CG 415/77

THE INSTITUTE OF MARINE ENGINEERS

Transactions

Modern Bearing Design and Practice

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Reprinted from Vol. 88, 1976

Paper read at the Institute on Tuesday, 30 March, 1976

MODERN BEARING DESIGN AND PRACTICE

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The paper is a review of bearing practice with particular reference to marine machinery. It attempts to define the factors relevant to any bearing system, and then briefly describes the latest developments and trends. The final section deals with some problem areas and the diagnosis of the causes of observed damage.

INTRODUCTION

The subject is a vast one and quite impossible to cover, even superficially in a single paper, and when the author was asked to give a review of modern bearing design and practice, it proved rather difficult to decide what to include and what to omit.

Bearings are used in all engineering fields including structural. Development proceeds steadily and instances of dramatic development, such as the introduction of the tilting pad bearing in its day, are infrequent.

It was decided, therefore, to limit this review to sliding surface bearings, and to those developments of immediate interest to the practising marine engineer. Even so, it is not possible to deal with bearing theory in depth, and the author has indicated developments broadly, so that, once apprised of their existence, interested parties can follow up the specific issues independently. A great mass of bearing theory has been published in text books and in papers which are there for the reading. The translation of theory into practical application is, however, just as important and this is where experienced judgement is still necessary.

A section on bearing damage diagnosis has been included, since this is not only of interest to the operator, but is of great importance to the designer of the machinery. It is unfortunately still true that much more is to be learnt from failures than from their absence.

Historically, the rate of machinery development has often been dependent upon the state of the bearing art, and in some cases has been delayed until suitable bearing developments had taken place. An example is the way in which the development of geared steam turbine machinery was for some years inhibited by the lack of a suitable thrust block, so that it only became a practical proposition with the invention of the tilting pad thrust bearing by Michell and Kingsbury. And the development and uprating of internal combustion engines has to a large extent depended upon the development of suitable bearing materials, as well as design understanding.

The bearing specialist provides a design and manufacturing service to machinery builders, but the performance of his product is largely determined by circumstances over which he has little control. He must be continually developing a fundamental knowledge of design, materials and manufacturing techniques to meet the changing and ever more onerous requirements of the machinery builder. To do this he must maintain close touch with machinery development, and must attempt to forecast the future needs of the industries he serves, perhaps before they themselves are aware of them.

In recent years bearing development has tended to proceed rather steadily in staircase fashion with moderate upward steps. Any dramatic advance tends to take so much longer to gain acceptance that the overall rate of progress is not much changed. This is inevitable because machine makers tend to be wary of new developments; which is hardly surprising when one considers that the cost of damage to a machine from a bearing failure is often out of all proportion to the value of the bearing.

Probably the most noteworthy development in recent years is in the use of the computer for bearing design. Its use has made design work possible in a way which could never have been undertaken by manual methods.

In bearings themselves, perhaps the most noteworthy development in recent years has been the introduction of externally pressurized, sometimes termed hydrostatic, bearings which have found useful applications in machine tool slideways and for precision spindles.

THE BEARING SYSTEM

By definition, a bearing is a support or guide, by means of which a moving part is located with respect to other parts of the mechanism. But the bearing, as it is commonly understood, is only one component of an complex interactive system, the other two being the co-operating surface, usually a shaft journal or thrust face, and the lubricant, all within the influence of their environment. Each of these has its own properties and features which interact with the other two so that successful operation depends upon each component behaving satisfactorily in the presence of the other two. Since it is impossible to discuss bearings without reference to the other factors it may be useful to look at the characteristic properties in the whole system, and Fig. 1 is an attempt to define this.

The composition of both surface materials has been included because it seems fundamental. In reality, however, its importance from the bearing viewpoint is in the properties it confers — Item 2 *et. seq.*

THE CO-OPERATING SURFACE

This is usually, but not necessarily, a shaft journal or thrust face, the necessary properties from the bearing aspect being listed in Fig. 1.

The material must be sufficiently strong to withstand the forces applied and the surface bearing stresses without deformation. Since the material is chosen for its ability to accept the normal operating stresses in the machine, this strength requirement is usually met automatically, and it may be easily overlooked in those cases where it assumes major importance. Typical examples are where a very thin tubular shaft may be used, or where a shaft is sleeved with a different material for corrosion resistance or compatibility reasons.

Hardness is important, chiefly to provide wear resistance, particularly where operating with a hard bearing material. It is usually advantageous to have the shaft hardness much higher than that of the bearing in order to confine the wear to the cheaper component, which is the bearing. A hard surface confers side benefits in that it minimizes risk of seizure, in part because it assists the production of a high grade surface finish, and it reduces the severity of damage if seizure should occur.

It is important that the shaft surface be compatible with the bearing surface and with the lubricant, and this point will be dealt with in greater detail later.

The shaft surface must resist corrosion resulting from the lubricant or the environment. Where oil is the lubricant this is normally taken care of fairly automatically, although entrained water can, at times, cause problems, particularly where machinery is laid up for lengthy periods. Where other fluids are used, such as sea water, then there is the added risk of electrolytic attack between the shaft and the bearing surface or the close environment. The shaft surface material should be selected so as to be preferably non-corroding or at least cathodic in this situation.

Under the term "geometric form" are grouped the mechanical features.

* The Glacier Metal Company Limited.

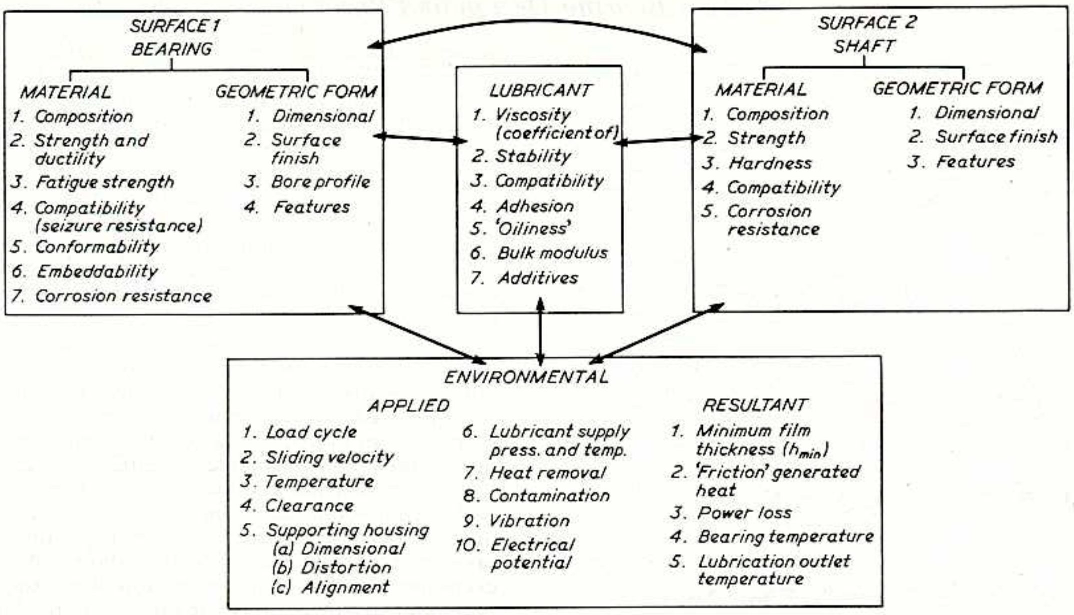

FIG. 1—*The bearing system—Interaction of components and the environment*

In early days the bearings were usually matched to the shaft by scraping and bedding in so that slight imperfections of size or geometric shape were not important. However, with the requirement for interchangeability and the necessity to avoid fitting on site, it is necessary to work throughout to much closer limits.

Dimensional accuracy must be such as to allow interchangeability and the maintenance of acceptable variations in clearance, taking into account the stackup of all tolerances.

But in addition to being to the correct size, the shaft journal must be cylindrical and its axis coincident with the shaft axis. Its surface should not be conical, out of round, barrelled, waisted or wavy. The axis of the journal should not be eccentric or wobbling relative to the shaft axis.

Thrust faces must be flat and normal to the axis. They should not be either convex, concave or have a wobble.

The usual convention is that the envelope described between upper and lower limits of the dimensional tolerance also describes the permitted geometrical variations. In the case of bearing surfaces, this is frequently not fine enough and it is advisable to specify the geometrical form to closer limits than the actual dimensions. For example, a shaft journal of 250 mm diameter may reasonably have a tolerance of ·05 mm, i.e. 2 in 10 000, whereas the geometrical limits for out-of-roundness, taper, etc. should be not more than 1 in 10 000 and probably less. Another typical example may be a thrust collar where the tolerance on thickness could well be 1 mm, but where tolerance on flatness should be within ·01 mm. It is impossible to generalize because the geometrical accuracy depends upon the application, and where very thin oil films are involved the accuracy must be commensurate.

Fortunately, the realization of suitable geometric tolerances can often be achieved without much difficulty by the choice of suitable machining techniques, whereas dimensional limits depend upon the skill in producing and measuring to size.

Designers should always establish the broadest limits which are acceptable, in order to reduce production cost, and therefore it is often beneficial to determine both dimensional and geometric tolerance limits. All too often a designer will specify a close limit on diameter, leaving the rest to chance, although it may be acceptable to specify a somewhat broader limit on diameter with suitably tighter geometric limits.

Another very important related characteristic is that of surface finish, usually defined by a roughness value. For convenience, the latest ISO values are compared with the familiar C.L.A. values in Table I. Surface finish is especially important where bearings must accept starting under load, or where the operating minimum oil film thickness is very small. It is not possible to generalize, since so much depends upon the application and loading conditions, but in practice suitable attainable values have become established by experience for many types of application.

However, roughness average cannot be used as the sole criterion and there is a need for the development of a further designation of surface finish, perhaps best described as the texture. Surfaces produced by different machining methods will have different textures, although they may have the same value of roughness average.

TABLE I — COMPARISON OF SURFACE ROUGHNESS TERMS

ISO Standard Roughness Grade Number	Roughness Average* Ra		Approx. R max or Rt Value **	RMS Value Ø
	μ m	μ inch	μ m	μ inch
N12	50	2000		
N11	25	1000		
N10	12·5	500	40	550
N9	6·3	250	25	275
N8	3·2	125	14	138
N7	1·6	63	8	70
N6	0·8	32	4	35
N5	0·4	16	2·5	18
N4	0·2	8	1·6	9
N3	0·1	4	0·63	4·4
N2	0·05	2	0·4	2·2
N1	0·025	1	0·25	1·1

μ m = micrometre = ·001 mm

* Prior to international standardization this was termed CLA (Centre Line average). It is the arithmetic average value of the departure of the profile both above and below its centre line over the measuring or "cut-off" length.

** Previously known as Peak-to-valley. This is a measure of the total depth of surface irregularities within the sampling length.

There is no direct relationship between Ra and R max. The ratio depends upon the finishing method used and the values of R max shown are only typical approximations for comparison.

Ø This is the S.A.E. standard used in the USA.

Fig. 2 shows at (a) and (b) two typical turned surfaces and at (c) a typical ground surface of similar roughness. Observe that the crests in (a) are much sharper than (b) although both exhibit a similar pattern of periodicity. The

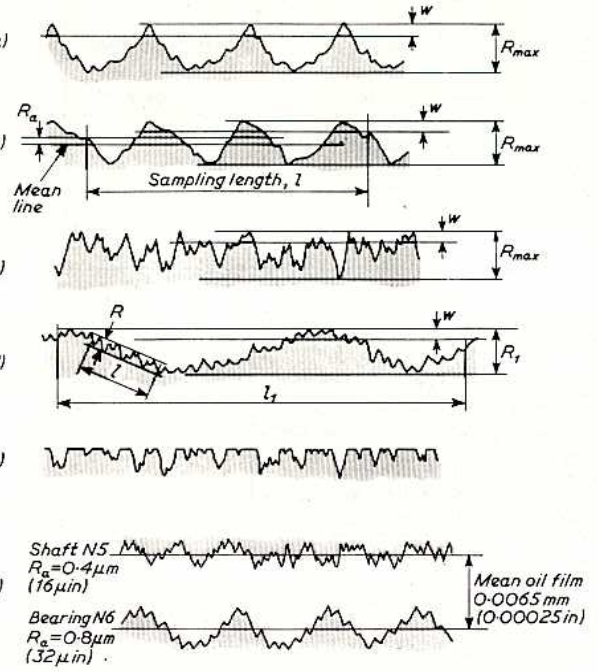

FIG. 2—*Surface profiles*

ground surface (c) is typically random but with quite sharp peaks and valleys. A similar amount of wear, shown as (w), by truncating the peaks will produce a totally different surface pattern, the resulting plateaux occupying a greater proportion of the surface in the case of (b) and (c) than (a).

In measuring surface finish it is important to take a sufficient sampling length. A typically wavy surface is indicated at (d). If a short sampling length, l, is taken, then the surface roughness reading may well appear as R. If, however, a length l_1 is taken, corresponding to at least one wave length, then the true roughness value of R_1 will be obtained. Note again the result of a similar amount of wear (w).

The effect of running-in is to truncate the peaks by the wear process, abrasion and minute seizures paring away the crests and the resulting debris being washed out with the lubricant. The mechanism usually occurs at the highest rate when the oil film is very thin and during the starting and stopping cycles, when boundary lubrication pertains and there is no true oil film to separate the surfaces. If the peaks are disproportionately high, then the wear process cannot occur gradually enough to avoid disaster and major seizure is likely to occur. It follows then that the surface roughness of both shaft and bearing must be arranged to suit the conditions of the application, and particularly starting and stopping cycles. At (e) is shown a typically run-in surface. This can be produced artificially by the machining method adopted, i.e. by lapping, honing or super-finishing, usually carried out after a fine grinding operation, for the sake of production speed. Lapping or super-finishing will produce roughness values from 0·2 μm down to an optical finish < 0·025 μm. Commonly 0·1 μm (4 μ inch) or better can be achieved on hardened shafts.

It may be useful to comment here that it is reasonable to believe that it is possible to attain too good a finish. The author recalls a nuclear pump thrust bearing, fitted with carbon faced pads, to run on water. The collar faces were lapped to an optical flatness with a mirror finish of about 1 micro inch. Considerable trouble was experienced with scuffing, which was finally narrowed down to the starting up cycle. After some hesitation it was decided to break down the high finish on the thrust collars by regrinding and lapping to produce a surface similar to that shown at Fig. 2(e) and to everyone's relief no further trouble was experienced. In this particular instance the collar faces were of stainless steel and it is possible that it was partly an adhesion or wettability problem.

Except in the case of engine crankshafts where there are commonly drilled holes for the transfer of oil to or from the bearing, it is rare to have any features in the shaft surface. There are, however, a few occasions when an oil feed groove in the shaft is requisite, such as for example where the load vector is fixed relative to the shaft, i.e. a rotating load situation, or a stationary shaft with a rotating bearing.

THE LUBRICANT

The lubricant may be a fluid, liquid or gaseous, a grease or a solid. However, the great majority of the bearing applications being discussed here are oil lubricated: to which this review will be restricted. The properties of the lubricant from the bearing aspect are shown in Fig. 1.

The property most used in bearing design is that of viscosity, a property very dependent upon temperature. Fig. 3 shows typical grades of oil and the variation of viscosity with temperature.

Where machinery must operate in very low ambient temperatures, it becomes necessary to consider the pour point of the oil, to ensure that it will be fluid before the machinery is started up. It may be necessary to incorporate heaters to ensure this.

Stability of the oil is important in that it should retain its

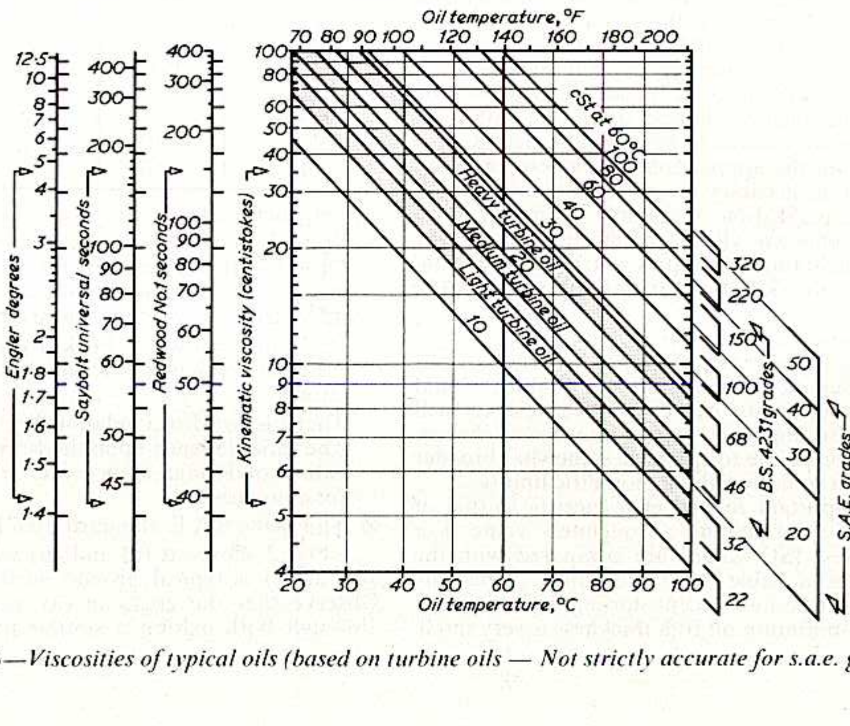

FIG. 3—*Viscosities of typical oils (based on turbine oils — Not strictly accurate for s.a.e. grades)*

properties over lengthy periods of service. Mineral oils are fairly stable but when exposed to high temperatures they become acidic due to oxidation. Stability can also be affected by contamination from the environment.

Compatibility with the bearing, the shaft and the environment is also an important characteristic of the lubricant, and fortunately most oils are compatible with the general run of the surfaces used.

Adhesion and "oiliness" are properties which are important, but little understood. It is probable that many bearing problems are due to lack of these properties; they are likely to be related and are molecular in nature.

The classical theory of lubrication on which nearly all design calculations are based assumes that the oil adheres completely to both moving and stationary surfaces, as a result of which a film of oil is drawn in between the surfaces so that they become separated by the entrained film of lubricant. Except in very high speed applications which require special treatment, the flow of lubricant is usually assumed to be laminar with a uniform rate of shear across the thickness of the film. It is the rate of shear, coupled with the viscosity which determines the drag, or apparent friction, and hence the power loss in the bearing. If one considers adhesion to the moving surfaces to be zero, then no lubricant would be drawn in and the bearing could not operate. Conversely, if one considers adhesion to the stationary surface to be zero, then the rate of shear would diminish to zero and the surfaces would be separated by a film of lubricant which was generating no heat.

