and riveting carried out on a large scale to a standard design, with plates flanged to templates by hydraulic means and sub-units assembled at the works for completion on the river site.—*E. O. Stephens, 'The Marine Engineer", Vol. 66, No. 789, April, 1943, p. 89.*

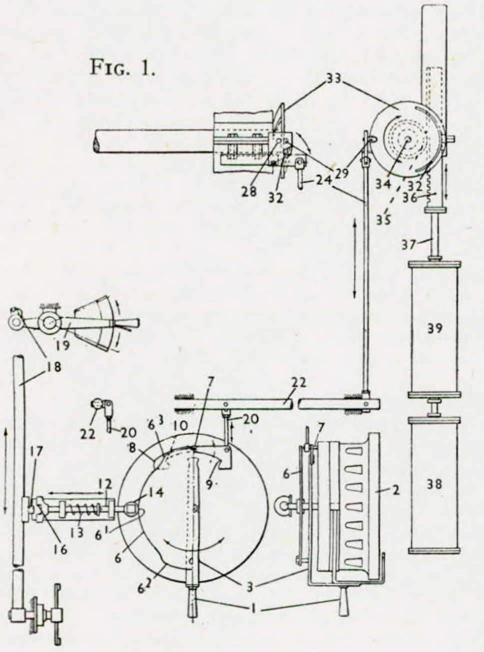
Foreign Developments Needing British Investigation.

It is suggested, among other things, that the following developments in Diesel machinery reported from foreign countries, should receive attention here: *(1)* the internal-combustion turbine using oil fuel, already constructed in Switzerland in sizes up to 5,000 h.p.; *(2)* the employment of variable-pitch propellers for large ships; *(3)* the adoption of a high degree of supercharging in two-stroke engines in conjunction with exhaust-gas turbines, which Sulzer Bros. have developed to a commercial stage, and which it is proposed to install in a ship shortly after the conclusion of the war; *(4)* the production of high-powered Diesel engines operating at speeds of 750-1,500 r.p.m. Already 1,600-b.h.p. engines running at 700 r.p.m. are being utilised in numerous American ships of various sizes, some of which will probably be operated by British owners.—*"The Motor Ship", Vol. XXIV, No. 279, April, 1943, p. 3.*

An Engine Room Telegraph Safety Device.

A recently published British patent covers a safety device for an engine-room telegraph mechanism which prevents the reply lever from being operated to transmit a signal to the bridge until the reversing gear of the engine has been moved in accordance with the order received. Referring to Fig. 1, and assuming that an order to go ahead has been rung down from the bridge to the E.R. telegraph receiver (*2*), the engineer on watch will lower the lever (*19*) to open a by-pass valve on the servo-motor oil cylinder (*38*) and thereby also admit compressed air to the top of the air cylinder (*39*), causing the piston rod (*37*) to pull down the rack (*36*). The lever (*19*) is then brought to the centre position and the supply of air to the air cylinder is cut off, while at the same time the by-pass valve on the oil cylinder is closed, so that the reversing gear is locked in the ahead position. During the operation of moving the rack (*36*) down to the bottom, the disc (*33*) rotates clockwise and the helical blade (*32*) makes contact with two rollers (*28, 29*). The rod (*24*) is moved down and rotates the shaft (*22*), thereby depressing the rod (*20*) and moving the rocking lever (*8*) which turns on a pivot (*7*). In this position the engineer can reply to the bridge telegraph by moving the lever (*1*) to the right, as the hook (*10*) on the rocking lever (*8*) has been raised to clear the arm (*3*), while the hook (*9*) has been lowered to prevent the arm from moving in the wrong direction. When the engineer swings the lever (*1*) to the right the disc (*6*) revolves and the cam 6^3 exerts pressure on the roller (*14*). This causes the rod (*12*) to overcome the pressure of the spring (*13*), so that the groove (*16*) engages the stop (*17*), which is fixed to the reversing rod (*18*). As this rod cannot now be moved up or down, the reversing gear is locked. The lever (*19*) and the rod (*18*) can be moved when the telegraph is at the "Standby", "Stop" or "Finished with Engines" positions, owing to these locations corresponding with the peripheral surface of the cam disc ($6, 6^1$) between (6^2) and (6^3). If the engines are running ahead and an order to go astern is received, the engineer is prevented from answering the order by means of the telegraph (*2*), since the arm (*3*) is blocked by the hook (*9*) of the lever (*8*). Thus, he must bring the lever (*1*) to the "Stop" position and then move the reversing control lever (*19*). When this has been effected, the mechanism which connects the locking lever (*8*) with the servo motor (*38*) will make a corresponding change in the position of the locking lever and the engineer is then able to answer the telegraph.—*"The Motor Ship", Vol. XXIV, No. 280, May, 1943, p. 68.*

FIG. 1.

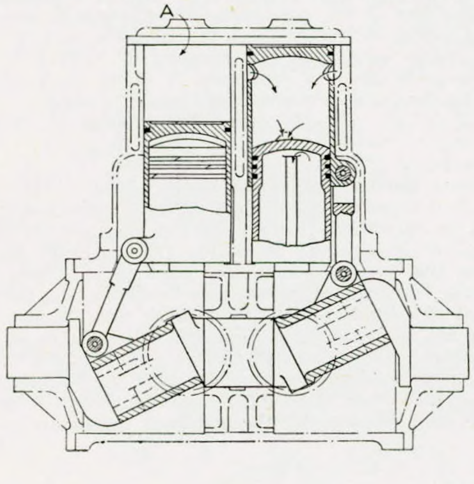


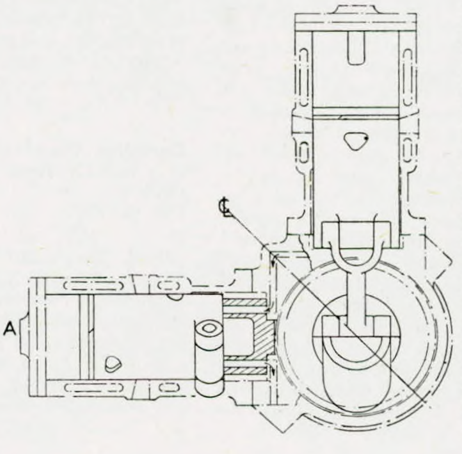
Triple-screw Diesel Tug for Service in Ohio River.

The Standard Oil Co.'s triple-screw Diesel tug "Sohioan" recently built by the St. Louis Shipbuilding and Steel Co., for the towage of oil barges on the Ohio River, is the first vessel of her type to have a streamlined deck structure in conjunction with a scow-model bow. The hull is 160ft. by 38ft., with a draught of 6ft. 6in., and the propelling machinery is aft. The triple screws are driven by three 7-cylr. Fairbanks-Morse two-stroke Diesel engines, each developing 805 b.h.p. at 300 r.p.m. The engines are operated from a single control stand located on a platform at the forward end of the engine room, to which the electrically-operated E.R. telegraphs are also led. The telegraph levers, as well as the steering levers, are located on a special control desk in the pilot house. There are three steering rudders and six backing rudders, all controlled separately, and actuated by means of electro-hydraulic gear. The pistons of the main engines are oil cooled whilst the cylinder jackets are cooled by water circulating in a closed cooling system. An unusual feature of the latter is that in normal operation the ship's hull is used as a heat exchanger, although a separate heat exchanger serves the centre engine during the hot summer months. The main engines drive their own scavenging, fuel-oil, circulating and lubricating-oil pumps, but the remainder of the E.R. auxiliaries, as well as the stand-by pumps, are driven by electric motors for which current is supplied by two 60-kW. generators driven by 90-h.p. F.M. Diesel engines. The crew's quarters are air-conditioned throughout. Accommodation is provided for 24 officers and men. It is proposed to operate the main engines with crude oil, of which 300 tons can be carried in the tug's fuel tanks. The "Sohioan" is usually employed on the towage of crude oil barges, so she can take in fuel from these without interrupting towing operations. The crude oil is passed through a centrifuge before being discharged to the main fuel tanks, from which it is passed to the service tanks through filters. A second set of filters is fitted between the service tank and the injection pumps.—*"Motorship", Vol. XXVIII, No. 4, April, 1943, pp. 264-266.*

An Innovation in Sleeve-valve Engine Design.