In practice neither of these extremes is likely to happen but it is probable that they do occur on a partial scale, and this could well be the mechanism which accounts for the poor behaviour of certain shaft materials. The phenomenon is sometimes referred to as "non-wettability".

By "oiliness" is meant the ability of certain oils to prevent scuffing and seizure under boundary lubrication conditions. The vegetable and animal derived oils were generally high in this property, whereas unmodified mineral oils are sometimes relatively low. Other fluids such as water, possess no oiliness and therefore will not prevent seizure between metallic surfaces under boundary conditions. It is then necessary to build a property into one of the surfaces by choosing a suitable material.

Classical theory ignores the bulk modulus of the lubricant and so long as this is a liquid, the error is negligible, except in the case of very high specific loads much higher than are normally encountered in the marine field. However, it may be noted that the bulk modulus of the lubricant can be modified if there is extreme entrainment of gas, i.e., foaming, in the circulated lubricant, when the resulting greater compressibility will result in thinner oil films and may result in breakdown.

Nearly all modern lubricating oils are based upon mineral oil, the characteristics of which, however, can vary with the area of origin. Additions of various chemicals are made in order to modify the oil and improve the desired property. Thus, there are additives to lower the oxidation rate, increase the rate of settling out of entrained gas, hold carbon particles in suspension, provide longer life before acidity becomes too high, and so on. This is the area of the oil expert and there is neither the space to discuss it here, nor has the author the competence. The object has been simply to alert readers to the aspects of the lubricants which are important.

THE BEARING

There is rarely complete freedom of choice and therefore the bearing design must be a compromise. The co-operating surface, the lubricant and the environment all place constraints on the bearing so that the bearing designers' freedom of choice is often limited to material and geometry.

The basic properties required for a bearing material are summarized in Fig. 1.

Some of these requirements are difficult to quantify. In order to reduce the number of subjective judgements which have to be made, tests of an empirical nature have been devised which, although they give reliable relative results, are difficult to correlate in absolute terms or even to compare with similar tests carried out by different investigators.

It is axiomatic, of course, that the material shall be practical; that is, it can be produced in a suitable form, to consistent quality standards and at an acceptable cost.

The composition of the bearing material determines its properties to a very great extent and Table II shows the range

TABLE II—COMMON LINING BEARING ALLOYS

Code **	Lining Material	Nominal composition Al per cent	Cu per cent	Pb per cent	Sb per cent	Sn per cent	Other per cent	Sapphire Fatigue Rating* MN/m^2	Hardness HV	Relative Rating out of 10: Fatigue Strength	Seizure Resistance	Corrosion Resistance	Embeddability Conformability
L2	Whitemetal Tin Base		3·5		7·5	89		32	28	2	10	10	9
GM155	Whitemetal Lead Base			84	10	6		32	16	2	9	10	10
87S	Whitemetal Tin Base		3·5		8·5	87	Cd 1 + Cr + X	38	32	2	10	10	9
AS45	Tin Aluminium	60				40		59	28	4	9	10	9
AS15	Tin Aluminium	79	1			20		99	40	7	7	10	8
AS11	Tin Aluminium	92	1			6	Ni 1	102 (P)113	45	8	5†	10	6
SL	Copper Lead		70	30				(P)119	45	8	5†	5†	7
SP	Lead Bronze		73·5	25		1·5		>(P)124	52	9	4†	5†	5
SX	Lead Bronze		73·5	22		4·5		>(P)124	55	9	3†	5†	4
SY	Lead Bronze		80	10		10		>(P)124	110	10	2	5	2
AS78	Aluminium Silicon		88·5	1			Si 10·5	>112 (P)124	58	9	8	10	4

** These are proprietory to The Glacier Metal Co. Ltd.
(P) = with Pb Sn overlay ·025 mm thick.
* Ratings are comparative. They vary with lining thickness and figures quoted are for representative thicknesses for usual applications. Operating temperature of bearing metal in rig. 100–130°C according to loading.
† These apply to the material itself — these alloys are usually supplied with an overlay 0·025 mm thick of Pb-10 per cent Sn which improves the rating to 10 initially.

of alloys in common use, listed in general order of performance from lowest to highest, including the latest developments. Relative ratings have been assigned to the different properties.

Strength and ductility are important to enable the material to carry the loads imposed. Fatigue strength, whilst largely dependent upon these, is of sufficient importance to be considered separately for bearings subjected to dynamic loading. In some materials these properties are temperature dependent, and therefore the temperature at which the bearing surface will operate becomes of great importance in assessing the material to be used.

The fatigue ratings are obtained by subjecting a specific bearing to dynamic loads of increasing intensity. Three different rigs in use produce different numeric values which, however, it has been found possible to correlate. Because of the idealized form of the test rig with minimal misalignment and deflexion, these values are some two to three times higher than can be achieved in most practical applications.[1, 2]*

Compatibility or seizure resistance is a very important property when bearings are operating under boundary lubrication conditions, such as occur during slow speed running, say, when on turning gear, and during start-up and rundown.

Conformability and embeddability are both properties related to hardness. The former is the ability of the bearing material to accommodate high spots, edge loading and so forth without damage. The latter is the ability of the material to permit foreign particles to embed themselves into the surface, thus avoiding scoring of the bearing and the co-operating surface.

Finally, the material must resist corrosion by the lubricant, by products resulting from degradation of the lubricant during use, and from other fluids which may become entrained in it. Typically, such products are acids resulting from oxidation of the oil itself, acids and peroxides resulting from combustion products washed into the lubricating oil system, and entrained water and salts from condensation or leakage.

No single material has all these properties to a high degree, which is hardly surprising in view of the large range of machine requirements.

Historically, the first bearing material which can be considered as having been invented for the purpose was whitemetal. The invention is credited to Isaac Babbitt, who patented it in USA in 1839, where it is still known as Babbitt Metal. This was a tin base metal containing 11 per cent antimony and 6 per cent copper, a specification which is still broadly in use today. Since then the number of whitemetal alloys has proliferated dramatically, with compositions ranging from almost 100 per cent tin to almost 100 per cent lead.

Not many years ago there were over 450 whitemetal specifications in use, and in one year alone the author's company produced 125 different alloys. Many of these were of dubious differential value and under increasing economic pressure a rationalization programme was started in 1968, resulting in the number of alloys in general use being reduced to under 20, with the bulk of production being confined to four or five.

Whitemetal remains the best all round bearing material within its strength limitations. The most commonly used tin base and lead base alloys are shown in the Table. In practice whitemetal cannot be run at surface temperatures exceeding about 130°C, due to its low melting point. It possesses the useful property of self-healing after a minor wipe. The strength is very temperature dependent and Fig. 4 illustrates how the fatigue strength of the tin base alloy falls with rising temperature.

To improve the strength, and particularly the fatigue strength of whitemetal, many additional alloying elements have been tried, the most successful, developed in the early 50's, containing 1 per cent cadmium and about ·2 per cent nickel. Recently, however, it has been found possible to include small quantities of chromium and other structure refining elements which has resulted in further improvement in strength, without sacrifice of the other good properties.

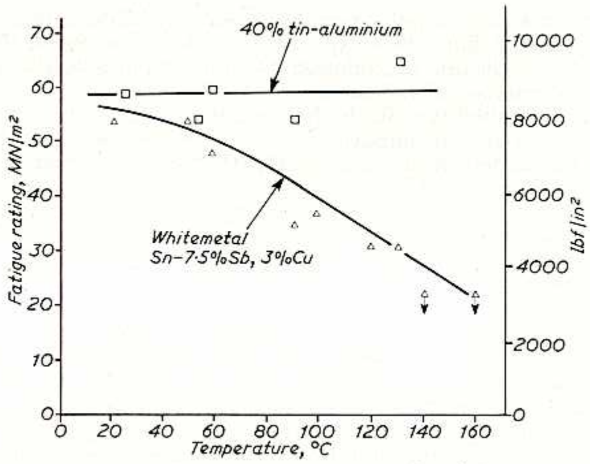

FIG. 4—*Fatigue strength of whitemetal and 40 per cent tin-aluminium*

This alloy is shown against the code 87S.

In the search for stronger materials for internal combustion engines, the copper-lead and the lead-bronze alloys were introduced in the late 1920's, and in the early 30's a method was devised of applying these alloys as a lining to steel strip.

Although having the highest strength ratings, these alloys have a poor performance in terms of properties, Fig. 1, ([4, 5, 6] and [7]). Performance can be improved considerably by applying an overlay of lead-tin or lead-tin-copper, usually to a thickness of 0·025 mm. Once the overlay is worn through, however, then corrosion resistance falls and the resulting corrosion may cause the onset of fatigue. Seizures tend to be hard, causing heavy damage to the shaft, and it is normally necessary to use hardened shaft surfaces to minimize this risk.

In the search for a better balance of properties a solid aluminium based bearing alloy containing 6 per cent tin was developed, but owing to the difficulty of maintaining nip in steel housings, it was not adopted to any great extent.

In the early 50's, however, the author's company pioneered a process for bonding a thin layer of tin-aluminium alloy on to a steel back by cold welding, and the alloys AS11 and AS15 containing respectively 6 per cent and 20 per cent tin were introduced. Whilst the strength is slightly lower than that of the copper-lead alloys, it is quite adequate for the majority of applications, whilst the general balance of properties is superior, corrosion resistance being particularly good. The alloys have gained wide acceptance.

In the late 60's a third alloy, AS45, containing 40 per cent tin was developed, primarily for use in slow speed diesel engine crossheads, where it was felt that whitemetal was reaching the limit of performance but a soft alloy with good surface properties was still highly desirable.

The fatigue strength of the tin-aluminium alloys is not temperature dependent — compare with whitemetal in Fig. 4.

The latest development in high strength aluminium alloys is AS78 containing 10·5 per cent Silicon. This alloy is as strong as the lead-bronzes but has very much superior surface properties and corrosion resistance, and all in all, appears to have a very promising future.

If any simple advice can be given on the selection of a bearing material, it is probably this — select the material with the best compatibility, consistent with its having adequate strength for the application. Where high strength is required, then compatibility must be obtained by overlay plating with lead-tin. In fact for difficult applications such as the crosshead bearings in low speed diesel engines, it has been found advantageous to overlay plate even whitemetal. The soft overlay allows high spots to be run-in relatively quickly without raising local temperatures to the point at which the fatigue resistance of the underlying whitemetal is reduced.

This completes a brief survey of the commonly used metallic bearings. But bearings do not necessarily have to be of metal and in the early days of engineering wood was frequently used. Indeed this extended until recent times in the use of hard wood, *lignum vitae*, for lining sterntube bushes. With the advent of the large variety of plastic materials, most of them have been considered for bearing use and some found to be worth adopting.

* These numbers refer to the Bibliography, at the end of the paper, which is designed for further reading.

Plastic bearings can be divided into two broad categories — those intended to operate with a lubricant and those intended to operate dry, that is without a fluid lubricant.

Taking the latter first, these are usually based on polytetrafluorethylene (p.t.f.e.) or on graphite. Both materials act as a dry, solid lubricant, transferring minute quantities to the interacting surface by the wear mechanism.

Neither of these materials is sufficiently strong to be much use in the pure form and they are usually incorporated in a stronger matrix. Since their thermal conductivity is poor, they cannot be used in the solid form at high sliding speeds, and their wear performance must be based upon the PV relationship (specific load × sliding velocity).

Probably the best of the p.t.f.e. containing materials is a proprietary one in which a mixture of p.t.f.e. and lead is impregnated into a porous bronze sinter, bonded to a steel, or bronze, backing. The bronze sinter is about ·5 mm thick, overlaid by the p.t.f.e./lead mixture to a thickness of about ·025 mm. Wear is relatively rapid until the bronze sinter is reached, when it stabilizes at a very low rate. Because of the porous bronze and the steel back, thermal conductivity is good and the wear performance and strength are considerably greater than for p.t.f.e. alone. However, the removal of frictional heat still imposes a limit and applications are generally found with relatively low speeds, but up to very high loads. Wear is proportional to PV. The temperature limits of this material are −200°C to +280°C.

Another proprietary "dry" bearing material is a compacted sinter of metal containing graphite up to 14 per cent by weight (40 per cent by volume). Applications include many at temperatures above those at which conventional lubricants can survive. Performance is improved if a fluid is present even in small quantities and there are many applications where water is the lubricating fluid — e.g. rudder pintle bushes.

A thermoplastic material which has been found to give very good results, particularly under boundary lubrication conditions, is an acetal copolymer. To provide adequate strength this is bonded to a steel backing by a porous bronze sinter. In this case the polymer has a thickness over the bronze of about ·3 mm permitting the surface to be machined. There is, however, a temperature limit of 120°C, and the PV criterion, particularly V, is limited by the ability to remove the frictional heat. The material finds its main application in high load, low sliding speed applications, in many of which, by using a pocketed surface to retain grease, the bearing may be greased for life. The performance under boundary lubrication conditions is extremely good and superior to even whitemetal. The wear performance compared with metals is shown in Table III. Unfortunately the material is subject to attack by the acids present in degraded engine oil and therefore is not suitable for engine applications.

Other plastics which are under development appear to have superior bearing properties to the acetal copolymer, but without its limitations.

The last of the important plastic materials are the reinforced thermo-setting resins. Asbestos reinforced cresylic and phenolic resins have been in use for many years using water lubrication but have come into some prominence only recently for use with oil lubrication. They have been giving satisfactory service for some years in sterntubes. As with all plastics, they have a low thermal conductivity and there may be more difficulty in dissipating the heat generated in the oil film than with metal bearings. Their high swelling rate requires a greater than usual oil clearance to prevent seizure in the event of water entering the system. However, they possess good boundary lubrication properties and, due to their low modulus of elasticity, the conformability is quite good.

This list of bearing materials is by no means exhaustive but it has been chosen to cover those most likely to be of interest and value to the marine engineer.

Looking at the mechanical requirements for the bearing, it is axiomatic that it must be dimensionally accurate to provide the correct operating clearance against the co-operating shaft and the surface must have the necessary accuracy of form if interchangeability is to be ensured.

However, in the case of the bearing there is one further consideration; it is itself usually fitted in a supporting housing. Many modern bearings are thin walled — i.e. where the ratio wall thickness/bore is 1/25 or less. Such bearings take their final shape from the housing in which they are contained, and to ensure the necessary interference fit to avoid fretting, the circumferential length of each half bearing in its free state is greater than that of the housing bore, the difference being referred to as the "nip". In these cases the bearing wall thickness must be held within close limits so that when assembled the bore configuration is within the prescribed tolerance. In view of the stackup of tolerances and the small operating clearances often required, the bearing wall must be held to very close limits.

Until comparatively recent years, the practice of scraping bearings continued. This commenced through the need to obtain the necessary accuracy of fit of shaft to bearing against a background of less accurate machining capability. Whilst there is reason to believe that a well scraped surface can be beneficial, particularly in cases of slow speed, sliding or oscillating movement, such as occurs in machine tool bedways and large diesel crosshead bearings, there is no doubt that poor quality scraping can be highly damaging. The need to ensure interchangeability and minimize fitting on site, has gradually caused scraping to be abandoned in favour of accurate machining. Since, however, this calls for general improvement in accuracy of approaching a whole order, it throws a heavy burden upon the production processes, which can generally only be resolved by introducing precision machine tools.

Because of the relative softness, a high degree of surface finish on the bearing is at the same time more difficult to attain and less important than in the case of the shaft, since the peaks of the bearing surface will abrade away readily and with little damage to form a run-in surface.

Typically, one may expect a surface finish of ·8 μm (32 μ inch) on large whitemetal bearings, but the latest machine tools are achieving 0·5 – 0·4 μm (20 – 16 μ inch).

To illustrate the importance of surface finish Fig. 2(f) shows representative shaft and bearing surfaces to scale with a mean oil film of 0·0065mm (·00025). During starting and stopping all bearings must pass through a period of thin film conditions. Many must operate on films no thicker than this.

The most commonly used bore profile, when viewed from the end, is a plain circle. It is simple and suitable for the great majority of applications. However, it has inherent deficiencies which have been realized with the development of modern high speed machinery. Under high speed, light load conditions, instability develops, resulting in a half speed whirl, where the shaft precesses around the clearance space at about half synchronous speed. This is most likely to occur at speeds above 4000 rev/min for stiff horizontal rotors although the precise threshold speed depends on shaft flexibility, bearing load and clearance. Moreover, the circular bore bearing does not have the stiffness and damping

TABLE III — WEAR OF PLASTIC LINED BUSHES COMPARED WITH COMMON BEARING ALLOYS

Material	Wear mm
Acetal copolymer	·005
Acetal homopolymer	·005
Nylon 66	·013
Whitemetal — tin-base	·015 — ·018
20% Tin-aluminium	·063 — ·086
Lead-bronze (25 per cent Pb 5 per cent Sn)	Seized after 5 — 10 hours (2 tests)

All bushes steel backed.

Test cycle of 9 min. rest/1 min. run, with lubrication of 1 drop of oil per min:

Load 25 N/mm^2 (3600 lbf/in^2)
Speed 1·4 m/s (270 ft/min)

BEARING TYPE	Resistance to half speed whirl	Load capacity	Stiffness	Damping	Acceptance of rotating load	Remarks
1 Circular bore Two axial groove	1	5	2	2	2	
2 Circular bore Circumferential single groove	1	3	2	2	5	Best for rotating load. Some gearbox bearings. Note:- This type and half groove type are widely used in reciprocating machinery
3 Lemon bore	2	4	3	2	2	Low stiffness in horizontal direction
4 Offset halves	4	4	5	5	2	Not suited to reverse rotation
5 Tilting pad	5	3	3	2	2	Pivots liable to fretting with dynamic or rotating load
6 Fluid pivot	5	3	3	4	3	New introduction. Lubricant bled from the hydrodynamic film through an orifice to provide a hydrostatic support. Application experience limited to small sizes
7 Three lobe	3	3	4	3	2	Sizes up to about 100 mm
8 Four lobe	2	2	3	3	2	Sizes up to about 150 mm
9 Floating bush	3	2	1	3	3	Smaller sizes
10 Three pocket	1	1	1	1	1	Low load capacity unless pocket tapers kept very fine (about 0·001 mm/mm)
11 Dammed groove	1	5	2	2	1	Poor load capacity in top half, liable to fatigue damage in top. Types 3, 4 and 5 preferred

Performance ratings — 5 = excellent to 1 = poor. Arrow in bore shows suitable direction of rotation. The shaded zones indicate suitable regions for the load vector. Load capacity outside these will be reduced, possibly down to 10%

FIG. 5—*Typical profile bore bearings compared with circular bore*

coefficients sometimes required in modern machines.

To overcome these defects a large variety of more complex bore profiles have been devised and these are indicated in Fig. 5. It must be emphasized that the relative ratings (5 = excellent down to 1 = poor), are only very approximate.