The accompanying diagrams show the lay-out of a completely new form of internal-combustion engine which is the subject of a recent British patent. Among the novel features of the design is the substitution of a Z-shaft for the usual crankshaft, the employment of static pistons anchored to the crankcase and internally water-cooled, and the fact that the working cylinders are formed by blind-ended sleeve valves which are connected through short rods (in which the stresses are mainly those of tension) to the Z-shaft. The sectional diagrams show the general arrangement of a V-type (90°) four-cylinder two-stroke oil engine of $4\frac{1}{2}$in. bore and stroke, in which the distance between the crankcase centre-line and the automatic air valves in the cylinder covers of each bank of cylinders is only $19\frac{1}{2}$in. A development of such an engine would be an X-type eight-cylinder unit. By reason of the connection of the sleeve-piston to the Z-shaft, the movements of the sleeve are both linear and rotational. This sleeve uncovers air-inlet ports leading from the cylinder proper into the combustion space between the sleeve piston and the static piston, whilst other ports are provided for the outlet of the exhaust gases and uncover, at the appropriate moment in the cycle, a port connecting the combustion space with the end of the fuel injector. Air enters the cylinders through the automatic inlet valves (A) in the covers being drawn in by the motion of the sleeve piston towards the Z-shaft. When the sleeve pistons reverse their motion, the air is compressed between the crowns of the sleeves and the cylinder covers until the air-inlet ports are opened, whereupon the air passes into the combustion spaces. Some degree of supercharge should be possible, because each air-pumping cylinder has a capacity about 25 per cent. greater than that of the working cylinder within the sleeve piston. The elliptical motion of the sleeve pistons should facilitate the lubrication of their outer and inner peripheries, whilst the application of water cooling by means of





ordinary external water jackets and internally in the static pistons, assists even heat dissipation, the absence of the masses of metal due to the gudgeon-pin bosses in the conventional type of engine, making such heat dissipation more effective. The crowns of the sleeve piston are, of course, cooled by the air which is drawn in through the air inlets and which is practically at atmospheric temperature before it is compressed. The design of the engine would appear to indicate that the weights of the various reciprocating parts would be lower than with a conventional design, whilst at the same time no difficult production problems would be likely to arise.—*"The Oil Engine", Vol. XI, No. 121, May, 1943, p. 25.*

Domestic Heating Systems for Motorships.

Although steam for heating the living spaces was nearly always employed in earlier Diesel vessels, a much cheaper and more satisfactory installation can often be made if hot water is used, more especially in the case of small craft. The steam system is recommended where the number of radiators or convectors involved would require the use of a two-pipe, forced-circulation hot-water system; where zoning would require a large water volume; or where water-filled radiators and piping in the remote quarters of a large vessel might freeze up. Earlier types of hot-water heating systems involved the use of two pipe mains, one for supplying water to the radiators and one for returning the water to the boiler. The recent development of circulators, flow checks and special take-off fittings for branch risers, however, now makes it possible to use a single-pipe main of relatively small bore, with the result that the entire system is much simpler. In a typical installation of this kind, the boiler (fitted with an automatic burner designed to burn Diesel oil), is placed in the engine room. The hot water leaves the boiler through a 1½-in. pipe near the top and returns to a similar connection near the bottom through a small circulator. The latter is simply a very small centrifugal pump, driven by a $^1/_{10}$-h.p. motor, which may be thermostatically controlled. This pump serves to keep the water circulating through the boiler and the supply main. Just above the boiler, in the main, is a flow-control valve which prevents the circulation of water through the radiators when the circulator is not in operation. The circulator operates only when the room thermostat calls for heat. During the summer months the boiler will probably be required for the supply of domestic hot water only, and as the thermostatic control will not call for heat, the circulator will not operate and the flow valve will remain closed. When the thermostatic control calls for heat the circulator will start, the flow valve opens and water will circulate through all the radiators that are open. The flow of water to and from each radiator through a single pipe is ensured by the use of a simple fitting embodying the Venturi principle. Each radiator is connected to the circulating main on the by-pass system, the inlet connection being a standard T-fitting, whilst the outlet is connected to the throat of a Venturi fitting. The pressure in the throat is less than that at the inlet and outlet of the Venturi, and it is this difference of pressure which maintains a constant flow through the radiator. The design of heating plant for a given ship is usually based on some empirical method of calculation, that recommended by a well-known American firm specialising in marine heating equipment being as follows: The coefficients of heat transfer used for the purpose represent B.Th.U./hr. per ft.^{2}/degree F. temperature difference, so that each coefficient must be multiplied by the difference between the temperature of the outside air and the temperature at which the air in the spaces is to be maintained. In America, it is usual to assume the former to be 0° F. and to make the inside temperature 70° F., so that the temperature difference is 70° F. In spaces such as the engine room, store rooms and passages, etc., a temperature difference of 60° or 65° is used. Taking each space in turn the cubical content is first multiplied by the coefficient 0·06 and by the temperature difference. This coefficient is based on a minimum of three changes of air per hour. Next, determine the ft.^{2} of glass surface and doors in the walls and multiply by 1·13 times the temperature difference. After calculating the ft.^{2} of exposed wall, deduct the area of glass and doors previously determined, and multiply the remainder by the coefficient of heat transfer of the relevant material (obtainable from an appropriate table), applying the temperature difference in the manner stated above. After the above three calculations, add up the results and the total will represent the heat loss of the space concerned. Do this for each space in turn, add up the results and the total will represent the heat loss for the whole ship. The total for each space is used to determine the radiator surface required for that space, whilst the grand total serves to determine the size of boiler required. In a hot-water system with an average water temperature of 190° F. in the radiators, the B.Th.U. emission per ft.^{2} of radiator surface will be 180 B.Th.U., whilst with a water temperature of 197° F. the emission will be 200 B.Th.U. In order to keep the radiator size as small as possible, the boiler controls are usually arranged to maintain a temperature of 205°-210° F. Allowing a temperature drop of 10° F., the average radiator water temperature will then be 195°-200° F., so that the total heat loss of any given space, divided by 200, will give the ft.^{2} of radiator surface required for that space. In the case of the ordinary L.P. steam system the emission per ft.^{2} of radiator surface will be 240 B.Th.U., so that the latter figure should be used as a divisor when calculating the steam radiator surface. The nearest size radiator can then be selected from the manufacturers' catalogue. When determining the size of the boiler required where domestic water heating is included, the water-heating load must be added to the radiation load. If the number of pounds of water to be heated per hour is multiplied by number of degrees that the water is to be raised in temperature, the result will be the B.Th.U. load to be added to the boiler. This domestic water load in B.Th.U. is converted into equivalent ft.^{2} of either steam or hot-water radiation and added to the space-heating load already determined. With this figure a boiler of the correct size can be chosen from the maker's catalogue. In these catalogues the boiler capacity as expressed in "recommended radiation", "net rating", or "standing radiation" in terms of ft.^{2}, is much less than the capacity in B.Th.U. at the outlet. This practice is followed in order to provide additional capacity beyond that anticipated when the boiler is installed, as well as a factor of safety where the shipbuilders depend upon rough estimates of heating capacity, which are usually too low.—*"Motorship", Vol. XXVIII, No. 4, April, 1943, pp. 270, 271 and 307.*

A British-built Norwegian Motorship.

Among the new ships built in this country for allocation to the Norwegian, Dutch and Belgian shipping companies as a partial replacement for the heavy losses which they have been called upon to bear, is a motorship constructed for the Norwegian Shipping and Trade Mission, the general design of which is based on that of the "Economy" motor cargo vessels developed in this country just before the war. The ship has a d.w. tonnage of a little over 10,000, the gross register being 7,073 tons. The engine room is slightly aft of amidships, with three holds forward and two aft. The total bale capacity of the holds and 'tween decks is about 510,000 cu. ft. The five main cargo hatches on the upper deck are similar, and there are two W.T. hatches on the second deck over the deep tank aft of No. 3 hold. The cargo-handling equipment consists of ten 5-ton and two 10-ton derricks, served by 12 winches. The deck machinery also includes an anchor windlass on the forecastle deck and a warping winch aft. The whole of this machinery, as well as the steering gear, is steam-driven. There are six D.B. tanks, of which Nos. 1 and 6 are used for water ballast and the remainder for fuel oil or water ballast. Under the E.R. floor are lubricating-oil tanks with a capacity of 3 tons, flanked by F.W. tanks for make-up boiler feed holding 60 tons. The P. and S. fresh-water tanks on the second deck have a total capacity of over 36½ tons. In addition to the D.B. fuel tanks, there are two fuel-oil tanks at the forward end of the engine room. The total amount of fuel that can be carried is 1,233 tons, exclusive of the 1,000-ton capacity of the deep tank. The main engine, which is of the standard three-cylinder Doxford type, is self-contained in the sense that all the pumps required for the operation of the engine at sea are driven from the crankshaft. Nearly all the E.R. auxiliary machinery is steam-driven; it includes two 15-kW. 110-volt dynamos. There are two cylindrical boilers arranged in a special compartment on the second deck, forward of the engine room and abaft the funnel. One of the boilers is fired by the heat of the main-engine exhaust gases and provides the whole of the steam normally required at sea, but this boiler is also equipped with an oil burner. A smaller oil-fired boiler is provided for harbour service, but is also available for use at sea if required. The E.R. personnel of this Norwegian vessel is made up of three engineer officers, two assistant engineers, a pumpman and seven greasers.—*"The Motor Ship", Vol. XXIV, No. 280, May, 1943, pp. 46-55.*

Canadian Diesel-electric Ferry.