The lemon bore, type 3, gives an improvement on the plain bearing, is simple and suits either direction of rotation. Type 4 gives much improved stability and control but has the defect of being only suitable for uni-directional rotation.

Of all types, only numbers 5 and 6 are stable under all practical speed and load conditions, and consequently number 5 is widely used in vertical axis applications. Number 6 is a new development which is at present finding application mainly in high speed gears and compressors.

Even though more expensive, the tilting pad journal bearing, type 5, has advantages over the simpler forms of profile bore because:

1) it is more tolerant of widely ranging speed and load conditions;
2) the threshold instability speed is within excess of that required in most currently practical applications:
3) its load capacity is comparable to plain bearings but it is not sensitive to load direction, nor to overheating under no-load conditions, e.g. as occur in a

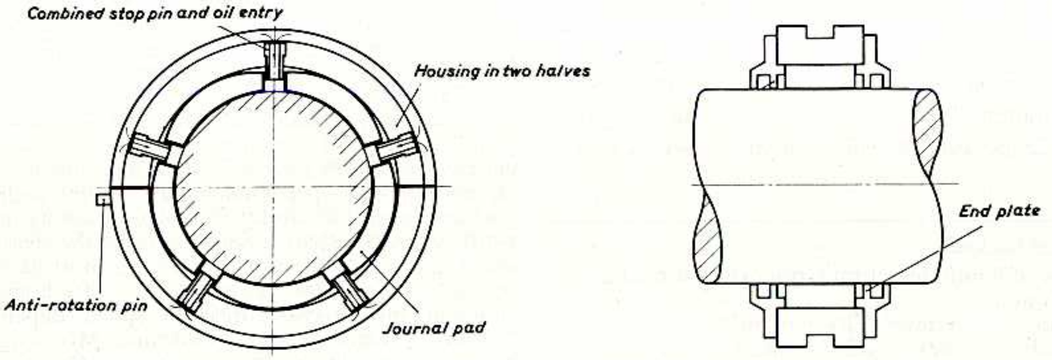

FIG. 6—*Tilting pad journal bearing*

locked train primary pinion bearing;

4) it can be made to give a "preloaded" effect under suitable conditions.

A typical bearing is illustrated in Fig. 6. Lubricating oil is supplied from an annular groove in the housing through jets to the inlet edge of each pad.

Computer programmes, based on both theory and test results have been devised for most of these types of bearing to evaluate dynamic stiffness and damping coefficients, as well as film thickness, power loss and temperatures.

In many of these bearings, and particularly where a high degree of shaft control is required, i.e. a high stiffness coefficient, the clearance is critical and must be kept to a minimum value. This is dealt with at greater length below under "ENVIRONMENTAL."

The term "features" embraces all forms of grooving for oil entry or discharge. The commonest form of grooving is two axial grooves at the bearing joint line, see Fig. 5, No. 1. This is used for most steadily loaded bearings but it will be observed that there are limitations to the position of the load vector for best performance. (See later, Fig. 12). For dynamically loaded applications, and where the load vector may act in any direction, type 2 is to be preferred. It may be noted that very often, particularly in engine bearings, the circumferential groove extends only around one half of the bearing and the other half is plain to obtain increased load capacity.

Occasionally one observes very elaborately featured bearing bores, but the value of such is doubtful and generally they can be replaced with a simpler profile or grooving system with no ill effects. As an example, the author recalls highly featured bearings of type 11 in the turbines of a class of VLCC which gave repeated trouble in service. Subsequently they were replaced by type 3 which gave excellent service. For certain specific applications, particular types of grooving are essential, and these will be dealt with later in the paper.

the transmission of heat into the bearing from the adjacent structure is greater when the machine has stopped after a period of operation, and this may well be the most critical time for the bearing. In extreme cases it may inhibit the use of whitemetal and make it necessary to use a material capable of withstanding a higher temperature.

Clearance is usually considered one of the characteristics of a bearing but in practice it is dependent upon the size of shaft as well as that of the bearing bore, and moreover, in the case of thin walled bearings, is dependent upon the housing also. Therefore, it has been included in this section to emphasize this interdependence. It may be observed that, traditionally, it has been common practice to make the shaft nominal size and to obtain the clearance by increasing the bearing bore. This very much inhibits standardization of bearings and modern practice is, therefore, to make the bearing bore the nominal, fixed dimension, providing the clearance by reducing the journal diameter. It is thus possible to standardize bearings and still use different clearances to suit individual machine applications.

Clearance is necessary to allow the formation of the load carrying oil film. The actual clearance selected is largely a matter of judgement and experience — too large a clearance will reduce the load carrying capacity — too small a clearance will increase power loss and may cause seizure. Fig. 7 shows recommended minimum clearances for rotating plant bearings carrying a steady load and Fig. 8 shows recommended clearances for diesel engine bearings. It should be noted that these are minima and the maximum clearance will depend upon the stackup of tolerances. Where permissible tolerances make the maximum clearance too high, then consideration must be given to reducing this minimum clearance, reducing tolerances, or both.

It may be pertinent here to mention that in selecting a minimum clearance, some consideration must be given to the transient conditions which occur when a machine starts up from cold. When machines run up to speed very quickly the initial rate of heat input to the bearing is much higher than at normal operating temperature. The shaft expands, whilst the inside of the bearing, heating up much more rapidly than the housing, expands inwards, both factors serving to reduce the clearance. There is a delay until the housing warms up and temperatures are stabilized, during which the bearing must operate with reduced clearance, and since the reduced clearance in itself increases the rate of heat generation, there is a real danger of the clearance being reduced to seizure point. The simple solution is to use an adequate clearance. Where this is not possible because a high degree of shaft constraint is needed with a special bore profile, then it may be

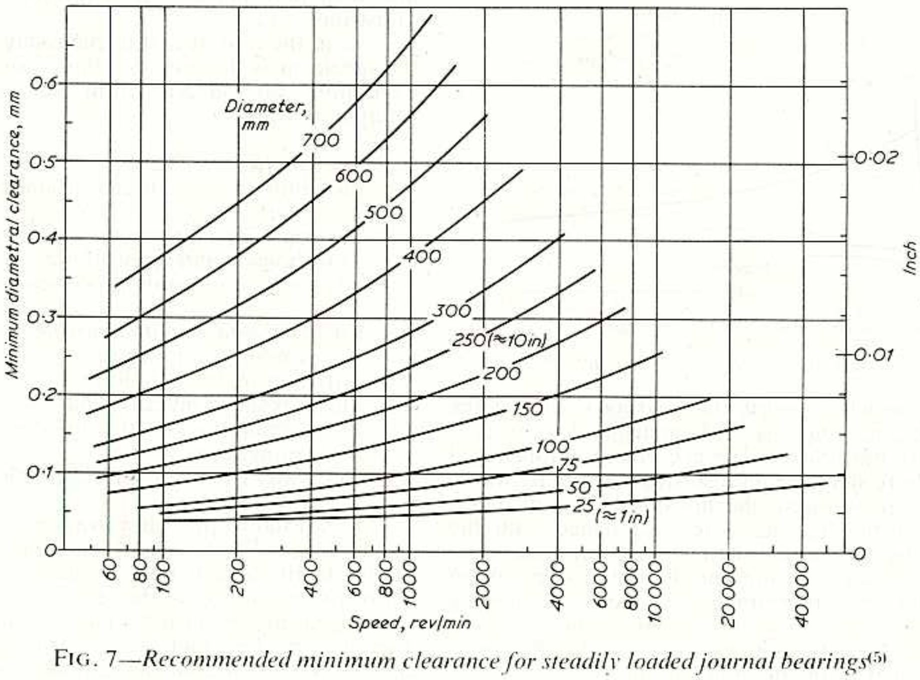

FIG. 7—*Recommended minimum clearance for steadily loaded journal bearings*[5]

ENVIRONMENTAL

So inter-active are the various components of the bearing system that it is difficult to separate them and deal with them in isolation. This section has been split into two parts — the applied conditions are those imposed on the bearing system, whilst the resultant characteristics are those which derive from the former.

The load cycle which may be a steady continuous load, a fluctuating load or may be a dynamic load vector, as in the case of engines, and the sliding velocity resulting from the speed of rotation, are established by the machine.

The temperature condition mentioned here is that imposed on the bearing from the environment, that is adjacent heat in the machine, e.g. a turbine shaft. In many instances

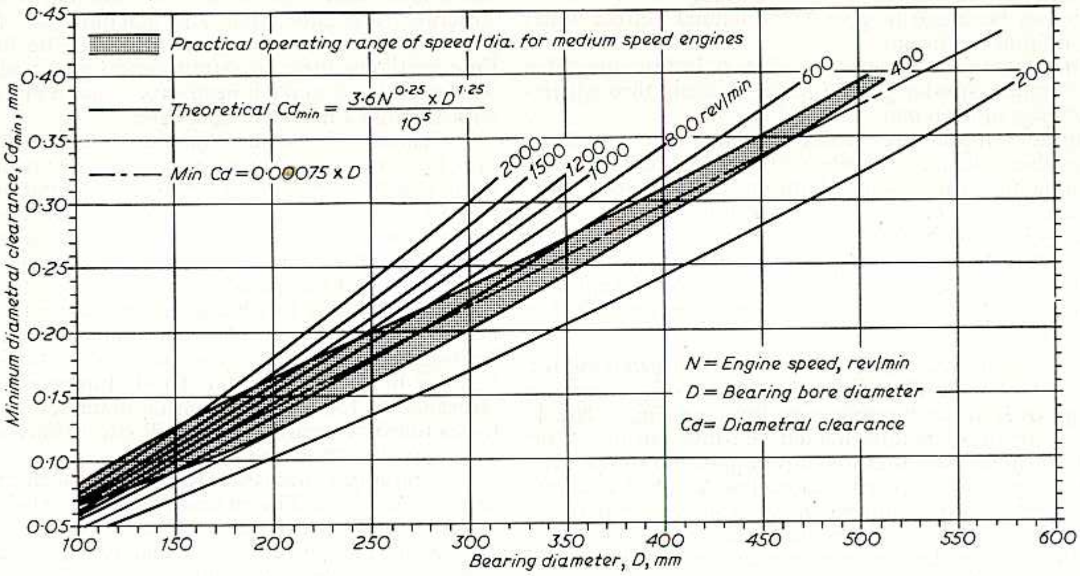

FIG. 8—*Recommended minimum oil clearance for diesel engine bearings*[3]

necessary to use a slow starting cycle, to preheat the oil, or eve both. Fortunately such cases are fairly infrequent.

Fig. 9 illustrates the transient loss of clearance, measured at the bearing ends, in a bearing during the starting cycle, at different speeds. At 5000 rev/min seizure occurred through loss of clearance. Subsequent measurement showed the bearing bore to have increased by ·04 mm at the centre[21].

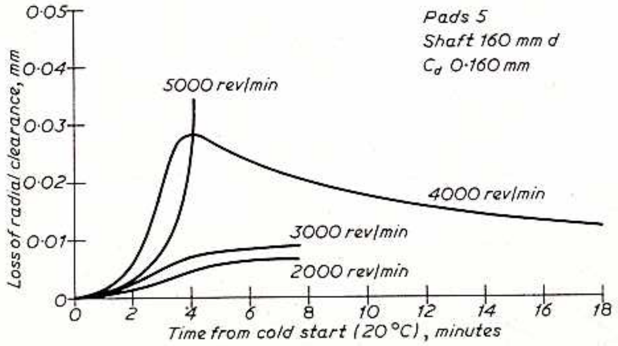

FIG. 9—*Transient loss of clearance on start up*

The housing which supports the bearing is one of the prime factors in the bearing system. Dimensional accuracy is necessary to ensure that the bearing is correctly gripped and to ensure that its bore shape is not distorted. Especially where thin walled bearings are used, the housing bore will determine the shape of the bearing bore, and hence both the dimensional and the geometric tolerances are important. It is also important that the housing should not distort, either under the applied operating forces or under the bolting forces. In practice some distortion is inevitable and the designer's aim must be to keep the distortion within the limits which can be accepted by the bearing oil film.

Similarly, the alignment of the housing under all loading conditions is equally important. Not only must the alignment be correct when the machine is erected, but changes in alignment due to operating deflections or thermal distortions must be within the limits tolerated by the oil film. Where it is impossible to build in sufficient rigidity, then there is the alternative of designing in flexibility so that deflections of shaft and housing are conjugate and alignment is maintained.

For correct operation a bearing requires a certain volumetric rate of lubricant supply and to achieve this the pressure and temperature of the lubricant supply are both important.

Since heat is generated in the oil film in a bearing, it is necessary to arrange for removal of the heat at the same rate, in order to establish thermal equilibrium. In forced lubricated bearings the heat is removed with the escaping lubricant and is extracted in the lubrication system. In self contained bearings it is necessary to provide for removal of the heat from the bearing casing by some suitable means such as water cooling. Since the rate of heat generation diminishes with rising temperature, and the extraction rate of a cooling system increases with rising temperature difference, thermal equilibrium is fairly easily established, providing of course that the cooling system is adequate at the operating temperature: if not, then the operating temperature will rise, possibly to a disastrous level.

As in the past, it is still true today that contamination of the lubricant is the greatest single cause of bearing damage. Contamination can arise from many sources, the following being typical:

Solid particles:

a) built in — millscale, foundry sand, machine shop swarf;
b) ingested — from dusty or dirty atmosphere;
c) abrasion and wear products — metallic;
d) from combustion processes — carbon.

Chemical contamination arising from:

i) combustion processes;
ii) dilution by fuel oil;
iii) leakage of process fluids;
iv) degradation of the oil due to oxidation or high temperature;
v) water from condensation or leakage.

It is usual to provide filtration in the lubrication system to remove solid particles. Unfortunately filters which are commercially acceptable are rarely capable of filtering the lubricant to a sufficient degree of fineness to prevent damage to the bearings by particles passing through the minimum oil film. For modern high duty bearings, particularly in engine applications, filtration should be down to about 5 µm to prevent damage to the bearings, but in practice it is unusual to find filtration much finer than 10–15 µm and even as coarse as 30–50 µm. Better results can be achieved by fitting, in addition to a strainer type full-flow filter, a centrifugal by-pass filter which will remove a high proportion, about 80 per cent of particles down to about 2 µm.

Little can be done to remove chemical contamination except by oil change.

Vibration is present to some extent in all machinery and indeed engine bearings may be considered to spend their working lives under conditions of severe vibration which generally they survive. However, vibration damage does occur, usually appearing as fretting between surfaces, and fatigue of linings.

Tilting pad bearings are prone to fretting of the pivots and supporting surface when subjected to dynamic loads. As a general empirical rule cyclical load variation should not exceed about ±30 per cent if damage is to be avoided.

Examples are shown in the section on bearing damage.

Electrical potential between the surfaces can have disastrous results in a short time. It can arise from stray currents in magnetic machines or from build up of static electricity in machines such as fans and turbines. It has even been known to occur in oil systems.

Examples are shown in the section on bearing damage.

Arising from the combination of all the factors already described, the bearing will operate with a certain minimum film thickness (h_{min}) and this has become the accepted criterion of bearing performance and safety.

With so many variables it is not possible to give an absolute statement of what is a safe minimum film thickness but experience and judgement indicate what may be acceptable, and Fig. 10 may be used as a guide for steadily loaded journal bearings.

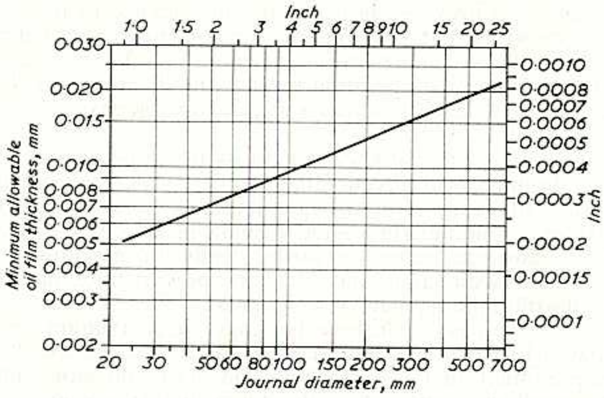

FIG. 10—*Minimum allowable oil film thickness for steadily loaded journal bearings*[5]

What is generally termed "bearing friction" arises in fluid film lubrication from the viscous shear, or drag in the oil film, and appears as heat transmitted to the lubricant and to the surfaces.

Power loss is the term given to the power absorbed by the bearing due to this friction.

In operation the bearing will attain certain temperatures which vary over the surface, approaching a maximum at a point between the minimum film and the outlet. In research, and sometimes in practice, these temperatures are monitored by thermocouples placed as near to the bearing surface as possible, but these methods have the disadvantage of requiring rather delicate leads which are easily damaged during erection or operation. It is, however, this temperature known, or unknown, which determines the durability of the bearing surface.

The lubricant outlet temperature derives from the volumetric flow and the heat generated in the bearing and is perhaps the most commonly used monitoring device for bearing safety. As such it is not very reliable because a rapid seizure can occur before any warning is given by the outlet oil temperature. Nevertheless it does give some indication of bearing behaviour and will indicate any slow deterioration, giving time for corrective action. It survives because of its practical convenience and low cost.

Having discussed the bearing system in general terms, it is useful to consider specific types of application.

DIESEL ENGINE BEARINGS

The most important single development in this field has been the increasing adoption of thin wall, prefinished steel backed insert bearings instead of a bearing alloy cast into the housing. First introduced in 1952, they are today fitted in nearly all medium and high speed engines.[3]

The rapid acceptance of the prefinished thin wall bearing was largely to provide the engine designer with a much wider range of bearing materials. Before the adoption of these the standard lining material was tin base whitemetal cast directly into the bearing housing. With the continual uprating of the power output of medium speed diesel engines, bearing loadings soon passed beyond the capability of whitemetal. Copper-lead bearings were next used, followed by tin-aluminium, both dictating the need for insert bearings, as these materials can only be produced in such a form.

Even where whitemetal is still used, as at present in the slow speed engines, the insert bearing offers marginal improvements due to the better control of the casting techniques which can be achieved. Also, the insert bearing is a precision product which can be fitted directly into the housing without the need for scraping or fitting work to be carried out.

In the slow speed 2-stroke diesel engine, the top end bearings have, for some years, been among the less reliable components.

Between 1962 and 1965 trials of thin wall insert bearings were made in the crossheads of most leading makes of slow speed diesels. Following these tests, Doxford and MAN began to adopt the design in production engines. From 1970 all leading makes have used insert bearings in crossheads. In Europe most engines still use whitemetal lined insert bearings, although service trials of 40 per cent tin-aluminium lined bearings have been ongoing for some years and these have been adopted by Sulzer for their RND-M engines. In Japan, however, some makers have been fitting 40 per cent tin-aluminium for four years with success.[25]

Even so, improvements in design and materials have barely matched the engine builders requirements for uprating, and these bearings still operate with too small a margin of safety. In a two-stroke engine the bearing must operate under slow speed oscillating conditions which produce very thin oil films, and there is no load reversal, which in a higher speed engine allows the entry of oil, providing a squeeze film.

So far it has not been found possible to produce any bearing theory which enables an assessment of the operating characteristics to be made. The design has largely evolved by experience and Fig. 11 illustrates what is believed to be the best design practice.

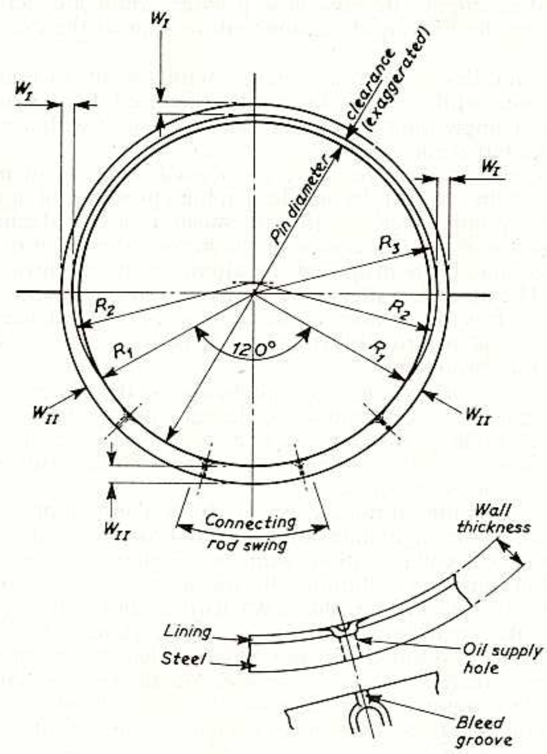

FIG. 11—*Diesel engine crosshead bearing — general configuration*

The load carrying half bearing has a bore profile consisting of a 120° "bedded" or "fitted" arc, relieved at each side to match a concentrically bored unloaded half which contains the clearance.

Ideally the radii of pin and bearing should be equal, but with the need for production tolerances it is usual to limit the difference to 0·00025 of the pin radius.

Referring to Fig. 11, dimensions for a typical bearing, nominally of 700 mm diameter are, in mm:

1) pin diameter	719·910/719·880
2) housing bore	750·000/750·070
3) bearing wall thickness WI	14·685/14·645
4) bearing wall thickness WII	15·045/15·005
5) assembled radius R1	359·955/360·030
6) assembled radius R3	360·315/360·390
7) top clearance (diametric)	0·360/0·540
8) top clearance ratio	0·00051/0·00077
9) R_1 minus pin radius	0/0·090
10) Ratio of 9 to pin radius	0/0·000257

In practice it is sometimes necessary to modify the wall thickness to counter the effects of housing distortion under either bolt load or operating forces. Fitting tests should always be carried out in the development stages.

Some research has been carried out, but owing to expense this has been limited to small scale trials, and in the absence of any clear knowledge regarding the laws of similarity it is not possible to extrapolate the results to full scale bearings. The angular groove spacing should not be greater than the swing of the rod. Work carried out in Japan indicates that the spacing should also not be appreciably less than the rod swing, so as to maintain the largest possible load supporting area.

Due to the very thin oil films, the flow of oil through the bearing is very restricted, and bleed grooves are provided to increase the oil flow through the supply grooves to provide better cooling.

In most designs the pin is hardened and polished to a fine finish of at least, and preferably better than, 0·1 µm (4 µ inch). In this context it should be stressed that the geometric shape is equally important and there should be an absence of any waviness in the surface. The bearing surfaces and geometrical tolerances are equally important, as also are those of the housing bore.

Most engine designs now provide conjugate deflexion whereby the bearing deflexion follows that of the crosshead pin.

Some designs favour a large diameter pin to minimize deflexion, whilst others favour the inverted T construction, where a single long lower half bearing is used with a narrow upper strap at each end.

Various palliatives have been tried, and are in use, to improve the oil film. In one design, high pressure oil injection from a pump fitted on the crosshead is used. Mechanical separation of the surfaces near the bottom dead centre using springs, has been proposed, or alternatively eccentric bearings. The latter is understood to have given promising results in trials. It is the author's personal view that, on balance, high pressure oil is probably the best long term answer to this particular problem.

For main and bottom end bearings, the advent of the computer has revolutionized design. By analysis of the bearing polar load diagram and by careful use of design procedures, oil film conditions under dynamic loading patterns can be established.

In 1958 the author's company built a simulator to study the behaviour of dynamically loaded bearings, and in 1962 commenced collaboration with Nottingham University to develop computer techniques for the design of such bearings.

In 1965 a 600 hp (447 kW) Ruston and Hornsby VEB 6 cylinder medium speed diesel engine (kindly loaned by Messrs. Ruston and Hornsby) was installed and work began on the correlation of the theory with practical results obtained from the engine. This culminated in the development of computer programmes for the design of dynamically loaded bearings, with the assistance of Professor J. F. Booker of Cornell University. Since 1968 over 6000 dynamically loaded bearing cases have been handled.[4]

Various computer programmes are available, but because the assumptions vary, they cannot be compared one with another. However, the results obtained from any one programme may be used comparatively to optimize the bearing design and to appraise the effect of design and operating changes.

The assumptions made in the programme used by the author's company are:

1) The alignment of the shaft and bearing throughout the cycle is perfect.
2) The perfect shaft and bearing always remain rigid and truly circular.
3) The viscosity of the lubricant is constant around the bearing throughout the cycle.
4) Negative oil film pressure may be neglected.

The correlation of this data with known engine operating results provides the experience to allow a confident assessment to be made, at an early stage in the engine design, with the likelihood of achieving satisfactory performance. This improved bearing design technology, coupled with reliable feedback of operating performance, represents a growing contribution by the bearing designer towards improving the rating, reliability and, ultimately, the competitiveness of engines.

This, however, is not the total story and it remains to achieve the necessary supporting characteristics such as shaft and housing rigidity, alignment and so forth. It is also essential to provide the necessary interference fit to prevent the insert bearing fretting in the housing by providing a suitable back contact pressure; generally 5MN/m² is aimed at. In this area co-operation between the bearing and engine designers in the early stages is most advantageous.

TURBINE AND GEARING BEARINGS

In the previous section the emphasis was on the need for thin wall bearings in stronger materials to withstand high dynamic loadings. In geared turbine installations, however, the loads are generally fairly steady in magnitude and constant in direction, and up to the present time have been well within the capacity of whitemetal.[5, 6].

In the case of turbine bearings, the maximum specific load which can be imposed is largely determined by the requirement that the machine can start and stop and be operated on turning gear without wear in the bearings.

With the tendency towards higher speeds, particularly associated with gas turbines, bearing metal temperatures are pushing towards the limit of whitemetal at 130/140°C. There is also some evidence that certain EP oils corrode whitemetal at high operating duties. These aspects are focusing attention on the 40 per cent tin-aluminium alloys as a possible substitute for whitemetal, but still retaining its kindly surface properties.

The more common problem with modern turbines where speeds are high and rotors relatively light, is the tendency to instability, and profile bore bearings are being used, notably types 3 and 4, Fig. 5. Large land-based machines are now tending towards the use of tilting pad journal bearings of assymetric design, using pads of different size to suit the position of the load. In this case the complication is warranted in order to minimize power loss in bearings where it may well amount to some hundreds of kW.

In the design of bearings for marine propulsion gear boxes, care is needed to obtain the correct positioning of the oil inlet grooves. Particularly in naval gearboxes, with the advent of combining gearboxes for two gas turbines, or gas turbines and diesel engines, and including reversing drives, many of the bearings can be subjected to different positions of the load vector, depending in changes in the operating conditions.[27]

The oil groove position should avoid the loaded area so that the maximum length of convergent oil film is available to develop hydrodynamic pressure. Generally, the grooves should be placed between 60° and 120° on either side of the load line. If the bearing is subjected to changes in loading conditions, the load line under these conditions may lie near a groove, in which case the load capacity of the bearing will be reduced.

Fig. 12 indicates a normal bearing arrangement at (a), whilst at (b) and (c) are shown the effects of closer approach of load line to groove. If the same h_{min} is to be obtained, then the load capacity of the bearing is reduced, Fig. 3(b). Alternatively, if the same load is applied, then pressures will increase and h_{min} will be reduced, Fig. 3(c), resulting in the bearing operating nearer to, or possibly beyond the limits of safety.

To assess the effect of proximity of load line to groove, Fig. 13 has been prepared and should be used with its notation.

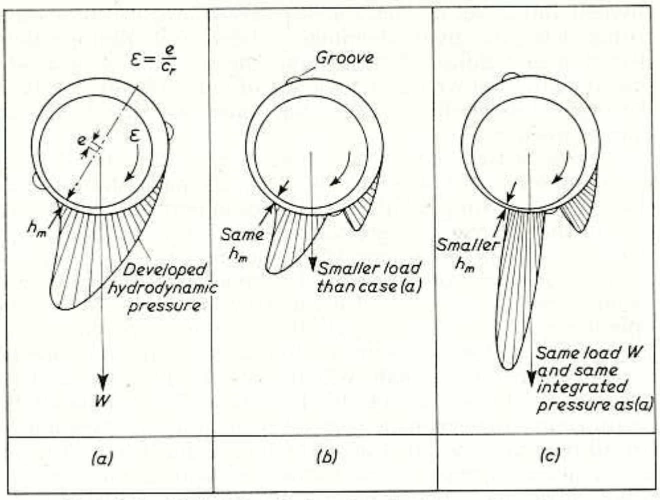

FIG. 12—*Reduction in bearing load capacity due to interruption in the converging oil film*

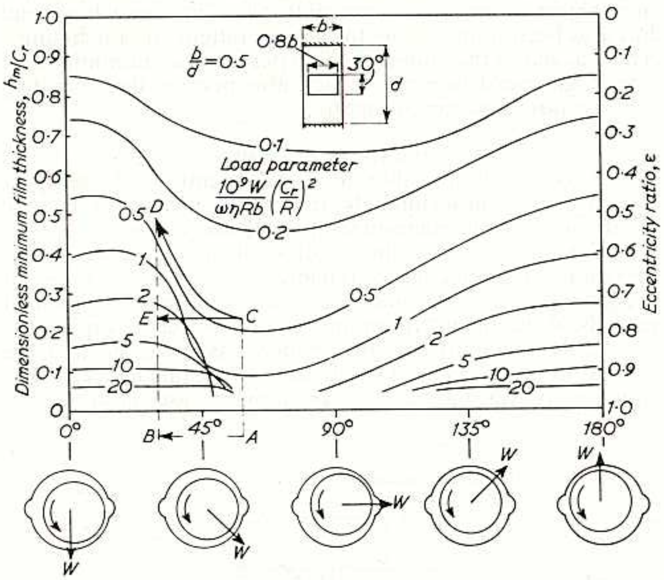

FIG. 13—*Variations in load capacity with change in direction of load*

Notation: FIG. 13

b bearing length (mm)
Cr radial clearance (mm)
d bearing diameter (mm)
e distance between bearing and journal centres (mm)
hm minimum oil film thickness (mm)
R bearing radius d/2 (mm)
W load (N) (1N = 1 Newton = 0·1 kgf or kp)
ε eccentricity ratio ($\varepsilon = \frac{e}{Cr} = 1 - \frac{hm}{Cr}$)
η absolute viscosity (centipoise) (1 centipoise = 10^{-3} Ns/m^2)
ω rotational speed (radian/s)
Δ dimensionless load parameter

For above units $\Delta = \frac{10^9\ W}{\omega\eta\ Rb}\left(\frac{Cr}{R}\right)^2$

An example is shown on the chart. By moving the groove 30° further from the load line, i.e. from A to B, the minimum film thickness, h_m may be doubled for the same load, line C to D. Alternatively, the load W may be increased by a factor of 7 with the same h_m, line C to E.

It is necessary to consider the load line position for each operating condition so that the optimum position for the oil grooves can be determined. Provision should be made to rotate the bearing within the housing during assembly to bring the grooves to the desired position. It is not recommended to move the grooves in the bearing away from the joint line because the joint would cause a discontinuity in the bearing surface.

Where it is impossible to avoid interference between grooves and load line, it may be necessary to adopt a single circumferential groove, as Fig. 5 type 2, and to design the bearing accordingly.

In locked train gearboxes cases of overheating of the primary pinion bearings have occurred. This was due to the pinion shaft being positioned by the gear teeth, and running concentrically in the bearings, thus very much reducing the oil flow. Adoption of lemon bore bearings increased the oil flow, reducing the temperature to normal.

TILTING PAD THRUST BEARINGS

The tilting pad thrust bearing is well established in marine practice and has a high reputation for reliability. With the growth in size, loading and particularly speed of turbo machinery, especially gas turbines, limitations began to appear and it was realized that not enough was known about the operating mechanisms. Since 1960 research has been concentrated on a better understanding of the mechanics of these bearings, with particular reference to reduction in power loss and improved operating factors at very high speeds.

For many years thrust bearings were designed on a basis of specific load, limiting this to values considered safe in the light of experience. Allowance was made empirically for the reduced load capacity at low sliding speeds.

The criterion now used for bearing design is the minimum oil film thickness, evaluated by computer techniques. From these, charts have been produced to enable designers easily to determine a suitable bearing and also to ascertain the power loss and oil flow requirements.

The minimum film thicknesses adopted for the safe operation of thrust bearings are given in Fig. 14.

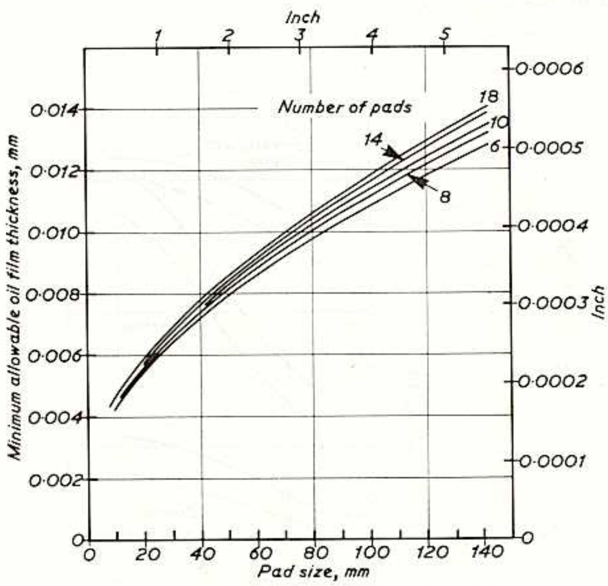

FIG. 14—*Minimum allowable oil film thickness for thrust pads*[26]

But contrary to earlier beliefs, it has now been established that the load carrying capacity is not only limited by film thickness at low speeds, and by structural strength, but also by the surface temperature at high speed[7, 26]. Fig. 15 illustrates how thrust capacity varies with speed for two sizes of bearing. It will be observed how the larger bearing has a lower load capacity at the higher speeds.

At high surface speeds, and 100 m/s is common in modern practice, the power loss and cooling oil requirements are very high, whilst surface temperatures become critical

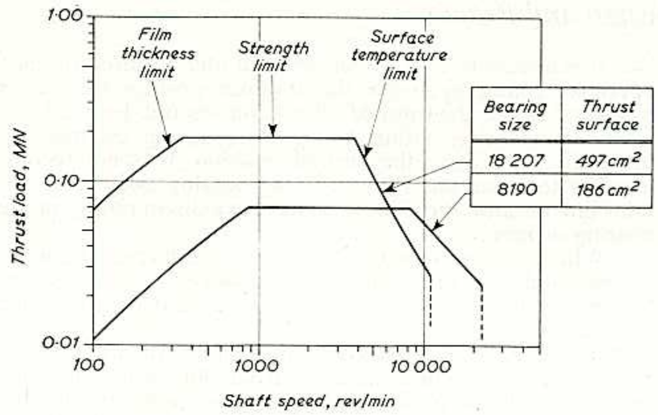

FIG. 15—*Limiting factors on thrust bearing capacity*(7, 26)

even with a high oil flow. Of the orthodox designs, with the casing flooded with oil, the single inlet/outlet system with flow across the bearing has been found to absorb the least power. However, it became evident that if power loss was to be reduced significantly, it would be necessary to minimize contact of the oil with moving surfaces.

In the USA a common practice is to restrict the oil flow to the bearing and provide a free discharge. Whilst this method was found to reduce power loss, it is felt to be inherently uncertain and rather hazardous. It is sensitive to oil flow and below a critical flow the surface temperature rises rapidly.

The most efficient system proved to be "directed lubrication" where the oil is sprayed at high velocity on to the collar surface between the thrust pads, the bearing being freely drained. Not only is the power loss reduced considerably, but pad temperature is also reduced and oil film thickness simultaneously increased — all factors making for greater safety. The system is not sensitive to oil flow.(8)

In Fig. 16 power loss is compared with oil flow at (a) for the three systems mentioned, and at (b) test results for the same bearing with pressurized casing and directed lubrication are compared. The latter shows a distinct gain in all parameters.

For many years there has been a certain degree of controversy over the relative technical merits of centre pivoted and offset pivoted thrust pads. In practical terms the former have the great advantage of being suitable for either direction of rotation, thereby removing the hazard of incorrect assembly for whatever reason, and indeed being essential for applications where the bearing must accept rotation in either direction.

Tests carried out on the same size bearing fitted with centre pivoted, and then with off-set thrust pads, showed that there was no distinguishable difference in performance at low sliding speeds, and no significant difference in power loss at any speed. The minimum film thickness was also nearly enough the same for all practical purposes, being if anything slightly less with the offset pivot. However, the offset pads operate at a greater angle of tilt than the centre pivoted, thus increasing the flow of cooling oil and resulting in rather lower surface temperatures than with the centre pivoted pad. The results are plotted in Fig. 17. It is therefore beneficial to retain centre pivoted pads with all their practical advantages for all bearings operating at lower speeds, but where surface temperatures approach the limits for whitemetal there is some advantage to be gained by adopting offset pads, accepting the greater complication. Some comparative results are illustrated in Fig. 18.

However, it has been found only recently that at very high sliding velocities, whitemetals, particularly lead-base, are subject to attack by some E.P. oils. Tin-base whitemetal has also been found prone to thermal fatigue or "ratcheting". This has led to the adoption of 40 per cent tin-aluminium for very high speed bearings, which also permits the operating temperature to be raised slightly.

STERNGEAR BEARINGS

It is hardly possible to avoid mention of sterngear bearings since, unfortunately, these remain one of the more troublesome components in the propulsion machinery.