The Dartmouth Ferry Commission, Nova Scotia, have recently placed in service the double-ended Diesel-electric ferry "Governor Cornwallis". She is a wooden-built vessel 150ft. in length with a 51-ft. beam, and has a service speed of 12 knots. Being of the so-called "three-lane" type, the whole of the main deck is free from stanchions and hatchways, and therefore entirely available for the storage of vehicles. Cabins on either side of this deck provide seating accommodation for 137 passengers, while a saloon on the upper deck can seat 118 passengers. The ship's two main propulsion motors run at 350 r.p.m. and are supplied with current by four 100-kW. d.c. generators driven by 8-cylr. V-type Caterpillar Diesel engines which are controlled from the pilot house. All the auxiliary machinery is electrically driven, and includes two 30-kW. Caterpillar Diesel generating sets.—*"Gas and Oil Power", Vol. XXXVIII,, No. 452, May, 1943, p. 112.*

Post-war Diesel-electric Developments.

An article in a recent issue of the house journal of Brown, Boveri & Co., Ltd., the designers of the machinery of the Diesel-electric cargo vessel "Wuppertal", discusses some of the probable post-war developments of Diesel-electric propulsion. Among the advantages claimed for this type of machinery are: *(a)* the fact that the electric current needed for auxiliary and lighting purposes at sea can be taken from the main generators; *(b)* a substantial reduction in the length of the propeller shaft and shafts; and *(c)* the appreciable saving in machinery weight and space made possible by the employment of high-speed Diesel engines coupled to a.c. generators. The latter can, if necessary, be located over the propulsion-motor compartment, whilst another alternative is the splitting up of the Diesel-engined generators into separate watertight compartments to ensure additional safety. The arrangement of a 6,500-b.h.p. Diesel-electric a.c. generating installation with two Diesel-driven alternators is shown in Fig. 3, from which it will be seen that the control position is outside the engine room. Although the cost of the machinery of the first large Diesel-electric ship of the "Wuppertal" class (7,450 gross tons and 7,800 b.h.p.) was very slightly higher than that of the machinery of a similar vessel with direct Diesel drive, a saving of 7 per cent. is effected in the present instance by the use of 360-r.p.m. Diesel-engined alternators instead of units running at 250 r.p.m., the net result being a 5 per cent. reduction in the overall cost of the installation. In the case of owners who, owing to war losses, will have to rebuild a large part of their fleet, it is not only a question of considering the propelling machinery for a single vessel, as in normal times, but of providing the plant for a number of ships, possibly of varying size and power. By the adoption of Diesel-electric propulsion throughout, it would probably be possible to employ standard Diesel-engined generator units despite the higher total power needed. As a result, the following advantages would be gained: *(1)* A reduction in capital cost; *(2)* a saving in the number and value of the spare parts required; *(3)* simpler maintenance and repair service; *(4)* replacement of E.R. personnel simplified. For ships requiring different powers, it would only be necessary to vary the number of standard generators. The choice of the unit to be employed would of course, depend upon the types of Diesel generators on the market, but the size should be such that one, or at most two units would suffice to provide all the current required for auxiliary purposes in port. As regards spare gear, in the case of a normal direct-drive Diesel installation, the value of the spare parts carried on board is about 10 per cent. of the capital cost of the main engine, but the installation of, say, five Diesel-engined generators of identical type, makes it possible to reduce the cost of the spare parts to be carried on board to 3 per cent. of the capital expenditure on the propelling machinery. Such a reduction in the spare part list is allowed by the classification societies in ships having a number of similar Diesel engines. Furthermore, with standard propelling machinery installed in a number of ships, the cost of the spare parts required to be kept in reserve on shore would be substantially reduced. With the system outlined, it would be possible to transfer engineers from one ship to another at short notice and with complete confidence. It would also be possible to construct standard Diesel-engined generating sets more rapidly than to build large engines for direct drive for corresponding ships. Although the fuel consumption per b.h.p.-hr. of a high-speed Diesel engine is more than that of a slow-running unit, the employment of electric drive makes it possible to choose a propeller speed giving maximum efficiency. The electric ship can, moreover, be built as a single-screw vessel in powers when, for various reasons, such an arrangement might prove undesirable where a direct drive is utilised. In such cases the propulsive efficiency of the electrically-driven ship may be as much as 5 per cent. higher. As regards transmission losses in general, the electrical losses are 6-8 per cent., as compared with 4-4½ per cent. with electric or hydraulic couplings and mechanical gearing. The propeller shaft losses are lower with the electric drive, as from 6 to 12 bearings are eliminated and the loss per bearing is in the neighbourhood of one-third of 1 per cent. of the total s.h.p. The overall transmission losses with the Diesel-electric drive amount to between 6½ and 9 per cent. of the total s.h.p. developed. The fuel consumption of a vessel with this system of transmission approximates to that of the ordinary motorship with geared or direct drive, whilst at speeds below about 60 to 80 per cent. of full speed, the last-named system is definitely less economical than the electric drive. This is due to the fact that one or more generators can be cut out of operation, whilst the remainder run at maximum efficiency. Arrangements can be made during most voyages for the required average speed to be maintained by running at a lower speed for a certain period and for the remainder of the time at a speed above the average, thereby making it feasible to run the Diesel engines in actual use at their maximum efficiency. Apart from the foregoing advantages of the electric drive, there is the question of increased reliability, since, in the event of

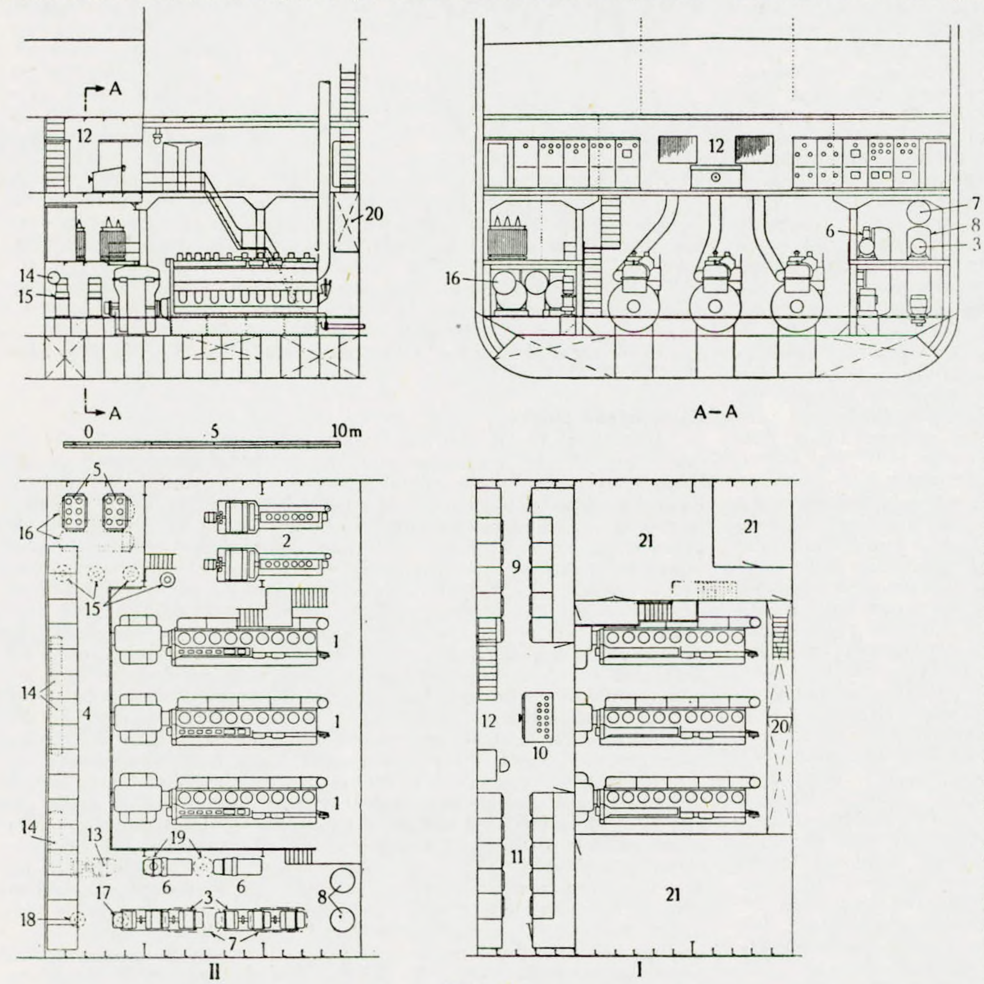


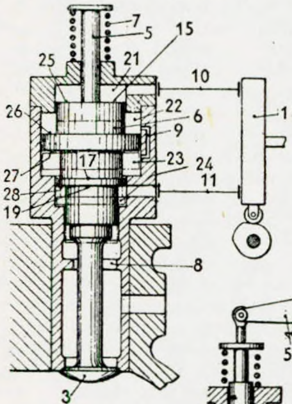
FIG. 3.