For many years the traditional *lignum vitae* water lubricated bearings gave reliable service on a basis of accepting steady wear and renewal of the lining at stipulated periods. With the growth in the size and power of ships in the early 50's, the limit of performance was exceeded and the periods at which renewal was necessary became unacceptably short. Moreover, the heavy wear, in many cases up to 12 mm

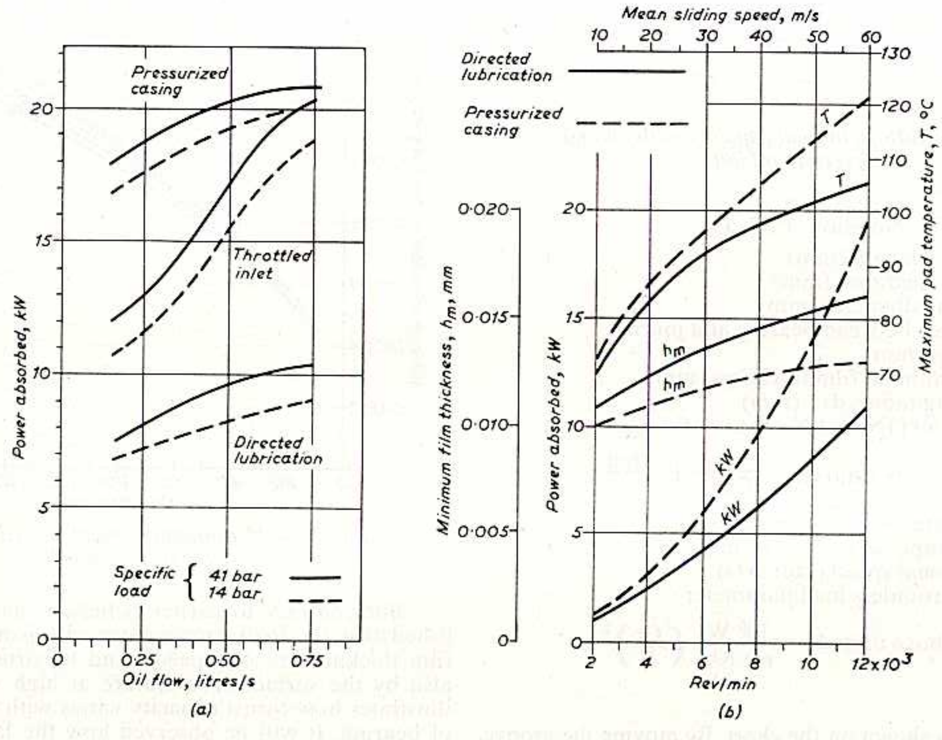

FIG. 16—*(a) Effect on oil flow on power loss*
(b) Comparison between pressurized casing and directed lubrication

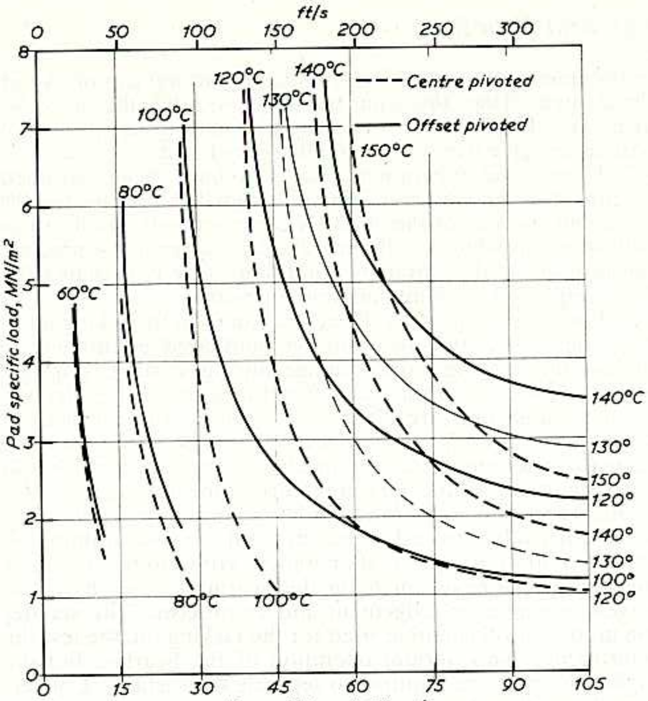

FIG. 17—*Operating temperatures of centre and offset pivoted thrust pads*

or so, had repercussions on the propulsion machinery bearings in aft installations. Thus there commenced the popular swing to whitemetal lined bearings running with oil lubrication, depending upon seals to contain the oil, and exclude sea water.

However, the size of ships continued to increase and in turn even these systems began to exhibit limitations. Deep draught conditions overloaded the seals which showed a distressing tendency to fail, leading in many instances to failure of the bearing due to dilution of the oil with sea water.

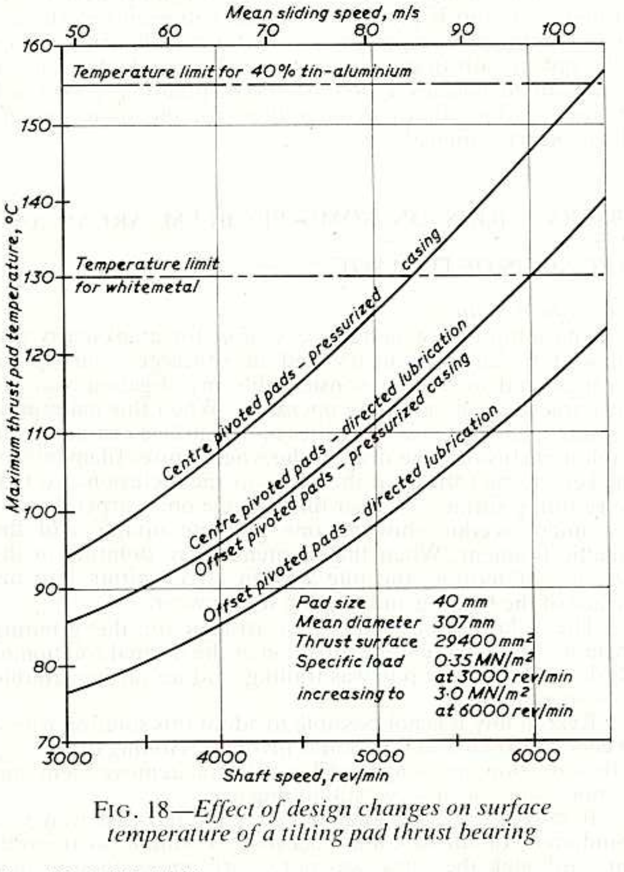

FIG. 18—*Effect of design changes on surface temperature of a tilting pad thrust bearing*

Large hulls with a high block coefficient tended to cause greater changes in shaft line, due to eccentric forces, whilst changes in hull deflection with draught, also affected shaft/bearing alignment. This is in addition to the uncertainty of alignment, in any event, with conventional designs of stern bearing.

The problems are well known and well documented elsewhere.(9, 10, 11, 12, 13, 17)

It is felt that this has been a somewhat neglected subject in the past and it is reassuring to see that serious thought is now being directed to it by some shipbuilders and Classification Societies.(14, 15, 16)

The stern bearing differs from most bearings in that it has a greater L/D ration, nowadays usually about 2, and even if initially aligned correctly, which is often doubtful, it is subject to changes in alignment during service to a degree greater than would normally be considered acceptable. But little information exists as to the actual changes in the load and shaft curve during the varying service conditions.

Some time ago studies were commenced on the operation of bearings with misalignment but the subject is unusually complex and work is still in progress.

To illustrate the effects of misalignment, and taking as an example a typical VLCC with a shaft diameter of 849 m, Figs. 19 and 20 have been prepared.

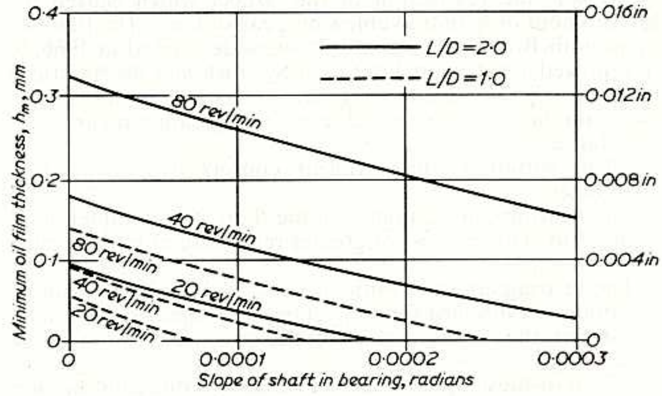

FIG. 19—*Effect of shaft misalignment on minimum oil film thickness for two length/diameter ratios*

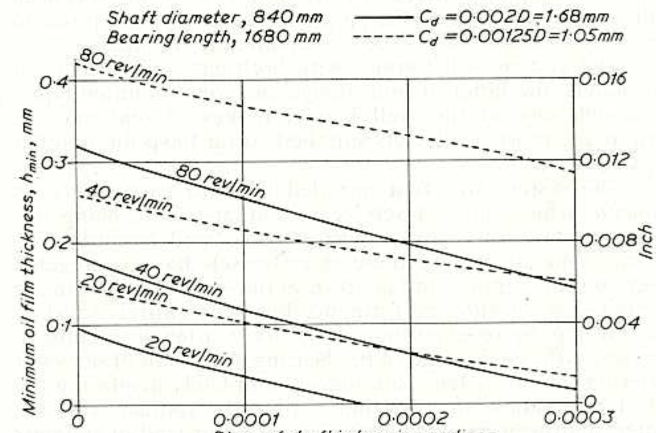

FIG. 20—*Effect of shaft misalignment on minimum oil film thickness for two clearance ratios*

Fig. 19 shows that above half speed h_{min} is adequate even at a slope of ±·0003 but at lower speeds there will be contact unless the alignment improves. Below 20 rev/min alignment must be better than ·0001. As a comparison, curves are shown for a short bearing, L/D = 1·0, from which it will be seen that this bearing operates with a considerably thinner film and is therefore more vulnerable. In Fig. 20 the same bearing is compared with one having a reduced diametric clearance, which increases the film thickness and makes the bearing less vulnerable to misalignment. This reduction in clearance would also be beneficial since it increases the stiffness and damping capacity of the bearing. It will be observed that at a slope of ·0003, interference by the top of the bearing at the "high" end is beginning to have effect.

The essential requisite for good bearing performance is the provision of good alignment under static conditions so that during starting, stopping and whilst on turning gear, damage to the bearing will be avoided.

The use of a single bush system is to be welcomed since it allows the shaft to be adjusted slightly in the sternbearing by moving the aftermost Plummer bearing. However, the alignment of shafting and setting of engines is still a difficult and tedious operation, and due to hull flexure when the ship is floated, there is no certainty of the achievement of good alignment in the sternbearing.

It is an interesting thought that in spite of the known unreliability of seals and bearings, the orthodox design remains completely inaccessible for servicing, unless the ship is dry docked and the propeller removed: a situation which greatly increases the impact and the severity of any emergency, such as a failure at sea.

It was the realization of this aspect which caused the development of withdrawable sterngear systems. The first two types, both British in origin,[17, 18] were developed in 1966, to be followed some years later by a Swedish and an American design. So far as is known the British designs, both of which use plain bearings, have achieved the greatest measure of acceptance.

The withdrawable sterngear confers two outstanding advantages:

(1) The bearing can be aligned to the shaft after completion of the ship. This makes for greater reliability of both bearing and seals.

(2) The bearing, and both inboard and outboard seals, can be withdrawn inboard for inspection, and repair if necessary, without the need to disturb the propeller or uncouple shafting.

Both of these operations can be carried out with the ship afloat, and at any draught, so that the ship is no longer dependent upon finding a drydock. Survey and repair time is much reduced, especially in the case of c.p. propeller installations.

The system introduced by the author's company has been fully described in an earlier paper[17] and it is not proposed to give further detail here, except to update the situation.

The system is in service with both c.p. and fixed pitch propellers, the latter of both flange and cone mounted types. Although any of the well known makes of seal may be employed, most of the sets supplied so far have the popular type of lip seal.

The system was first installed in a car carrying vessel *Laurita*, which entered service in January 1970, being followed by two sister ships in October 1970 and January 1971. At the time of writing these three vessels have aggregated over 16 years service, and apart from the single incident in the second ship *Torinita*, performance has been entirely satisfactory. It may be recalled that the *Torinita*, after six months in service, suffered damage to the bearing as a result of seawater entering the oil system and almost completely displacing the oil. Investigation showed that a plug, in contact with sea water, had been incorrectly made of mild instead of stainless steel, and had corroded away. However, although this incident caused some anxiety at the time, it did serve to prove the concept, because the bearing was withdrawn, taken ashore for repair and replaced while the ship was alongside the wharf, unloading cargo. The delay to the ship amounted to only 38 hours. After this repair and the correction of the offending plug, the vessel gave no further cause for anxiety, and all three ships underwent their four year survey without incident.

Altogether some fourteen sets are in service with a total service time of over 30 set-years, in vessels ranging from 5000 to 265 000 dwt.

The equipment has been withdrawn "underwater" on many occasions for inspection, demonstration, or training purposes, without any untoward incident. Experience to date has been that the time taken for withdrawal, inspection and replacement is about 8 hours, requiring the services of two or three men. That this can be done without the need to dismount the propeller or to uncouple shafting, has proved of particular value where c.p. propellers are fitted.

It may be mentioned that sets have been supplied recently for semi-submersible self-propelled drilling rigs.[19] Although the size of the shafts is comparatively small, it has still been possible to obtain adequate space for servicing. Because of the deep draught conditions, face type seals have been employed, both inboard and outboard.

For many years it has been customary to fit jacking oil to large heavily loaded bearings on landbased equipment to protect the bearing from damage during starting, stopping and operation on turning gear. Essentially this is the injection of oil at high pressure directly into the bearing surfaces to provide a separating oil film. It has always been an important design consideration that the jacking system oil inlets should have the least possible derating effect on the normal hydrodynamic lubrication. To this end it is normal practice to fit non-return valves to each injection point to avoid draining oil from the film under normal operation. Hitherto the design of the high pressure oil inlets on the bearing surface has been largely a matter of judgement and empiricism. The smaller the groove or oil chamber used for the jacking oil, the less the interference with normal operation of the bearing, but the higher the pressure required to separate the surfaces. Conversely, large chambers enable a low pressure to be used but require a high oil flow, as well as causing greater disruption under normal hydrodynamic operation. The important criterion is the pumping power required for the high pressure oil.

Computer studies have been carried out recently to arrive at the most efficient form of oil inlet. On the basis of minimum pumping power for the high pressure oil supply, and minimum disruption to the normal operation of the bearing, the best inlet arrangement is two short axial grooves spaced one on each side of the load line and 20° from it. This arrangement requires almost minimum power consumption with a peak pressure no higher than about 8 times the bearing specific load.

Some large vessels have been fitted with this system. The use of jacking oil will certainly prevent damage to the bearing, particularly when operating for long periods on turning gear, and if properly designed will even be effective under conditions of misalignment, but it must be stressed that it is not a substitute for correct alignment. In fact, if misalignment reaches a slope of ·0003, pumping power will be three to four times that required for the same bearing when correctly aligned.

OBSERVATIONS ON SOME PROBLEM AREAS AND THE DIAGNOSIS OF DAMAGE[20, 21, 22]

1. *Cast Iron Shafts*

The adoption of nodular cast iron for crankshafts was followed by cases of heavy and inexplicable wear in the bearings, and only after considerable investigation was the cause traced to the grinding operation. When this material is ground, due to the discontinuities in the surface caused by the graphite inclusions, the drag of the wheel causes filaments, or whiskers, to be formed at the edges of the inclusions, so that the resulting surface is rather like the pile on a carpet. Fig. 21 is a micro section showing one graphite nodule and the metallic filament. When the filaments were pointing in the direction of motion, the pile was, in effect, biting into the surface of the bearing and causing severe wear.

The solution was simply to arrange for the grinding wheel to rotate in the same direction as the normal rotation of the shaft, so that the pile was trailing, and no further trouble was experienced.

Regrettably it is not possible to adopt this solution where reversing or oscillating motion is involved. Honing or lapping in the direction opposing the filaments may remove them, but has not always been successful in practice.

It may be recalled that many years ago cast iron was abandoned for thrust collars because of similar wear problems, although the cause was not clearly understood at that time.

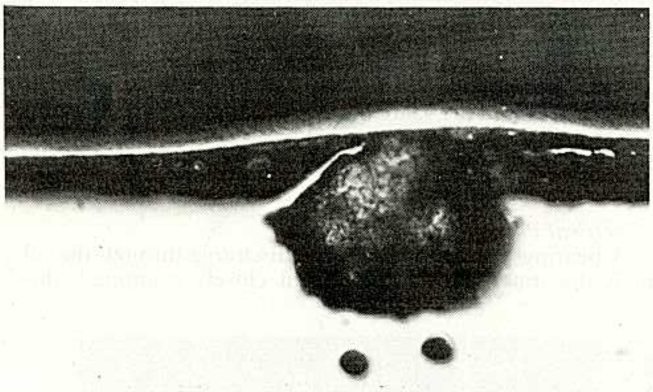

FIG. 21—*Micro section of filament on s.g. iron shaft*

2. *0·40 per cent Carbon Steel Shafts*

A problem similar in nature to the above was experienced some years ago when bearings supplied for large electric motors were subject to very rapid and inexplicable wear.

The cause was finally ascertained to be laminations at the surface of the journal and on machining down to a smaller diameter and fitting undersize bearings, the problem was cured. The rotor shafts were of 0·40 per cent carbon steel which, it seems, is susceptible to laminations at some distance below the forged surface. The practice of the parent company in England was to provide sufficient material on the forging to be certain of machining the laminations away. The subsidiary in Canada had not clearly understood this hazard and had ordered forgings which brought the laminated zone right on to the journal diameter.

3. *Nitrided Surfaces*

Bearing damage of a particular type is associated with shafts which have been hardened by nitriding and where the subsequent machining has not fully removed the layer of iron nitride. Fig. 22 shows damage from this cause with the characteristic 'V' formation. To avoid this it has been found essential to machine away at least 0·025 mm from the surface after nitriding.

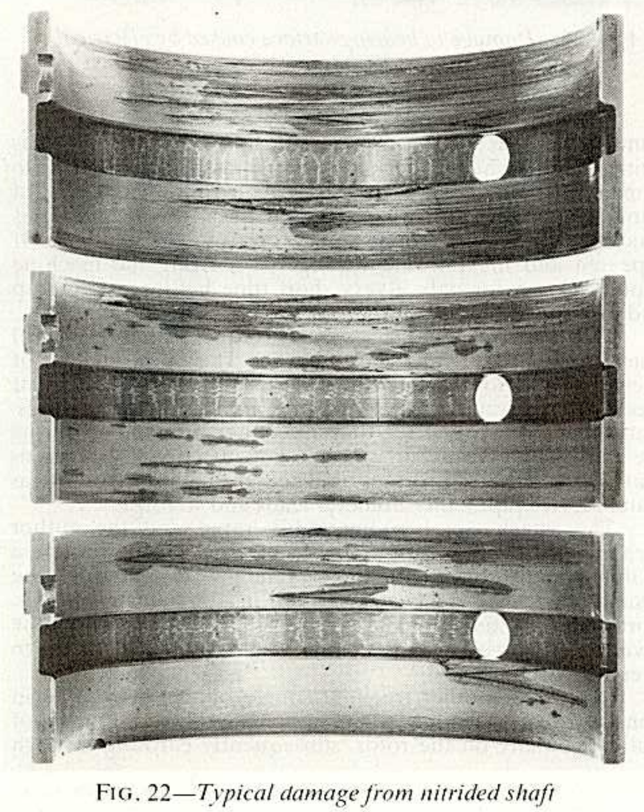

FIG. 22—*Typical damage from nitrided shaft*

4. *Failures Arising as a Result of Lack of Compatibility*

In describing the bearing system, emphasis was placed upon the necessity for mutual compatibility between the three elements. This is an area where there is little information readily available to designers so that pitfalls can be avoided. The following examples have been selected from the author's experience because they tend to be of a regularly recurring nature. The list is not by any means exhaustive.

(a) *Stainless steels*

It has generally been the experience that stainless steels of the 18 per cent chromium, 8 per cent nickel variety, have a very low compatibility with whitemetal oil lubricated bearings.

The tendency is perhaps rather worse with tilting pad journal bearings than with plain bearings, but in repeated instances wiping, pick-up and seizure have occurred with these steels. In many instances the bearing manufacturer was not aware of this shaft material until the failures occurred, but in all instances the problems were curable by reducing the shaft diameter and fitting sleeves of plain carbon steel shrunk on.

The problem is naturally worse at high specific loads but has been known to occur at quite low loads. The author has been told that results obtained with lead-base whitemetal were definitely superior to tin-base, although even then the situation was sensitive. Attempts which have been made to improve the adhesion of the oil by addition of fatty acids such as stearic acid were generally unsuccessful.

The only safe action would appear to be to avoid such steels with whitemetal bearings.

(b) *Machining failures*

Some years ago there was a spate of failures of similar type variously termed black scab, wire wool or machining failures.

The symptoms were the same: a hard black scab was formed on the whitemetal surface and the co-operating surface, journal or thrust collar, was machined away with the formation of masses of thin filaments looking like wire wool. The scab grew as the surface was machined away so that the failure was self-propagating.

In the beginning the failures were associated very largely with shaft materials of the 3 per cent chrome/molybdenum variety but in the investigative work carried out to find the cause of these failures it was found that the failures could be generated on most grades of steel, those containing above 2 per cent of Ni/Cr/Mo being most susceptible. The trigger mechanism was usually a hard particle becoming wedged in the bearing surface and penetrating the oil film to touch the shaft. From this small beginning the failure grew with great rapidity.[23, 24]

Fig. 23 shows a typical example of black scab damage.

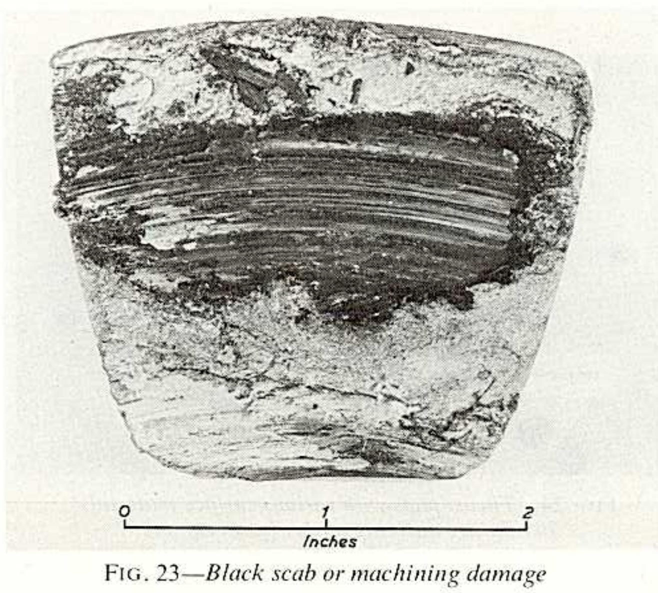

FIG. 23—*Black scab or machining damage*

One does not hear much about these failures at the present time so perhaps we have found out how to avoid them.

All this work was carried out with steel shafts, but the author recalls a similar experience some years earlier where this same type of failure occurred with shafts made of nickel/aluminium bronze. Although there was little evidence of surface heat, the failure produced quantities of thin filaments or wire, of generally an oval section, one of which was even a jointless ring encircling but quite free from the shaft, which indicated that the mechanism appeared to be as much swaging as turning.

The failures could be repeated with certainty at specific loads as low as 0·6 MN/m^2 (90 lbf/in^2). Various oils were tested without success. After tin and lead base whitemetals, various other surfaces were tried, including bronze, cast iron, hard chrome plate and hardened steel. The last named gave the best results but still produced a hard seizure after only a short period. Apart from the shaft material the bearing was of entirely normal construction, and the problem was finally solved by changing the shaft to a different material.

(c) *Nickel and chrome plating*

Occasionally shaft journals are damaged, or accidentally machined undersize, and then it appears to be an attractive proposition to restore them to size by nickel or chrome plating. The author understands that shafts recovered by nickel plating have been operated successfully in automotive engines, but apart from this his own experience with nickel is entirely unsatisfactory. In all instances where rotating machines had journals built up by nickel plating, the results were unsatisfactory, scuffing and seizure occurring within a few hours operation.

Hard chrome plating, applied on top of nickel has, however, been found to give good results when the process has been properly carried out.

(d) *Tin oxide corrosion*

In the early 1960's several cases of severe corrosion occurred in tin-base whitemetal bearings. It was found that the surface was coated with a hard, dark grey, rather brittle deposit which broke away from the surface leaving uneven pits, finally leading to wiping and complete breakdown.

Lengthy laboratory investigation finally suggested the cause to be the presence of small quantities of sea water in the oil. One of the peculiarities of these cases was the sudden appearance of the damage in ships which had been in trouble-free service for many years. The incidents seemed to cease almost as quickly as the troubles appeared and the author has not heard of this problem for some years now.

(e) *Extreme Pressure oils*

One problem which appears to be of a recurring nature is associated with the use in gearboxes of certain types of E.P. oil. The problem is demanding attention at present because of attack in turbine thrust bearings. Fig. 24 shows a mixture of pads lined with different materials which have been run in the thrust testing machine as part of the investigation. It may be observed that lead-base whitemetal (GM. 155) is very heavily corroded whilst tin-base whitemetals (L2 and GM. 184) are corroded but to lesser extent. On the other hand, 40 per cent tin-aluminium (AS.40 and AS.45) show no traces of corrosion. Tests of the same materials immersed in the oil for a period of 80 hours at the same temperature as the bearing surfaces, show no corrosion, so that phenomenon seems to be related to high pressure and in particular, high rates of shear.

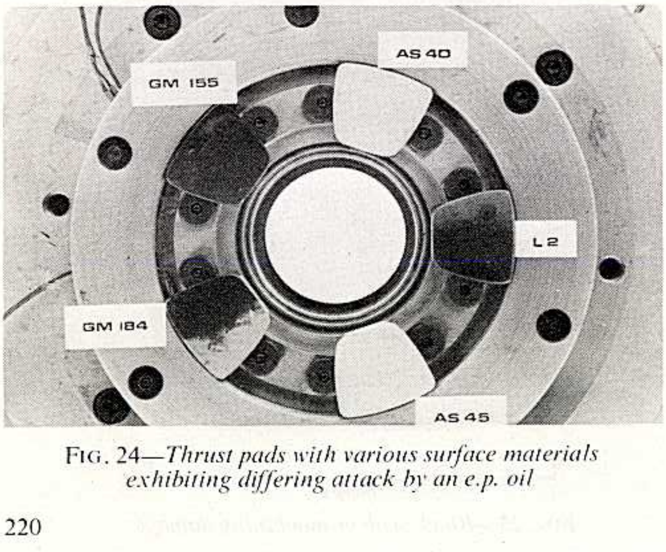

FIG. 24—*Thrust pads with various surface materials exhibiting differing attack by an e.p. oil*

5. *Electrical Potential*

A bearing damaged by electric discharge through the oil film is illustrated in Fig. 25. When closely examined, this damage consists of uniformly distributed pitting, the pits being generally hemispherical, with the intensity increasing to a maximum in the zone of thinnest oil film. This cause of damage is often difficult to isolate because the pitting is quickly followed by wiping which destroys the evidence. If repeated and inexplicable wiping is occurring the machine should be run for only a very short time before opening up and examining the bearing.

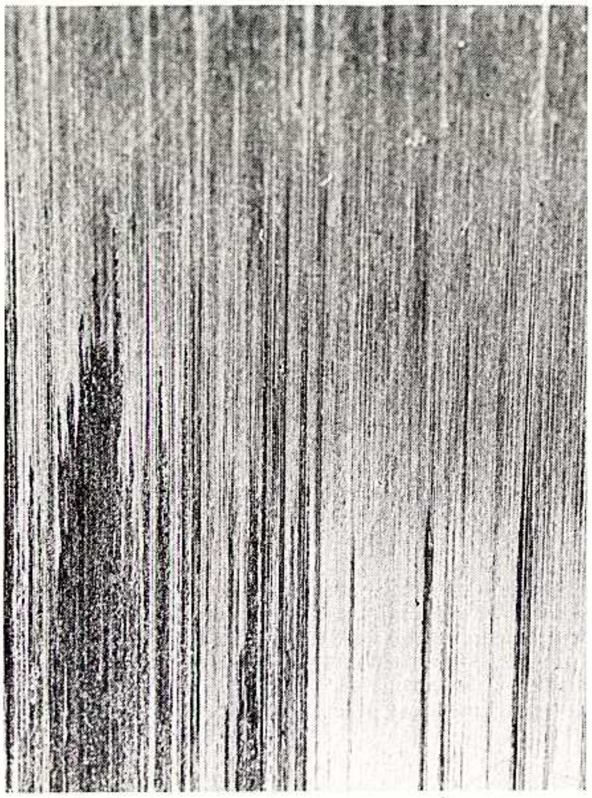

FIG. 25—*Damage to bearing surface caused by electrical discharge*

This type of damage occurs frequently in electrical machinery, due to stray shaft currents. The usual method of prevention is to insulate the non-driving end bearing of electrical machines, and sometimes in fact both bearings. Care must be taken to see that all connexions to the bearing are thoroughly insulated, since it not infrequently happens that an electrical path is established through such items as water service pipes, thermometer leads and so forth.

The attack can be remarkably rapid and the author recalls an instance in a large hydro-electric machine where a spanner carelessly left on the bedplate so as to bridge the insulation, caused the bearing to fail in a matter of minutes. Fortunately no one thought to remove the spanner before the investigators arrived, so the cause was found quite easily, to everyone's relief.

However, another frequent source of this damage is in non-electrical machines where it is caused by build up of static electricity on the rotor, subsequently earthing through the bearings. This has been known to occur on various types of fan and sometimes on turbines. The usual precaution is to fit an earthing brush close to the bearing.

6. *Fretting*

In dynamically loaded bearings, fretting will occur on the back or support surface where the interference fit or "nip" is insufficient for the dynamic forces involved. It may also be caused by a housing which is insufficiently rigid for the load cycle involved. Fig. 26 illustrated a typical example occurring in a medium speed diesel engine.

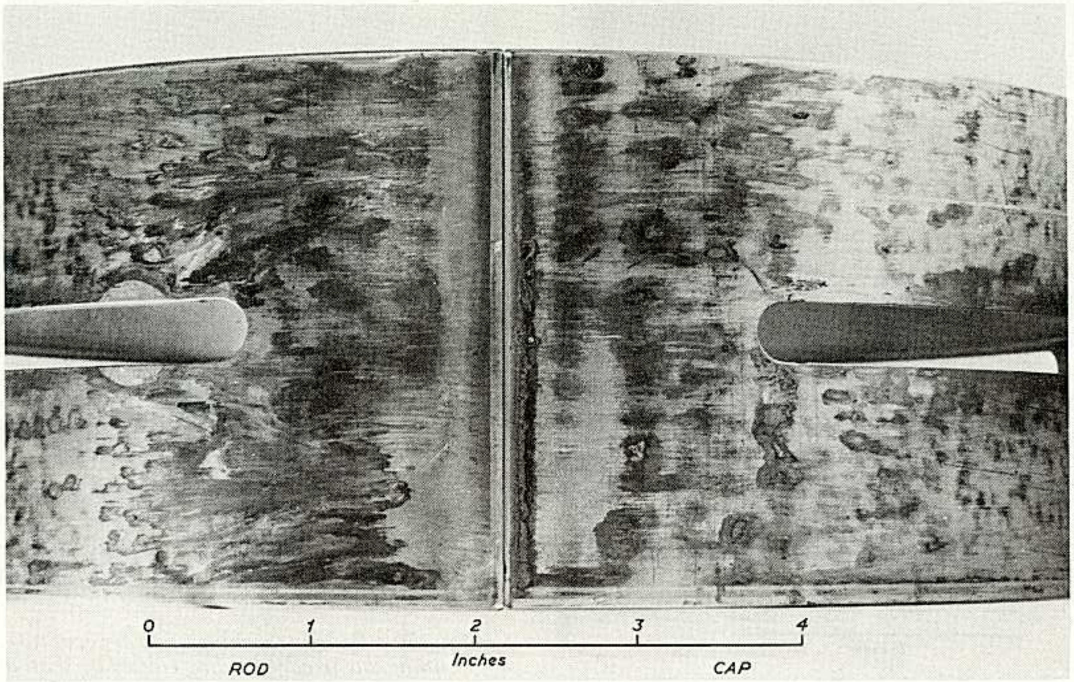

FIG. 26—*Fretting damage on back of bearing due to inadequate interference*

Pivoted pad bearings are peculiarly liable to fretting of the pivot if subjected to dynamic loading, such as may occur in a thrust bearing subjected to axial vibration or where the collar is not running true, or in a journal bearing subject to a rotating load due to out of balance forces. Fig. 27 illustrates fretting on a thrust pad pivot, caused by axial vibration.

Yet another form of fretting occurs on the bearing surfaces due to static vibration — i.e. where the machine is not in operation but is subject to vibration from the surroundings. There have been severe cases in multiple installations of diesel generating sets on shipboard, where vibrations transmitted from the operating sets through the structure have caused fretting on standby sets. Fretting has been known to occur in machines during transit and this confirms the need always to assemble bearings with well oiled journals to minimize such risk. Fig. 28 illustrates a typical example.

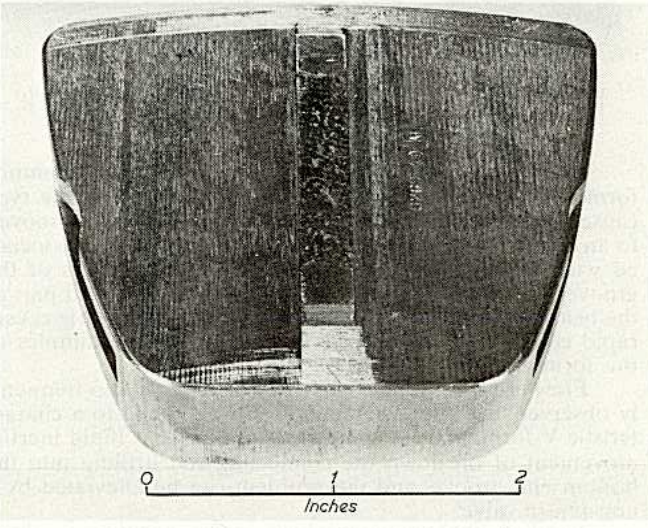

FIG. 27—*Fretting on thrust pad pivot due to axial vibration*

7. *Corrosion*

Fig. 29 shows the severely corroded surface of an unplated copper-lined lead bearing caused by attack of the lead phase by acidic oil oxidation products. It is normal practice to protect such bearings by a lead-tin overlay but the attack can commence where the overlay is worn through or scored. Very often, early stages of the corrosion result in the onset of fatigue in the resulting porous copper matrix.

Tin-aluminium bearings are not attacked by acidic oil products but an instance is known where corrosion attack followed leakage of an alkaline antifreeze mixture into the lubricating oil. Frequent monitoring of the oil condition is advisable.

8. *Fatigue*

Bearings carrying high dynamic loads are liable to fatigue damage, often caused by a concentration of load due to mechanical imperfection — e.g. poor geometric form, misalignment, distortion, swarf trapped behind bearing and so forth. Fig. 30 shows a typical example due to edge loading.

Whitemetal bearings are particularly prone to fatigue since any high loading not only increases the stress in the lining but the associated temperature rise reduces its strength.

9. *Wiping of Bearing Surfaces*

Wiping is evidenced where the bearing surface shows rubbing, melting or smearing. The cause is overheating and may be due to insufficient running clearance, inadequate oil supply or supply failure. It may follow disruption of the oil film due to extreme loading, excessive dynamic forces or local high spots. It can occur in bearings starting and stopping under load where either the specific loads are too high or the acceleration and deceleration are too low.

Fig. 31 shows wiping due to load concentration resulting from a barrelled journal.

Fig. 32 shows repeated wiping of a thrust pad, in thin layers, caused by excessive load at start up.

10. *Cavitation Erosion*

The form in which this type of damage commonly occurs differs with the type of engine.