1.—Main Diesel generators.
2.—Harbour Diesel generators.
3.—Exciters.
4.—High-voltage switchgear for main generators, propulsion motors and transformers.
5.—Transformers.
6.—Compressors.
7.—Air reservoirs for switchgear.
8.—Starting reservoirs for Diesel engines.
9.—Switchgear for main generators, propulsion motors and transformers.
10.—Manœuvring position.
11.—Switchgear for harbour Diesel generators, and exciter and low-pressure circuit.
12.—Control room.
13.—Reserve oil pump.
14.—Oil cooler.
15.—Fresh- and sea-water pumps.
16.—Fresh-water coolers.
17.—Ballast pump.
18.—Drain pump.
19.—Sanitary and fire pump.
20.—Fuel reservoir.
21.—Compartments for auxiliary machinery, workshops, spares, etc.

the failure of one engine, the speed of the ship is but slightly reduced and the necessary repairs may be carried out whilst under way. On most voyages the requisite periodical examination and refit of an engine can be carried out at sea, so that the engineering staff are freed from such work when in port. Finally, the use of similar engines in a number of ships would enable the E.R. staffs of the latter to gain such experience as would enable them to carry out the necessary overhauls and repairs very rapidly. As the engines of a Diesel-electric ship continue running throughout the entire manœuvring period of the vessel, no cold compressed air is admitted into the cylinders, as is the case when starting or reversing the engines of a motorship with direct drive. This makes it possible to utilise smaller compressors and air reservoirs, while the wear on the cylinder liner is greatly reduced, giving a greater length of life to the engine. Present-day propulsion motors and generators being of the synchronous type, are both lighter and more efficient than the earlier asynchronous machines. Having a relatively large air gap (up to 6 mm.), they are more reliable, even in bad weather. For starting and reversing of the propulsion motors, a powerful asynchronous winding is provided, and this enables manœuvring to be carried out in the simplest and most reliable manner.—*"The Motor Ship", Vol. XXIV, No. 280, May, 1943, pp. 40-42.*

Sulzer Starting-air Valve for Marine Diesel Engines.

Sulzer Bros., Wintherthur, have developed and patented an improved type of starting-air valve for marine Diesel engines, which is claimed to eliminate the possibility of closure during the upward stroke of the main engine piston, whereby the air entrapped in the cylinder would be compressed to such an extent that the engine would be caused to rotate, say, in the ahead direction, notwithstanding the setting of the controls for astern. In the improved type of valve, two forms of which are shown in Fig. 2, only a part of the force tending to open the valve acts at the beginning of its opening, while the full force is maintained when the valve is to be kept open. A stepped piston *(6)* is connected to the starting-valve spindle *(5)*. The valve *(3)* is closed by a spring *(7)*. In order to prevent the valve from dropping into the cylinder in the event of breakage, the opening *(8)* is arranged eccentrically. When the valve is closed, the spaces *(22, 23)* are connected by a passage *(9)*, so as to equal the pressure. At the commencement of opening, control air flows into the space *(21)* and begins to force the piston *(6)* down, but the face *(25)* is of such dimensions that the valve will open against the compression pressure only and not against the higher firing pressure. When the edge *(15)* of the piston clears the casing, control air flows into the space *(22)* and the equalising passage *(9)* is closed, the space *(23)* being then connected to the compressed-air inlet pipe *(11)*. During the time that the starting valve is maintained open, the two piston surfaces *(25, 26)* are under the pressure of the control air and the total force is sufficient to ensure that the valve cannot be closed by the counter pressure when the engine is slowing down. When the valve is closing, the control air flows at first into the spaces *(23, 24)* as soon as the edge *(19)* of the piston has cleared the casing. The air begins to force the piston *(6)* up, thereby closing the valve *(3)*. The control air pipe *(10)* is then connected through the valve *(13)* with the atmosphere and the air in the spaces *(21, 22)* is therefore able to flow away. When the edge *(17)* of the piston clears the casing, the space *(22)* is also isolated from the space *(21)* and the pressure-equalising passage *(9)* brings the spaces *(22, 23)* into communication. The face *(27)* of the piston is then no longer under the pressure of the control air, which acts on the smaller surface *(28)* so that the closing force is reduced. In the alternative arrangement shown on the right, the stepped piston *(6)* is not connected directly to the starting-valve spindle *(5)* but is located in a separate casing *(49)*. Its movements are transmitted to the spindle *(5)* through a lever *(50)*.—*"The Motor Ship", Vol. XXIV, No. 280, May, 1943, p. 68.*



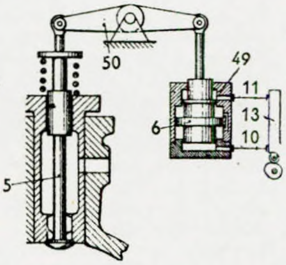


FIG. 2.

Centrifugal Blower Construction.

The impellers of centrifugal blowers may be either of the closed or open type, the former being more efficient but somewhat more difficult to manufacture, whereas open impellers have one side of the air passages clear and can, therefore, be drop-forged or die-cast. A recent British patent, of Swiss origin, covers an improved design of closed impeller made up separate parts which fit together without interrupting the smooth aerofoil shape of the vanes. Referring to the accompanying diagram (Fig. 3), the blower casing *(1)* is volute, and there is a gas-inlet opening *(2)* at one side, as well as an opening for the impeller driving shaft *(4)* at the other side. The impeller hub *(6)* forms the central part and there is a plate *(7)* having extended parts *(8)* of the vanes *(9)*. The other part *(29)* of the impeller comprises a ring *(11)* with an opening *(12)*. The impeller vanes *(13)* register with the corresponding parts *(8)*. It may be seen that the plate *(7)* and the ring *(11)* close the sides of the gas passages between the vanes so that only the inner and outer ends are left open. It is claimed that this method of construction allows the impeller to conform with the requirements of the closed type. The two parts of the impeller vanes have shoulders *(16, 17)* which resist the tendency of the part *(29)* to distort under the influence of centrifugal force at high speeds. The two parts of the impeller are secured to each other by means of bolts or rivets passing through a rim *(15)* which has a shoulder co-operating with labyrinth packing *(27)*. The thick part *(20)* of the plate *(11)* also has a shoulder which co-operates with packing *(28)*. The thickness of the vanes being greater from the centre towards the edges, the necessary allowance to enable their removal from the dies used in drop-forging is obtained.—*"The Oil Engine", Vol. XI, No. 121, May, 1943, p. 26.*

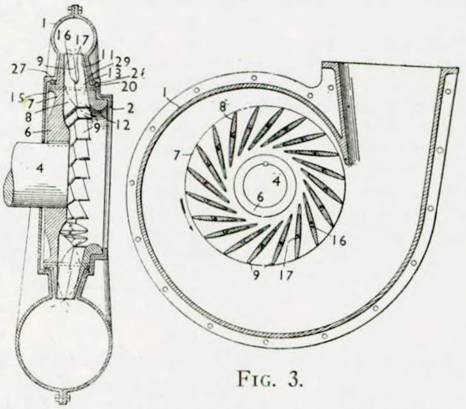


FIG. 3.

Gas-engined Ships.

In order to secure maximum operating efficiency of the machinery of the increasing number of gas-engined craft of relatively large size on the inland waterways of Germany, the employment of waste-heat recovery has been developed to a high degree. Steam is raised in an exhaust-gas boiler placed in the funnel, the gases being led direct from the engine to the boiler, whilst the feed water is heated by the circulating water from the cylinder jackets of the engine. Additional steam is generated by any surplus amount of heat in the gas producer. Two arrangements of this kind have been developed by the Humboldt Deutz concern. In one case there are two gas engines and a steam engine, the latter being used for reversing so that no reversing mechanism is needed for the gas engines. During the period of reversal the steam is raised by means of the gases in the gas generator. The second system provides for the use of a Lentz steam engine in conjunction with each gas engine, on a common shaft, the steam engines being used when reversing. An alternative scheme employs electrical transmission, the speed of the steam engine being independent of that of the two or more gas engines. It is reported that in a Rhine tug equipped with gas-cum-steam engines of this kind developing 700 b.h.p., the amount of steam raised is equivalent to 1·35lb./b.h.p., and a fuel economy of 36 per cent. as compared with the consumption of a normal gas-engined ship is claimed to be attained. This appears to be a remarkably high figure, since it is generally considered that with four-stroke Diesel engines the maximum amount of steam that can be raised at full output is not more than 1·01b./b.h.p. The gas engines in the Humboldt-Deutz arrangement are of high-pressure design and use 7 per cent. of ignition oil, the consumption being 14 per cent. lower than that of normal low-pressure gas engines.—*"The Motor Ship", Vol. XXIV, No. 279, April, 1943, p. 27.*

First B. and W. Gas-engined Ship.

The 2,000-ton motorship "Navitas", the first vessel to be propelled by the new type of Burmeister & Wain gas engine and producer, ran trials off Copenhagen on 26th February. She is a ship of 265ft.×42½ft., with a draught of 19ft. 5in. and a d.w. capacity of 3,030 tons. The propelling machinery consists of a 6-cylr. high-compression gas engine developing about 950 b.h.p. at 140 r.p.m. with a mean indicated pressure of just under 80lb./in.^{2}. The unit can be converted to Diesel-engine operation within a period of a few days in the event of the price of oil being more favourable than that of coal. In addition to the main engine there are two 100-h.p. gas-Diesel-engined generating sets and a third driven by a 50-h.p. engine. The ship's bunkers will hold sufficient coal for a voyage of 20 days, whilst for shorter voyages turf or wood may be used as fuel. The main and auxiliary machinery is arranged aft and there is a large funnel. The small auxiliary Diesel engine may be operated on tar oil obtained as a by-product from the gas-generator plant or by ordinary Diesel oil. Although the contract speed was only 10$\frac{3}{4}$ knots loaded, the ship attained a speed of 11$\frac{1}{2}$ to 12 knots on her trials, with the engine developing 1,480 i.h.p. at 146 r.p.m. It is stated that the fuel consumption at normal output is 0·72lb./ i.h.p.-hr. or 0·95lb./m.h.p.-hr. The ship's two main gas generators may be operated separately or in parallel, as desired. They are specially low in height and have rotating grates. In the lower part of each of the side bunkers is a coal-breaking machine fed direct from the bunkers, which is capable of handling the largest pieces of coal. The machine delivers the broken coal into a hoist by means of which it is passed through a sieve into storage bins, whence it is automatically fed to the producer. The ash and slag from the producer grates fall into electrically-operated ash hoists which discharge automatically overboard. The gas from the top of the producers passes to the cleaner, where dust and tar are removed by means of sea water. The clean gas is then delivered to a dryer, where the sea water is removed, and the gas is then ready for use in the engine. The water used for cleaning the gas is discharged to a settling tank, where the tar settles out and is led to a special tank for use in the auxiliary engine, whilst the water is pumped overboard. The gas generators are water-cooled to prevent the adherence of slag to the sides and a certain amount of steam is formed, which is drawn into the producer. Fresh water is distilled on board from sea water, the evaporator plant being supplied with the necessary heat by the exhaust gases of the main engine.—*"The Motor Ship", Vol. XXIV, No. 279, April, 1943, p. 20.*

Sulzer Frictional Slip Coupling.

An improved design of coupling which slips when the torque exceeds a predetermined amount has recently been developed and patented by Sulzer Bros., Winterthur. The construction of such a coupling is shown sectionally in Fig. 4. Fusible metal elements are provided to prevent the engagement of the frictional surfaces in the event of an undesirable rise in temperature, and the arrangement of the coupling mechanism is such that the torque is transmitted between the shaft (*1*) and a gear-wheel (*2*). There are two concentric rings (*A, C*), each having internal and external teeth, the coupling spider (*4*) being likewise provided with teeth which mesh with teeth (*19*) on the outer ring (*C*). The inner ring (*A*) has teeth (*17*) meshing with teeth (*18*) on the hub (*3*). The frictional parts (*5*) are of steel plate, whilst the fusible elements (*9*) are of zinc. The rings (*A, C*) engage the oblique teeth (*13, 14*) in such a manner that when a torque is transmitted it produces a force in the axial direction which tends to move the rings (*A, C*) oppositely, the ring (*A*) bearing on the face (*15*) and the ring (*C*) on the face (*16*) of the annular parts (*10*). With a normal torque the force acting on the rings is not sufficient to separate the annular parts against the action spring (*11*), but should the torque be great enough to cause the frictional surfaces (*5, 8*) to slip and generate heat, the temperature of the covering plates (*7*) rises. If the parts are not separated in a few seconds (say, between 30 and 50 revolutions of the shaft), their strength may be impaired to such an extent as to cause breakage. This is prevented by the fusing of the zinc parts, the liquefied metal being removed by centrifugal force. A clearance is then provided between the frictional elements, and the annular members (*10*) can live with their faces (*15, 16*) against the spider (*4*), thus relieving the pressure.—*"The Oil Engine", Vol. XI, No. 121, May, 1943, p. 26.*

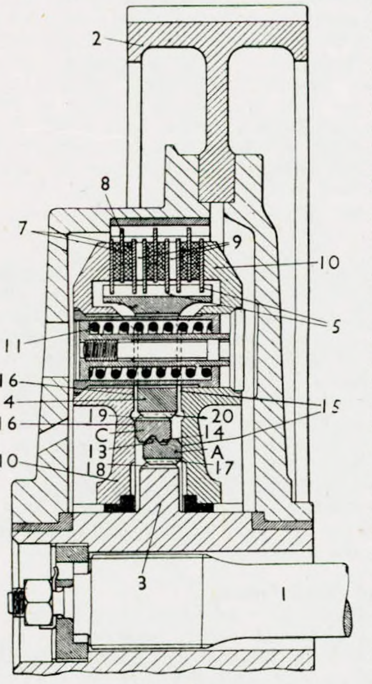


FIG. 4.

Gas Engines for Marine Work.

In a paper read at a recent meeting of the Institution of Mechanical Engineers by Mr. J. Jones, of the National Gas and Oil Co., the author stated that the thermal efficiency of the present-day high-compression gas engine is equal to that of a modern Diesel engine, but that it has not yet proved possible to construct an efficient form of gas producer using bituminous coal. When a producer of this type is evolved, it may prove feasible to employ engines supplied with gas from coal-fired generators for the propulsion of coastal craft of moderate size and with medium engine powers. By utilising dual-fuel engines of the type recently developed by Burmeister & Wain, the difficulty of getting a gas-engined vessel in and out of port would be overcome, since the engine could be run on oil and operate as a Diesel unit during the manœuvring period. One of the advantages of such an installation would be the additional amount of steam which might be raised by means of the exhaust gases. According to Major W. Gregson, the amount of steam which would be available would be of the order of 2lb./s.h.p.-hr., *i.e.*, about double the amount obtainable from a four-stroke Diesel engine. In the case of a small coasting vessel, it would, in fact, be possible to run all the deck and E.R. auxiliaries on the steam generated by the heat of the exhaust gases while the main engine was in operation. The main problem lies in the disposal of the producer tar. If the tar is washed out by the scrubbers, there will always be trouble with the authorities when in or near a port. The next step, therefore, would seem to be the development of a producer able to crack the tars into fixed gases.—*"The Motor Ship", Vol. XXIV, No. 280, May, 1943, pp. 37-38.*

Lubricating-oil Temperature Control.