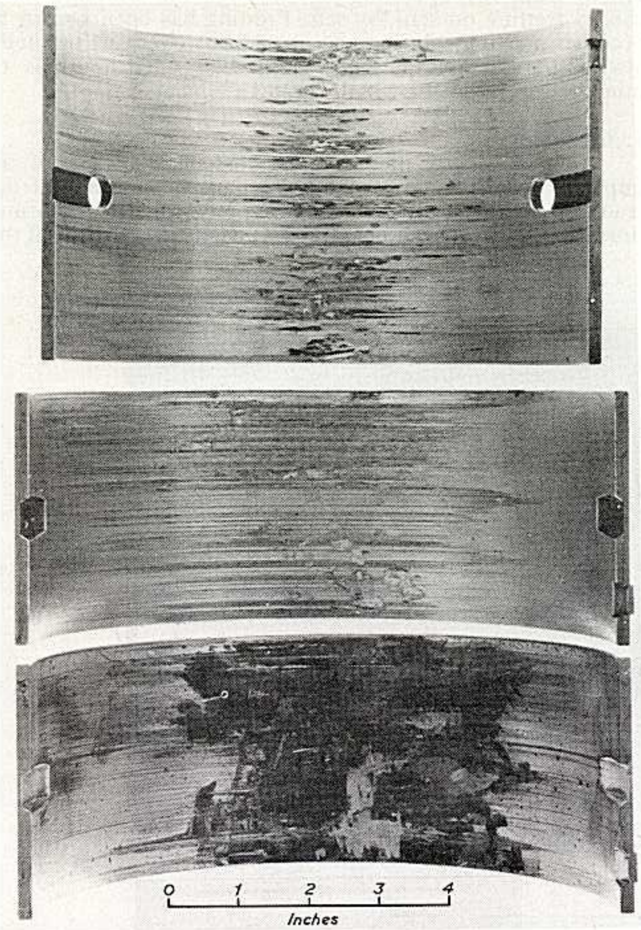

FIG. 28—*Fretting of bearing surface due to static vibration*

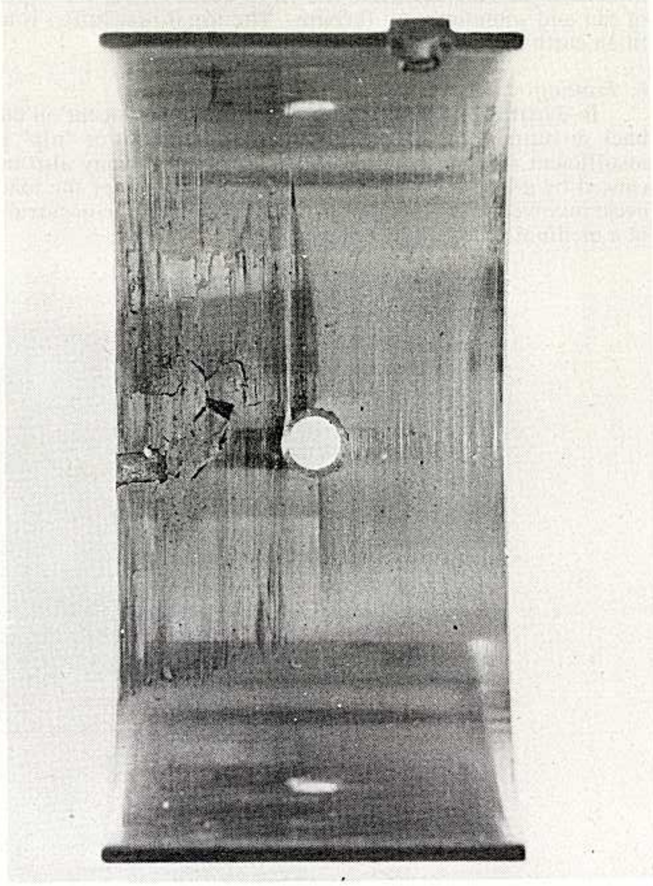

FIG. 30—*Fatigue cracking of 20 per cent tin-aluminium lining due to edge loading caused by shaft deflexion*

FIG. 29—*Corrosion of copper-lead lined bearing (not overlay plated) caused by attack of lead phase by acidic oil oxidation products*

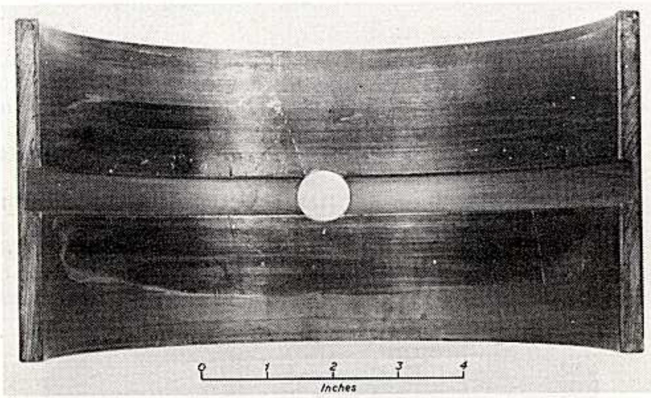

FIG. 31—*Overlay plated copper-lead bearing wiped due to barrelled journal*

In medium speed diesel engines the most common form is that associated with flow discontinuities of the type caused by the passage of a crankshaft oil hole from a grooved to an ungrooved part of the bearing. The damage is associated with areas of low pressure in the oil at the ends of the grooves, or, sometimes further around the ungrooved part of the bearing when a reflected pressure pulse in the hole causes rapid collapse of the bubbles already formed. Examples of this form of damage are shown in Fig. 33.

Flow erosion of the edges of oil grooves is also frequently observed, see Fig. 34, sometimes progressing to a characteristic V-formation. In some cases the cause is rapid inertial movement of oil down the connecting rod drilling into the bottom end groove, and the problem can be alleviated by a non-return valve.

Another phenomenon which seems peculiar to these engines is referred to as "dispersed" erosion. This appears as a rippling effect rather like the sand ripples on a beach as the

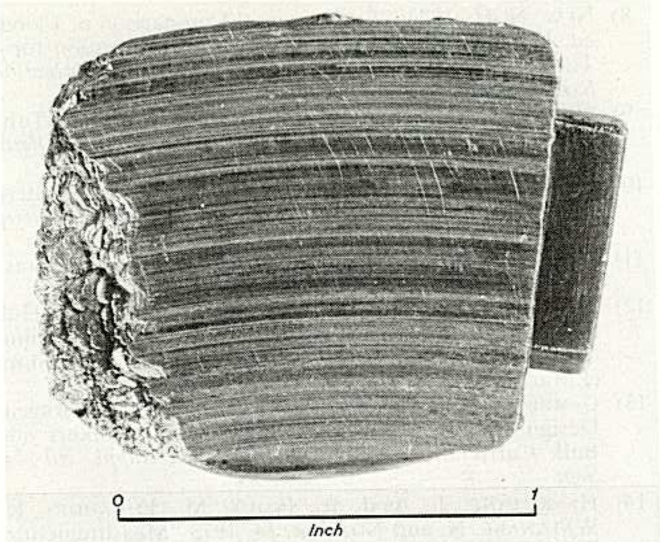

FIG. 32—*Successive surface wiping of thrust pad, in thin layers, due to excessive load at start up*

tide recedes. It is thought that the ripples are caused initially by bulk movement of the low modulus overlay under the influence of the oil film pressure, the erosion damage then resulting from the collapse of bubbles nucleating on the asperities formed by local fatigue of the high spots. It is known that bubbles will nucleate on sharp edges of grooves or even along score marks produced by dirt particles.

11. *Thermal Ratcheting in Whitemetal (Fig. 35)*

This is the deformation of tin base whitemetal by alternate heating and cooling. It is due to the anisotropic nature of tin crystals which have different coefficients of expansion along each axis. Repeated thermal cycling can cause faceting which can result in undulation in excess of 0·025 mm in severe cases. Its occurrence is generally an indication of high bearing temperatures.

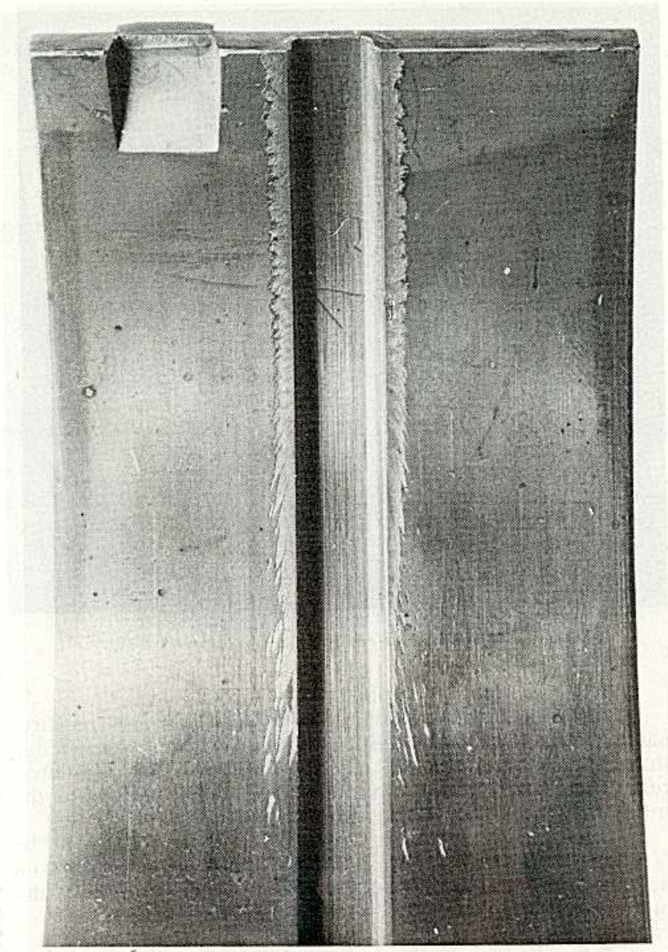

FIG. 34—*Cavitation erosion of a cap half (oblique split) overlay plated bottom and bearing from pressure fluctuation in connecting rod drilling*

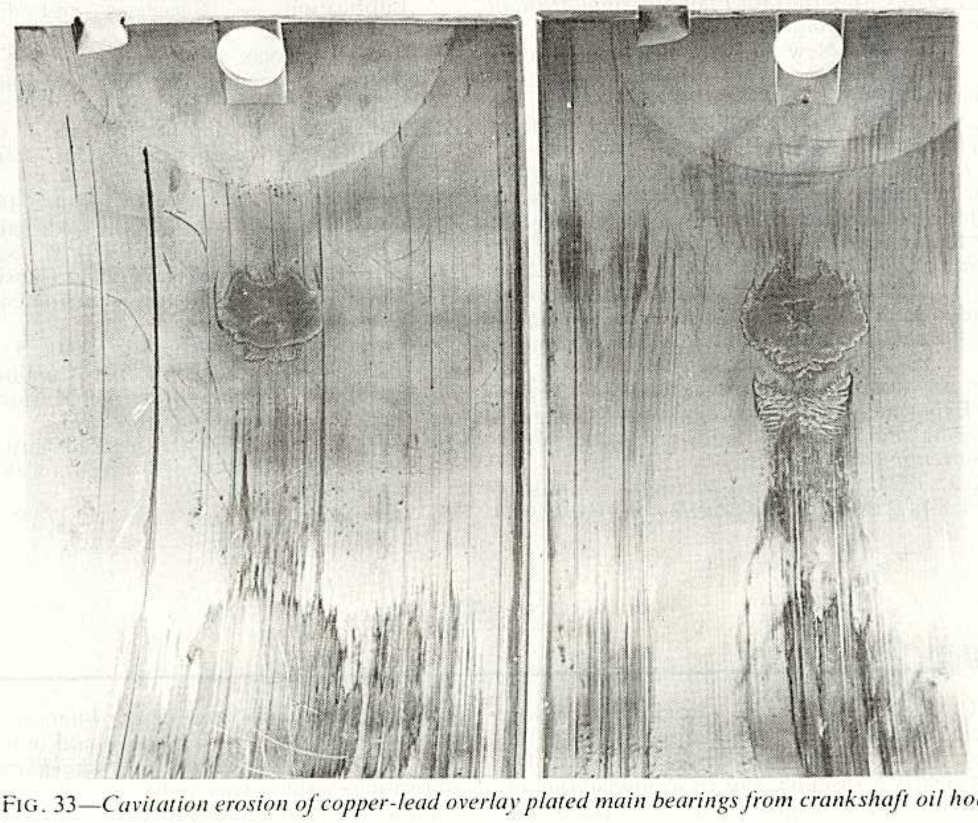

FIG. 33—*Cavitation erosion of copper-lead overlay plated main bearings from crankshaft oil hole*

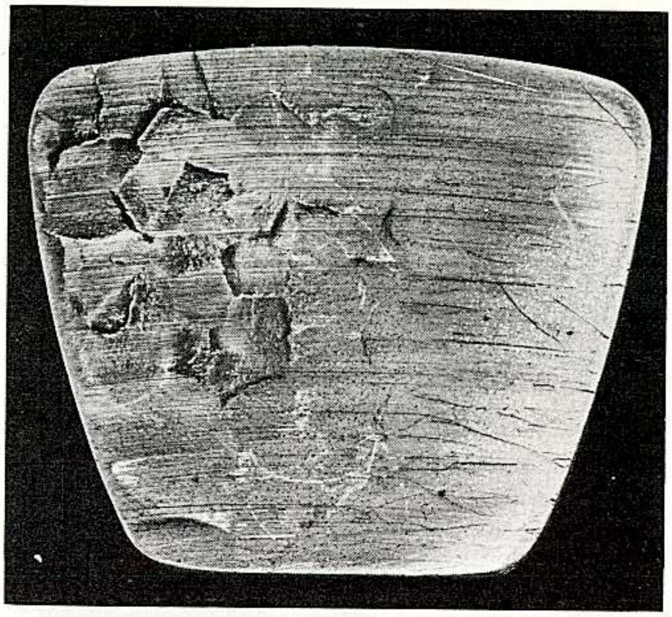

FIG. 35—*Thermal ratcheting in tin base whitemetal*

CONCLUSION

The high rate of technological development requires each newcomer to engineering to acquire more knowledge than his predecessors. In reaching out to grasp the latest refinements it remains equally important to assimilate the more rudimentary knowledge and experience which exists.

Much of the ground covered by this paper must be familiar to many, but if it, in any way, assists the reader to a better understanding of the subject or to avoid some of the pitfalls, the author will have achieved his purpose.

ACKNOWLEDGEMENTS

The author wishes to thank The Glacier Metal Company Limited for permission to publish the material in this paper, and all the staff whose efforts have produced the data used.

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Discussion

MR. P. J. C. MACK, F.I.Mar.E., said he was indeed privileged in being asked to open the discussion on this excellent paper on Modern Bearing Design and Practice. The subject was quite clearly a vast one, and he would confine his remarks to medium speed engine bearings because he thought this was an area of design where we had more trouble than enough. A marine bearing would only perform satisfactorily if its design was right for the particular engine, the lubricating system was right, and the designed operating conditions were maintained.

The author had boldly and honestly stated that in the pursuit of ever increasing specific power output, engine designers had been forced to forsake whitemetal "the best all round bearing material" and resort to prefinished steel backed copper and aluminium lined bearings. We should also bear in mind that the same designers had produced engines which were required to burn heavy fuel oils, likely sometimes to be of dubious composition, over which the seagoing engineer had little control and where the swept volume of the trunk piston formed an integral part of the crankcase. In connivance with the oil chemist, base lubricating oil was now required to be adulterated by an assortment of additives each intended to perform a particular function. A modern diesel engine, for example, was intolerant of water in the highly fortified lubricating oil whereas its predecessors had been known to soldier on with as much as 30 to 40 per cent water contamination without damage. Foaming, emulsification and bearing corrosion of the varieties now being experienced were relatively new problems as also was the requirement to maintain what amounted to almost clinical purity of the lubricating oil — a pious hope.

Many of these so called advances had had the effect, directly or indirectly, of limiting the tolerances on operational safety. Plant reliability was now to a large extent dependent upon the standard of running and maintenance. Mr. Mack considered statutory machinery surveys, by either the Department of Trade or the Classification Societies, to be but secondary although complementary, to the owners planned maintenance arrangements. Reduced manning scales and limited port time might contribute to a growing backlog of maintenance ultimately at the expenses of the safety of a ship whereas on the other hand surveys might be up to date. Piecemeal maintenance and bearing replacement could have disastrous results in the event of excessive local or even overall bearing wear.

He thought it was Somerset Maugham who had said that the most useful thing about principle was that it could be sacrificed to expediency. Short sighted expediency however would not always ensure long term ship reliability and profitability.

Many of these new problems were being encountered by operating staff who had neither the experience and awareness of the fundamental and significant differences between the thin shell bearing and its heavy whitemetal predecessor, and for this reason the paper was both timely and welcome. For the benefit of operating staff a paragraph or two on bearing assembly would have been beneficial.

The incidence of tin oxide corrosion was not uncommon even today. In fact we had had this in the majority of passenger ships under the UK flag in the course of the past few years particularly on the main thrust bearing pads. He would suggest that, because this condition was now so well known and readily identified, it was more often than not dealt with before serious bearing damage had taken place — without feedback to either the engine builder or the bearing manufacturer.

A less widely known type of bearing damage was associated with bacterial infestation of the lubricating oil, and in Southampton during the past year there had been at least two such instances in both steam turbine and diesel systems. Apparently these bugs had a short enjoyable life cycle during which they thrived on a diet of either the base oil or its condiments, breeding prolifically in the process. The visual symptoms were choking of filters, and darkening of the bearing surfaces and journals, by the soft impacted corpses which were fairly corrosive to steel. The prescribed treatment appeared to require numerous oil changes coupled with the application of bug killing additives.

Fig. 9 of the author's paper was a reminder of one aspect of the thermal transients which occurred and of the desirability of avoiding an excessive rate of application of power. Extended running at idling speed should also be avoided where possible and for similar reasons bridge controlled manoeuvring power might be generally limited to about 45 to 50 per cent of maximum power.

The author had stated that despite improvements in design and materials of construction, crosshead bearing still operated with too small a margin of safety. The same might be said of thin walled main and bottom end bearing in terms of their all too frequent failure rate, sometimes involving not only crankshaft renewal, but the risk to life in the event of a crankcase explosion. Bearing failures might be associated with alignment, lubricating oil problems or incorrect assembly.

In view of these new operational risks, equipment had lately been marketed for check testing and continuously monitoring crankcase lubricating oils for water content. Likewise equipment was now available for continuously monitoring the level of metallic debris in crankcase lubricating oil. Lubricating oil suppliers and at least one engine builder offered a diagnostic analysis of the used crankcase oil. Crankcase oil mist detectors were being marketed with improved sensitivity. Development had recently taken place aimed at improving the sensitivity of bearing temperature monitoring equipment.

Many years ago following the disastrous *Reina del Pacifico* crankcase explosion and intensive follow up research it was recognized that temperature monitoring would provide the best means of prevention but that it was quite impractical to monitor the temperature of all moving parts within the crankcase. The protective measures which were then introduced were limited to the mandatory provision of crankcase relief valves and the optional use of oil mist detectors. In view of the rapidity of final failure and seizure of thin shell bearings, temperature probes had sometimes been fitted adjacent to main bearings to monitor the mean temperature of oil returning to the crankcase from one side of the bearing.

However there had since been a number of serious machinery casualties involving the rapid seizure of thin shell bearings when neither oil mist detectors or bearing temperature probes of this type gave sufficient warning. The criticality of time in relation to these events was perhaps best illustrated by the hypothetical curve showing the salient points of rapid bearing overheating, (see Fig. 36).