The control of the lubricating-oil temperature of I.C. engines involves a number of conflicting requirements. Following a cold start, it is desirable to raise the oil temperature as quickly as possible, so that its viscosity may become suitable to give the best degree of "fling" and to form a freely flowing film. On the other hand, once the engine reaches normal operating temperature conditions, the oil should not get too hot. As the oil takes a great deal of heat away from the bearings and rotating and reciprocating parts of the engine, it is most desirable that this heat should be dissipated from the oil itself in order to prevent it from becoming too hot. This is an obvious application for thermostatic control, but as certain conditions have to be observed, such as prevention of back-pressure in the oil circuits and automatic adaptability to changes in operating conditions, many of the existing designs of thermostat intended for fluid control would be quite useless. A special design to meet the special requirements involved has been developed by a Gloucestershire firm of instrument manufacturers, the main function of the device being to by-pass the oil cooler when the engine is cold. When the engine is warmed up, the thermostatic device regulates the proportions of hot oil and cooled oil for re-admission to the engine, so as to give an operating temperature of a desired value, plus or minus a few degrees, under a wide variety of engine load and speed conditions. The thermostat is mounted in the oil circuit and has two oil inlets and one outlet, plus an alternative outlet. Of the former, one deals with hot oil direct from the engine, and the other with oil which has passed through the cooler. The outlet is common to both streams, and the thermostatically controlled shuttle determines the proportions of the hot or cool oil which are admitted. The two inlets are connected to opposite ends of a tube passing straight through the body of the thermostat. At the centre of this tube is a solid barrier, so that there is no possible straight flow through. Each stream is deflected radially through 90° by toroidal surfaces on the barrier, so that it has to flow into the body of the thermostat, through a ring of ports in the tube. As the total area of these ports is larger than the cross-sectional area of the tube, no appreciable back-pressure can be set up by the action of the toroidal deflector in changing the direction of flow of the oil. This arrangement allows oil to enter the thermostat body from both ends, the proportion admitted at each end being regulated by the controlling unit by means of a shuttle sliding on the tube and connected through a simple ball-and-socket mechanism to a lever forming part of an assembly containing a coiled bi-metallic thermostat element. The latter is located in the body of the instrument, immersed in the oil, and is therefore directly affected by the temperature. As the element expands or contracts in consequence of a rise or fall in the temperature, its effective length increases or decreases and so moves the arm and therefore the shuttle along the tube. If the oil becomes too hot, the thermostat expands and moves the shuttle so as partially to close the ports admitting hot oil and to open those admitting cool oil. The action is exactly opposite when the oil is cold, until it reaches the predetermined temperature. The selection of the working temperature is effected by the partial rotation of the thermostat element in relation to the cover of the device, a pointer on the spindle enabling the operator to observe the change in the setting with a high degree of accuracy. Although the instrument is not calibrated, this can easily be done by the user, if required. The makers claim that there is no possibility of any jamming of the shuttle through the passage of relatively large particles of foreign matter carried in the oil stream and that the operation of the lubrication circuit is not affected by mechanical damage of the device. In the event of a broken element, the setting can be arranged so as to give a fixed "full cool oil" position for the shuttle, thus entirely cutting out thermostatic control. Exhaustive tests of the device have shown that once the oil has reached working temperatures, it does not vary more than ±5° F., and that the working temperature is reached from cold more quickly than when manually operated cooler cut-out valves are used.—*"The Oil Engine", Vol. X, No. 119, March, 1943, pp. 282-283.*

The Influence of the Liberty Ships on Oil Prices.

By the end of 1943 the total gross tonnage of American merchant ships with oil-fired boilers will have been increased by over 16 million tons, and by the end of 1944, if conditions remain unaltered, this figure is likely to be about 23 million tons. Before the war, oil-burning steamships made up 30 per cent. of the world's shipping, and motor vessels represented about 24 per cent. By the end of 1943, bunker oil will be used in at least 45 per cent. of the world's tonnage, and by the end of 1944 the amount will probably be 60 per cent. In the meantime, the motorships, using the more expensive Diesel oil, will not make up more than the pre-war proportion of 24 per cent. It may, therefore, be anticipated that the demand for boiler oil relative to Diesel fuel will be doubled, and that the world price of bunker oil relative to that of Diesel oil will be less favourable than in the past. American plans to have completed over 2,000 merchant ships by the end of 1943, and another 2,000 are included in the recently issued programme. A large proportion of the first 2,000 will be "Liberty" ships burning 30 tons of fuel per day, whilst the numerous fast cargo liners will consume much more fuel. Considering only the "Liberty" ships and excluding war losses, each of these vessels consumes about 6,500 tons of oil per annum, so that 2,000 of them will require 13 million tons of boiler oil per annum. When the final series of a further 2,000 ships is completed, this figure will, of course, be doubled. Possibly this calculation is, to a certain extent, theoretical, for it is hardly tolerable that some thousands of ships should be allowed to continue in service burning at least three times as much oil as would be consumed by a corresponding number of motor vessels.—*"The Motor Ship", No. XXIV, No. 279, April, 1943, p. 4.*

War-time British Merchant Ships.

A number of photographs of typical merchant ships of various classes built in British shipyards during the present war have recently been released. Among the vessels illustrated is a Diesel-engined tanker of 12,000 tons d.w. with her propelling machinery and most of her accommodation aft and a high bridge forward. There is also a steam tanker of 14,500 tons d.w. with a high deck structure surmounted by a bridge almost amidships. It is reported that the new steam tankers include a high-speed type of vessel of 12,000 tons d.w., which has a different silhouette. Among the smaller classes of tankers are some vessels of 850 tons d.w. intended for coastal service, and some still smaller ones of 310 tons d.w. for the same purpose. These vessels have a low freeboard amidships, with an unusually high and long forecastle, to render them particularly suitable to operate in the short high seas they are likely to encounter. Among the special types of vessels illustrated is a steamer of 4,700 tons d.w. equipped with a considerable number of unusually powerful derricks and winches for the handling and transport of heavy weights. Another type of heavy-lift transport craft, not illustrated, is a steamer of 10,300 tons d.w. with the propelling machinery aft instead of amidships, as in the smaller vessel. A considerable number of colliers of various sizes have been constructed, and one of the photographs shows a collier of 4,300 tons with a massive bridge amidships and her machinery aft. Among the various classes of tugs built is a heavy-duty steam-driven type of 500 tons d.w., whilst the new coasting vessel include some Diesel-engined vessels of 320 tons d.w. with engines aft and large hatches served by two derricks. A still smaller type of coaster of 140 tons d.w., with a single pole mast and derrick, known in some districts as steam "puffers", has been built in some of our smaller shipyards, while other yards, employing a different class of labour, are turning out numbers of roomy reinforced-concrete barges of 200 tons d.w. Three examples of cargo vessels illustrated comprise a fast motor liner of 13,400 tons d.w. for refrigerated cargo, and a fast twin-screw Diesel-engined cargo liner of 10,300 tons d.w. A similar type of vessel, of the same deadweight, is specially equipped as a troop transport. Other types of motor cargo vessels of 10,300 tons d.w. include a number of ships of moderate speed designed to meet war-time conditions. One type of vessel, of 11,000 tons d.w., is propelled by reciprocating steam engines working in conjunction with exhaust turbines; this is a refrigerated cargo liner. In other cargo steamships of 10,300 tons d.w. the hulls are partly fabricated, welding having taken the place of riveting.—*"Engineering", Vol. 155, No. 4,029, 2nd April, 1943, pp. 267-268 and 270.*

Salvage of Scrapped Parts by Building-up.

As it is now necessary to preserve or recondition everything as far as possible, the writer has endeavoured to develop techniques which will: (*1*) add sufficient metal to replace that which has been lost; (*2*) not be over-expensive; (*3*) can be carried out with existing plant; and (*4*) does not take too long to carry out. The products involved were small, and it was not usually necessary to add much metal and so the processing time was usually short. The tests made covered: (*a*) metal spraying; (*b*) casting-on; (*c*) welding; and (*d*) electro-depositing. It was found that each process had specific applications. Metal spraying is very useful for the salvage of large areas for which only a small increase in the thickness of metal is required, but it is essential that the receiving surface should be free from grease and perfectly smooth before spraying. In an attempt to increase the bond on a small steel flange the latter was heated to redness, but the result was very unsatisfactory due to: (*1*) oxidisation of the metal surface; (*2*) burning of the spray metal as it was deposited; and (*3*) warping of the metal during coating due to the unequal contraction of the sprayed metal. It was eventually found that metal spraying was satisfactory and gave a fairly good bond for light loads and under alternating stresses if the surface was carefully prepared before by sand-papering. A batch of small castings was received with a porous core in one corner due to chilling of the mould. This defective part was therefore removed and the whole casting placed in an old metal mould with fresh runners and risers. The casting was then heated with a blowpipe and the defective surface raised to almost white heat, after which flux was thrown on to this surface and a fresh, clean melt was poured into this semi-mould. The results were highly satisfactory and the repaired castings stood up to shock tests very well. Soldering and welding may be regarded as being within the same category, except that the former process applies to the soft metals whilst welding applies to the harder ones. In both cases good adhesion was obtained and the results were very satisfactory. Electro-depositing proved to be the least satisfactory method of all. It took a long time to deposit a fair layer of metal on a small distributor tip, the current consumption was high, and the surface of the deposit metal was granular, in addition to which the adhesive strength of the layer deposited was poor.—*R. A. Collacott, B.Sc., "Mechanical World", Vol. 113, No. 2,936, 9th April, 1943, p. 390.*

"Hull Corrosion and Fouling".

The paper comprises a brief review of the work of the Marine Corrosion Sub-Committee of the Iron and Steel Institute and the British Iron and Steel Federation. The recently published *First Report* of this Sub-Committee contains an account of the work carried out up to the autumn of 1942, and includes full experimental details of many branches of the Sub-Committee's work. Some of the tables of data given in this paper are taken from the *Report,* and the author expresses the hope that the present paper may serve as an introduction to that more detailed document, and as a medium for the presentation of one or two additional topics. It is, however, mainly concerned with the fouling of ship's plates and methods of combating it.—*Paper by G. D. Bengough, M.A., D.Sc., F.R.S., read at a meeting of the N.-E. Coast Institution of Engineers and Shipbuilders on the 16th April, 1943.*

Electric Welding a Ship's Broken Stern Frame.

After a ship had been dry-docked, it was found that the after portion of the stern frame was displaced 1in. to starboard. The rudder was thereupon taken off and the fabricated rudder-post, with its streamlined plating, was dismantled. This left free the part of the cast-steel stern frame with the bottom pintle. The latter was taken away and machined at the break, whilst the remaining portion was cut by means of an oxy-acetylene torch, so that when the two parts came together again they gave a preparation resembling a double U in shape, the upper U being two-thirds of the total depth. The work was set up on the keel-blocks in the position it should occupy on completion of the repair, and reference lines were marked.

Point gauges were also made for the guidance of the welders. As it was necessary to use heavier electrodes in the upper U than in the lower (inverted) one, there would be a tendency for the heelpiece to lift; the latter was therefore preset a certain amount down before welding commenced. The weld metal was deposited underhand twice as fast as overhead, so that both U's were, say, half filled at the same time. Due to weld shrinkage, the pintle was lifting all the time, and to relieve the tension in the bottom weld the weld metal was peened between layers. The latter were deposited alternately, starting from the P. side and finishing at the S. side, and *vice versa.* The point gauges showed no lateral movement, whilst the vertical movement was controlled so that when the weld was completed the two parts were in a straight line and the point gauge dropped into all its marks. The rudder-post was then bolted up in order that the rudder might be tested on its pintles. The holes in the angles and plates forming the rudder-post fell opposite their corresponding holes in the bottom casting, including the tapped rivet holes, and the rudder swung freely. At the request of the surveyors the stern frame was strengthened at the weld by the addition of welded-on side fin-pieces, one P. and one S., and by welding on 2-in. plates at the top and bottom. The rudder was therefore unshipped again for the attachment of the fin-pieces and strengthening plates to the 19in. by 9in. section of the welded stern frame. The rudder-post streamlined plates were then riveted in place. Four arcs were used for welding the strengthening plates and fin-pieces, two on the P. and two on the S. side, in order to keep a balance, and the repair was then allowed to cool before flooding the dock.—*A. J. Louttit, "The Welder", Vol. XII, No. 82, July/December, 1942, pp. 305-306.*

Welded Ships' Lack of Flexibility.

The president of the American Bureau of Shipping, Mr. J. L. Luckenbach, attributes the break-up of the tanker "Schenectady" to a lack of flexibility which he considers to be an inherent feature of welded ships. This view is shared by Rear-Admiral H. L. Vickery, vice-chairman of the U.S. Maritime Commission, who states that cracks have developed in many welded hulls. Mr. Luckenbach declares that a mishap of the kind which befell the "Schenectady"would probably not have occurred in a riveted ship owing to the fact that there is a certain amount of flexibility in a riveted hull which has not yet been put into vessels of all-welded construction, but that further study of the welding process will eliminate this defect.—*"The Journal of Commerce" (Shipbuilding and Engineering Edition), No. 35,925, 1st April, 1943, p. 7.*

Application of Welding in Submarine Construction.

An article by E. H. Ewertz and R. D. West in the January issue of the American *Welding Journal* describes the methods and procedure adopted by the Manitowoc Shipbuilding Co., for the construction of submarines. The main hull structure with its internal foundations and attachments is welded throughout, except for certain removable plates required for the installation of the machinery and for certain portions of the superstructure. The close tolerances and strength requirements specified call for a high quality of workmanship on the part of the welders and ship fitters engaged on the hull work. In order to minimise distortion and locked-up stresses, a suitable sequence of welding operations has been evolved. The sub-assembly system adopted permits: *(1)* the correct positioning of a considerable portion of the welding work and the use of large-sized electrodes of the "hot rod" type"; *(2)* greater accuracy in fitting together the work on the part of the assemblers, who work from jigs or templates; *(3)* the employment of installation moulds; *(4)* straightening operations to be carried on smaller and more readily handled units; *(5)* the locked-up stresses in the completed structure to be reduced to a minimum because the final assembly welding or attachment comes last; and *(6)* minimising cumulative shrinkages. The facility requirements as affected by welding are as follows: *(a)* as loads up to 65 tons have to be handled, large building-berth crane capacity is essential, but where side launching is resorted to, crawler crane equipment may be utilised instead of the more expensive overhead or track-mounted types; and *(b)* adequate equipment for transporting bulky and relatively top-heavy sections between the assembly shop and the building berths must be provided. A special wide-tread crawler carrier was found to be satisfactory for this purpose. The article gives a summary of the hull shop requirements. In conclusion, the authors state that the lay-out of the welding equipment should be such as to provide the maximum flexibility at the berths, since a full welding crew is not required for the entire period during which a hull is on the slip. Shop welding takes up about 60 to 65 per cent. of the total welding man-hours, and berth welding 35 to 40 per cent., so that the equipment requirements can be determined accordingly. There is very little scope for the use of automatic welding equipment. The article is illustrated with 16 photographs. —*Abstract (No. 1412) in "Welding Literature Review", Vol. V, No. 2, May, 1943, p. 130.*

Welding of High-tensile Steel.

Some interesting remarks regarding the welding of high-tensile steel were recently made by Mr. W. G. John, Superintendent of Welding Development (Naval Construction), Admiralty, in a lecture at Stowe College, Glasgow. He expressed the view that the all-welded ship will, in due course, become a standard production, since there is no major problem to be solved which limits the extension of welding to mild steel. The welding of high-tensile steel of high carbon content is, however, a more difficult matter. The main longitudinal structure of a warship is usually built of high-tensile steel, and it has become the practice to rivet most of the main longitudinal items and to weld the others, although welding has been adopted for the forward 80 or 100ft. of all modern cruisers, where it has been found to be very satisfactory. The danger which exists when high-tensile steel is welded is that a hardened zone may be formed in the parent metal adjacent to the weld, which may crack during the cooling of the latter or at a later date. The formation of this hardened zone depends on the carbon or other alloy content, and types of high-tensile steel known as "D.W." and "S" have now been developed, in which the carbon content has been reduced to avoid this difficulty. Some slight sacrifice in the ultimate tensile strength has had to be made on this account, but this is counterbalanced by the improved weldability of the material. Mr. John gave some details of the procedure involved when welding steel of this variety. Armour and protective plating can also be welded, but in such cases special electrodes have to be used, and measures, such as pre-heating, adopted.—*"Fairplay", Vol. CLX, No. 3,132, 20th May, 1943, pp. 584-585.*

Refrigeration in Warships.

While it is not permissible, when describing the refrigerating installation of a certain type of warship much in the news at the present time, to give any details of the vessels concerned, it may be said that this particular plant has a capacity of 13,000 B.Th.U./hr. and is fitted with brine cooling. There are two cold rooms; one of 400 cu. ft. capacity can be lowered to 10° F., while another chamber normally maintained at 25° F., can, if necessary, be cooled to 16° F. The refrigerator is of 150 cu. ft. and is also capable of producing 120lb. of ice per day in two batches of 60lb. The methyl chloride compressor is of the vertical type and runs at 375 r.p.m., belt-driven by a 5-h.p. electric motor. The entire plant is automatically controlled, the thermostat being placed in the brine tank. The condensing tubes are of special type. The evaporator brine tank is fitted with four 15-lb. moulds for ice-making, while brine is pumped by a ½-h.p. motor-driven pump from the evaporator through the coils into the cold room, in which the piping is arranged on the walls and ceilings. There is also a brine coil in the fresh-water tank, which has a capacity of 20 gallons. In certain other warships of recent construction, D.E. plant of 6,000 B.Th.U./hr. capacity is installed, mainly for meat storage. The cold room is maintained at 16° to 18° F., and there is provision for making 10lb. of ice per freezing. The compressor is a methyl chloride twin-cylinder model, running at 375 r.p.m., belt-driven by a 2½-h.p. motor with automatic control gear. The tubular condenser is cooled by sea water, and the direct-expansion coils are arranged on the walls and ceiling. The cylindrical ice tank has circular coils and two 5-lb. circular moulds.—*"Modern Refrigeration", Vol. XLVI, No. 541, April, 1943, p. 75.*

The Failure of the "Schenectady".

The report of the sub-committee appointed by the American Bureau of Shipping to investigate the cause of the failure of the all-welded oil tanker "Schenectady" has now been published. This vessel, the first to be built at the new Swan Island yard of the Kaiser Company, was of the same design as the 23 tankers already completed by the Chester yard of the Sun Shipbuilding and Dry Dock Company. She was launched on 24th October last and completed satisfactory sea trials. On the evening of Saturday, 16th January, while the "Schenectady" was lying in the fitting-out basin of the yard, she broke in two with a loud report; fracturing suddenly across the deck at a point just abaft the after end of the bridge, *i.e.*, about amidships. The fracture extended down both sides to the bottom shell plating, which remained intact. All the deck, side and bottom longitudinal frames fractured, as well as the plating of the corrugated longitudinal bulkheads and the centre-line deck and bottom girders, but in no case did the fractures occur in the tranverse welds. The ends of the ship settled in the silt so that there was a gap of about 10ft. at the deck. This structural failure has been attributed to a combination of circumstances, but the report of the sub-committee considers that one of the principal causes was the "abnormal amount of internal stress locked into the structure by the processes used in construction". The report sets out briefly the conditions at the time of the fracture, from which it would seem that the fore-peak tank, the forward deep tanks, and the after-peak tank were full of water as ballasted for the trial trip, and that the cross bunkers immediately forward of the machinery space contained about 3,100 barrels of oil fuel. The resultant bending moment in hogging in still water was about one-half of the maximum design bending moment. It produced a tensile stress on the upper flange of the hull girder of approximately $4\frac{1}{2}$ tons/in.^{2}, which would certainly not have been sufficient in itself to cause any failure in the hull structure. The air temperature at the time was 23° F.—having fallen from 38° F.—introducing the fact of increasing brittleness of the steel. There was no evidence of an alleged bank of silt having been washed under the midship portion of the vessel. The report emphasizes that there was no question as to the sufficiency of the ship's structure, since there was no departure from recognised and proved design standards. All the factors affecting the failure, including structural design and methods of construction, were thoroughly investigated, and a complete physical, chemical and microscopic examination of the material and the welding in way of the fracture was carried out. The report is, however, severely critical of the workmanship in the shipyard, and points out that there was a tendency on the part of the yard personnel to depart from recognised fundamentals of good welded construction in the interests of rapid production. There were insufficient numbers of experienced welders and ship fitters available for the job at the rate of construction maintained, and an inadequate number of skilled welding supervisors with the necessary knowledge of the basic elements of good welding practice to exercise proper control over the welders. Furthermore, "there was neglect on the part of the personnel to realise the importance of adhering rigidly to established welding procedures and welding sequences necessary to reduce shrinkage stresses to a safe minimum". In the case of sister ships under construction, the report states that there was evidence of poor fitting of large sub-assemblies which made it necessary to have recourse to an excessive amount of jacking, etc., to force them into position. In other cases, open joints required the use of an excessive amount of welding to finish the joint, resulting in excessive shrinkage. A "serious accumulation of shrinkage stresses" also occurred in the automatic machine welding of the deck assembly joints, especially of the longitudinal joint at the gunwale attaching the sheer strake to the stringer plate. This particular welded joint was also found to be defective in way of the location where the failure started, as evidenced, not only by a longitudinal crack, but also by what appeared to be minute transverse cracks in the weld. The abrupt termination of the bridge-end fashion plates at the top of the sheer strake also constituted a serious point of stress concentration, which was augmented by the hogging stress due to the ballasted condition of the ship. The accumulation of the abnormal amount of internal stress locked into the hull structure together with the concentration of stress caused a tensile failure at the starboard sheer strake which was formed of steel of sub-standard quality, all of which was aggravated to some extent by the drop in atmospheric temperature. The defective welding was removed from the "Schenectady" and her sister ships, and severe bending tests were applied after the joints had been properly rewelded. The results were satisfactory. The committee have expressed the opinion that "the high stresses responsible for the fractures that have recently occurred in welded ships" was due primarily to the failure to adhere rigidly to the procedure necessary to keep shrinkage stresses within the margin of strength allowed in the design. Steps have been taken to ensure adherence to proper established procedure, and it is added that the managements of the yards are fully alive to their responsibilities in the matter. "The committee feel that closer control of welding procedure, in which the builders have already been instructed, will prevent a recurrence of such major failures". Finally, the committee suggest that the probability of fractures resulting from residual stresses in welded construction will decrease with the length of service.—*"The Shipping World", Vol. CVIII, No. 2,601, 21st April, 1943, pp. 415-416.*

Severe Hull Tests of Tanker "Schenectady's" Sister Ship.

The 16,500-ton d.w. turbo-electric oil tanker "Quebec", recently completed at the Kaiser Co.'s Swan Island yard, Portland, Oregon, is a sister ship of the "Schenectady" and is the second tanker to be built there. In view of the mishap which befell the "Schnectady" while fitting out at Swan Island, the U.S. Maritime Commission ordered the all-welded hull of the "Quebec" to be subjected to a series of unusually rigorous tests to make certain that her design and construction were quite free from the defects which caused her sister ship to break her back. After completion of the sea trials of the propelling and auxiliary machinery, including a highly satisfactory full-power run of six hours' duration, the "Quebec" returned to the yard to undergo a sag test during which the midship tanks were filled with water ballast whilst the forward and after tanks remained empty. This was followed by a hogging test for which the fore-peak, deep tank, cofferdam, feed and after-peak tanks were filled with water whilst the midship tanks were pumped dry in a deliberate attempt to break the vessel's back. However, the midship part of the hull rose upward a few inches, only slightly more than it had deflected downward during the sag test. It was estimated that the stress imposed on the hull structure by the buoyancy of the water under the midship bottom plates and the downward pressure at either end, due to ballast, machinery, fuel and stores, amounted to a bending force of about 305,000ft.-tons. This, it is stated, is 165 per cent. more than the strain which caused the "Schenectady's" hull to crack, and it is not anticipated that any tanker in normal service would be subjected to such excessive stresses and strains. It is reported that the remaining tankers of this series built by the Swan Island yard have undergone or will be subjected to similar hull tests prior to delivery.—*"The Log", Vol. 38, No. 4, April, 1943, p. 78.*

Propellers in Recesses Amidships.

An arrangement of propellers amidships, which forms the subject of a recently published British patent, is shown in Fig. 3. The hull (A) has a recess or channel (B) at each side, into which shaped casings (C) project. The propeller shafts (4) extend through these casings and the propellers (E) are closely spaced. They are preferably designed to run in opposite directions. The drive to the propeller shafts is effected by means of electric motors (G) for which the current is supplied by Diesel-engined dynamos or turbo generators (H). In the latter instance it is necessary to arrange for the provision of boilers (I) to supply steam to the turbines. The propulsion motors may be installed either in the engine room (F) or in separate compartments.—*"The Motor Ship", Vol. XXIV, No. 280, May, 1943, p. 68.*

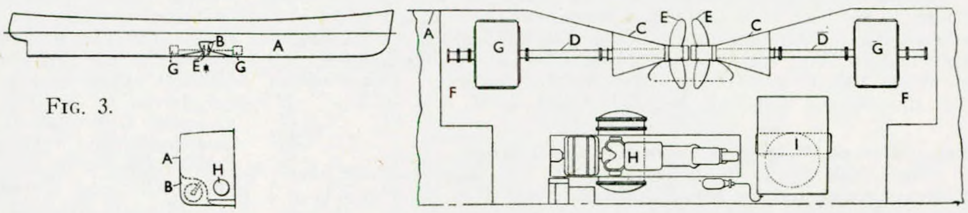


FIG. 3.

Airscrews for Ships.

The report that the Axis are using shallow-draught barges equipped with airscrews instead of water propellers in the Sicilian Narrows, is not devoid of interest. Airscrew propulsion for slow-speed craft of this kind is not particularly efficient, since the thrust per unit disc area must be kept to something like $\frac{1}{800}$ (the ratio of the fluid densities) that for an ordinary propeller of similar efficiency, and this involves a diameter which is hardly practicable. Even accepting a reduced efficiency, the diameter of the airscrew must, however, be inconveniently large, and it becomes necessary to gear down the high-speed type of engine used in order to avoid excessive tip speed and allow a reasonable pitch ratio and slip. Nevertheless, in the case of very shallow-draught craft employed on short sea crossings, there may be some scope for the air drive. Variable-pitch —or even reversible pitch—screws would be advantageous for manœuvring, but it is probable that the governing consideration was the availability of numbers of serviceable aero engines no longer suitable for use in modern fighting aircraft.—*"Shipbuilding and Shipping Record", Vol. LXI, Nos. 15/16, 22nd April, 1943, p. 343.*

ERRATA.

Through an unfortunate error the following illustrations in the preceding issue appear inverted :—

An American Floating Dry Dock, Fig. 4, page 53.

Lighter Poppet-type Steam Engine Valves, four illustrations unnumbered, page 60.

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