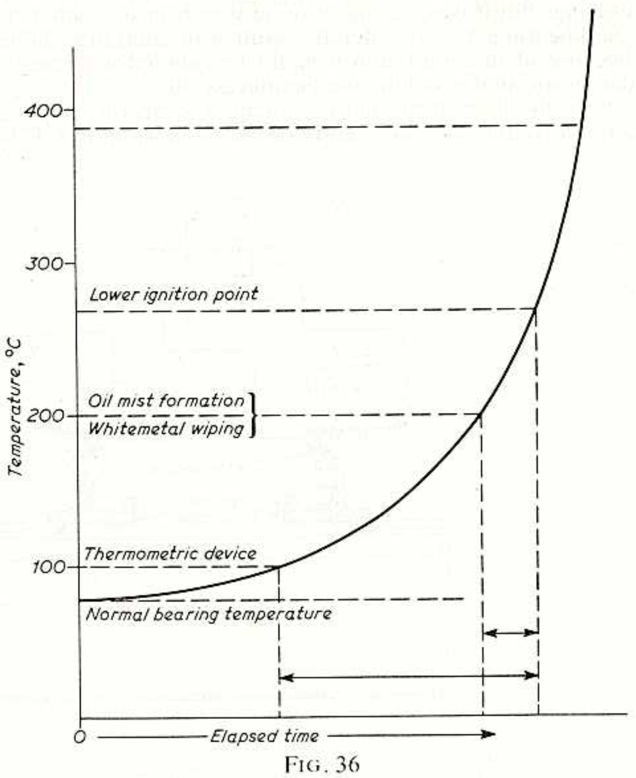


FIG. 36

At the origin a steady state bearing surface temperature of about 70°C had been assumed and that from this point the bearing started to overheat.

At temperatures above 100°C tin and indium began to migrate from the overlay. At temperature above 150°C oil degradation would commence.

It was of interest to note that at 200°C white metal melting and oil mist formation began; but when this happened there was very little time left for an oil mist detector to respond to it. When a white metal bearing overheated it was often relieved by wiping. In the process of total failure the white metal bearing would generally give ample audible prewarning, provided the engine room was manned and it would most likely generate a detectable quantity of oil mist. This did not appear to be so in the case of thin shell bearings where the onset of pick-up followed closely on the heels of overheating. Since the white metal overlay was originally less than 0·002 thick, seizure between the bearing lining and the steel journal followed rapidly in one continuous operation. This would probably produce an even sharper change of slope than as shown.

In relation to the response characteristics of oil mist detection equipment one could readily see the difficulty in achieving a prompt and safe response to rapid overheating of thin shell bearings, or even aluminium pistons, when the elapsed time between mist formation and detection might be greater than between mist formation and seizure.

The equipment for monitoring of metallic wear debris was extremely sensitive and would extend the elapsed time giving sufficient prewarning. If we were monitoring bearing temperature at, say, 100°C the difference in the elapsed time could quite clearly be seen. Any wear debris which was electrically conductive whether ferrous or non-ferrous was capable of detection. This type of detector promised to be useful as a course diagnostic tool as well as a valuable piece of safety equipment. If, however, rapid overheating was associated with lubricating oil failure to any particular bearing, the debris might not reach the detector grid in time.

Monitoring for water content alone presupposed that the presence of water in oil would be the sole mechanism of bearing failure although this had certainly been so in numerous instances.

It would seem that temperature monitoring of the bearing shells promised to be the best method of detecting overheating and averting bearing damage. Rising temperature was the one ever present feature of bearing failure. For practical reasons however its application was limited to main bearings but it was seizure of these very bearings rather than crankbearings which generally resulted in crankshaft failure. Because of this limitation it might be coupled with periodic diagnostic analysis of the used crankcase oil.

In the light of present day changes of bearing design an alternative type of temperature probe as shown in Fig. 37 had recently been developed. The probe differed from its predecessors in that it was completely buried in the bearing housing and fairly close to the interface of the shell bearing with the housing. The response to temperature change was very sensitive. He had been told that even a few degrees difference across the oil cooler could be picked up by the probe. These probes were being fitted to the PC2 and PC3 engines manufactured by Crossley Premier Engines. The alarm point of 80°C was set at 7°C above the full load running temperature and the shut down signal was given at 85°C giving an even further spread of the elapsed time and earlier prewarning.

The suitability of any of these devices would of course be dependent upon the particular machinery and the statistical risks. He would hasten to add that the medium speed engine had a relatively small and strong crankcase, as compared with the direct coupled slow speed engine, and because of these features the consequences of a crankcase explosion in a medium speed engine were generally less severe than in its slow speed engine counterpart.

Mr. Mack added that these were his own views and not necessarily those of the Department of Trade.

Mr. A. Galloway, F.I.Mar.E., said it was unfortunate that a great deal of information was lost to bearing manufacturers and designers due to lack of feed-back from ships, especially after the guarantee period. Reduced manning scales and automation had tended to eliminate the once common practice of examining bearings as a routine maintenance check, and opening up bearings was nowadays virtually confined to classification survey requirements. The result of this had been that bearing failures had only been noted when the failure was serious and the wiping which had taken place had obscured any indication which might have existed in regard to the actual cause.

The author's company was well known for the cooperation it extended and Mr. Galloway suggested that Superintendents and Chief Engineers, rather than merely attribute the wiping of a bearing to dirt, contaminated lubricating oil, or negligence of crew or repairers, should take advantage of the expertize offered and report all bearing failures in full — giving all the relevant details. If this could be implemented on a large scale, the continuing feed-back would benefit all concerned.

One other point which he considered relevant was that whilst ordinary lined bearings could be coped with by most sea-going engineers, the trend to thin shell bearings and more exotic materials created a situation where replacement bearings might be fitted incorrectly. There would appear to be a need for specific fitting instructions to be supplied with all bearings. The instructions would require to be in several languages, but the benefits accruing from having replacement bearings fitted correctly could be substantial.

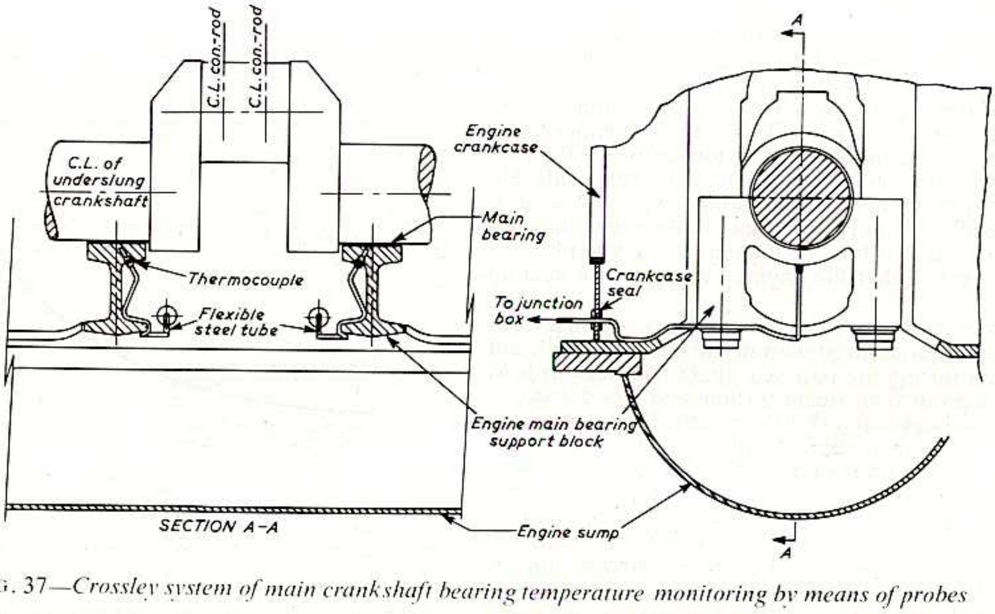


Fig. 37—*Crossley system of main crankshaft bearing temperature monitoring by means of probes*

Mr. R. C. Beckett, F.I.Mar.E., said that damage problem had been encounted with thin-shell, tin-aluminium lined bearings on the primary shafts of geared diesel gearboxes due to score marks and too tight a fit in the housing, after sea trials, which had to be replaced.

Subsequently modified bearings having a slight relief machined across the joint faces of the two halves of the shell to increase oil clearances were fitted, this practice being a recommendation of the gearbox makers.

After further trials, examination of the new bearings revealed slight damage only. Such damage was represented by indentation markings comprising two continuous lines starting at the extreme corner of the oil groove in the bottom half shell and finishing up at the top half shell. The cause of the markings was considered to have been the ingress of foreign matter (within the gearbox housing) which had entered the bearing through the oilway passages. Similar experience had been encounted previously with new gearbox units during the running-in period.

To avoid repetition of this kind of damage problem, would the application of the use of a higher viscosity lubricating oil — but only during the critical stage of running-in — be considered beneficial?

The problem of sterngear systems was equally recognized by many shipowners due to loss productivity of their vessels.

The proprietary sterngear arrangement, which permitted accessibility to bearing and sterntube sealings whilst the ship was afloat, offered a positive benefit compared to the conventional two bearing sterntube systems, of which the first (and often the only) method by which a shipowner could detect that something was wrong within the sterntube assembly was related to the lubricant.

Statistical evidence showed that at least once in the life-time of a ship (20 years) she would suffer sterngear troubles. Therefore, a proprietary split bearing type as described in the paper, offered an economic consideration, since the increased costs of drydocking and repairing together with off-hire time could well justify the initial cost of this type of sterngear which in terms of ship service years might well show a profit.

MR. J. A. DUNCAN, F.I.Mar.E., commented that the paper had generally impressed him as a comprehensive account of the present state of development of its subject. Its basic approach was factual and made commonsense, discussing the various aspects adequately although in some instances rather briefly. From the contents it could be concluded that most of the problems which had arisen up to the present in all types of rotating shaft bearings had yielded to the extensive research and development that had been carried out over the last twenty five years. It would appear also that as a result of this work means were available for dealing with future advanced design.

However, in the marine diesel engine field, which was his principal interest, while high and medium speed engines had received attention, and as a result of the introduction of the thin-shell insert bearing in the early fifties had had their performance greatly improved, in the case of the large slow running turbo charged two-stroke engine it was disappointing to note the author's report, under the heading Diesel Engine Bearings, that the top end bearings had "for some years been among the less reliable components" going on to say that "so far it has not been found possible to produce any bearing theory which enabled an assessment of the operating characteristics to be made. The design had largely evolved by experience". His Fig. 11 illustrated what was believed to be best design practice, i.e. the thin shell insert bearing arrangement; and much related detail was given in the text.

Speaking as a former superintendent engineer who had had to cope with the problem of the crosshead bearing in the modern large turbo charged two-stroke diesel engine during the fifties and sixties, and as one who had then pressed for the trial of thin shell bearings in that area as a possible solution to the problem, it seemed that the author was taking a discouraging view on the matter. When one considered that the type of engine involved came into general service about the middle of the fifties, and that the presence of persistent crosshead trouble was recognised early in the sixties by many owners' superintendents, and that by that time the design, manufacture and performance of the thin shell bearing in high load conditions were firmly established, Mr. Duncan found it surprising that it was not adopted earlier than it had been, and that a more confident view of its capabilities in this context had not been expressed by the author at this time.

To emphasize this view Mr. Duncan would draw Mr. Hill's attention to a report, by Mr. A. G. Ginty of B.S.R.A.* The study was initiated early in 1968 because of concern about the prevalence of white metal surface fracture failures leading to catastrophic failure of the bearings. The object of the work was to establish the extent and frequency of this type of failure so that it might be recognized and dealt with not as a random event to be handled on a normal maintenance basis, but as a frequent (and dangerous) repetitive failure, due to increased engine loadings, and requiring radical redesign of the bearing to eliminate the weakness.

Briefly, the survey showed that in the case of 160 ships, all built from 1960 onwards, in a total of 1541 crossheads at risk 812 crossheads suffered crippling failure. Of those 812 cases, 471 were typical failures by craze cracking of the surface of the metal, similar to those shown on the author's figure 30, in the paper, but more extensive than edge loading. The remaining 341 cases were classified as from other causes because craze cracking was not observed. But, because later stages of craze cracking led to overheating and wiping, and so on, it was possible that many of those other failures did in fact start as craze cracking. The sample of 160 ships represented about 78 per cent of the total returns for the survey, and all the crossheads had conventional design bearings. Table IV showed numbers of crossheads at risk year by year from 1960 to 1971; the number of incidents each year; and the annual number of incidents as a percentage of the total at risk each year. Fig. 38 showed graphs of the tabulated percentage figures. From there the trend of incidence could be seen. Nil returns for the first two and a half years was in accord with

TABLE IV

Year:	1960	1961	1962	1963	1964	1965	1966	1967	1968	1969	1970	1971	Totals
Total Crossheads	62	171	284	369	447	604	762	909	1110	1203	1313	1541	1541
Incidents 'D'	—	—	—	6	5	19	61	62	90	85	81	62	471
Incidents 'O'	—	—	—	5	5	44	20	48	56	42	58	63	341
Incidents 'D' + 'O'	—	—	—	11	10	63	81	110	146	127	139	125	812
Percentage of No at Risk 'D'	0	0	0	2·1	1·4	3·5	9·1	7·3	9·3	8·1	6·5	4·4	
'O'	0	0	0	1·8	1·4	8·1	3·0	5·7	5·9	4·0	4·7	4·5	
'D' and 'O'	0	0	0	3·9	2·7	11·6	12·1	13·0	15·2	12·1	11·3	8·9	

* "A Survey of Crosshead Bearing Failures In Slow Speed Marine Oil Engines for the period 1960–1971" *M.E.R. November 1974.*

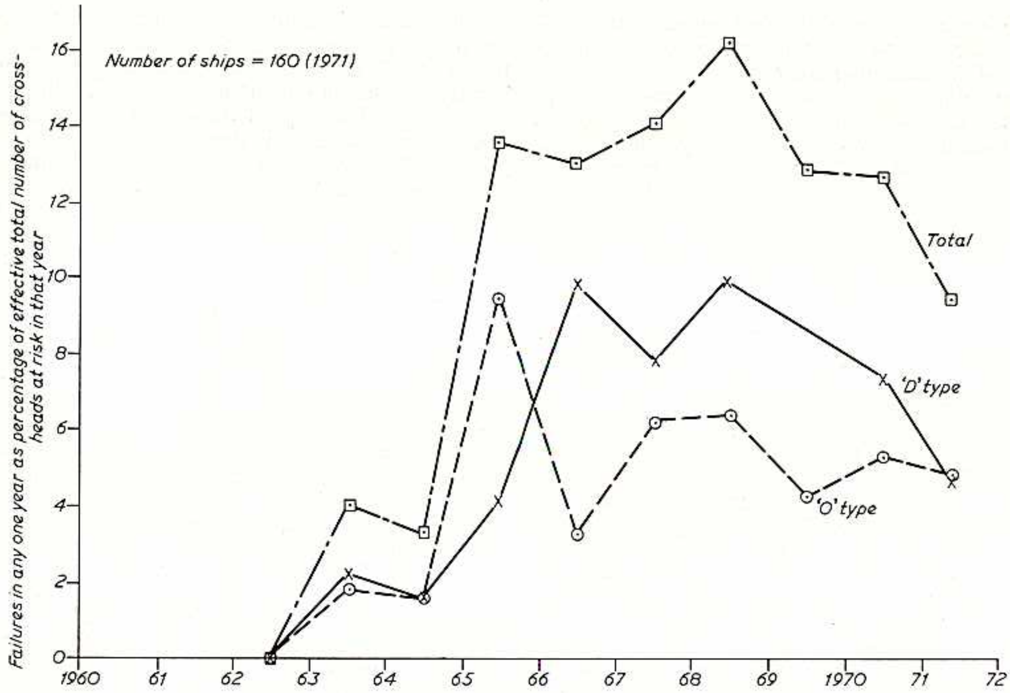


FIG. 38—*Summated data*

the average time required for the development of trouble from the "as new" condition. Thereafter there was a rapid rise to peaks in 1967 and 1968, followed by a decline towards the early 1970s. The appearance of this abatement by the early seventies could probably by accounted for by:

a) recognition of the nature of the failures leading to improvement in bearing manufacturing practice by engine builders;
b) recognition of the problem and re-organization of maintenance practice by owners ensuring shorter intervals between inspections of bearings, so preventing failure in service.

Neither of these approaches, of course, solved the basic problem: which apparently was still with us.

The author had stated that some research on the subject had been carried out but this had been limited in scale owing to the high cost of full scale investigation in an engineering laboratory. While not disputing this, Mr. Duncan felt that not enough had been done to explore the possibilities of using operating ships' engines for trials and investigations on the matter. He realized that the co-operation would be difficult — one could appreciate the owners natural reluctance to be either a guinea pig, or a subsidizer of experiments which would benefit the engine builder in the first place. But this seemed to be a short sighted view; when the total costs of a series of successful failures were considered it would surely be economically sensible for all concerned to subsidize a joint study in a few ships which might well produce the much desired results. While accepting that ships were not laboratories, nevertheless they could provide useful information if enquiries were organized, and also test beds for extended on site trials. In this context he had noted Mr. Hill's remarks that trials of thin wall bearings were made in crossheads between 1962 and 1965, and that, from 1970, all leading makes of engines had used insert bearings in their crossheads. Mr. Duncan was not aware that general use of insert bearings was established as early as 1970, but he knew that some owners, in attempts to deal radically with the crosshead problem had tried thin wall bearings before that year. However, it was certainly against this background that the application of thin wall bearings to crossheads had taken place, and he would be most interested to hear Mr. Hill's further views on the points mentioned regarding investigation, and testing on-site.

Finally, while he had been deeply involved as a superintendent engineer in crosshead bearing problems over the period of the survey referred to, Mr. Duncan had, in recent years, been rather less close to the matter, and so was not fully informed about present conditions. It seemed to him, however, that it might be useful, in the next year or two, to initiate a similar survey covering the current generation of engines from say 1971 onwards, to see how well they were performing in this respect, and to determine how far the introduction of thin wall bearings had gone towards solving the problems. Mr. Hill's comments on this suggestion would be welcome.

The CHAIRMAN Mr. M. V. Elliston, B.Sc., F.I.Mar.E., commented that Mr. Mack had raised the point that medium speed engines were far less tolerant of water contamination than slow speed engines. Recently, his company had experienced severe corrosion in lead bronze main and bottom end bearings in certain medium speed engines. The only reason given for these failures had been water contamination of the lubricating oil. Analysis of oil samples had shown that invariably the water content had been kept well below 0·5 per cent and the oil had been satisfactory in all other respects.

He would be grateful if Mr. Hill could give some idea of the maximum water contamination levels acceptable to ensure that corrosion problems were avoided with this type of bearing.

MR. E. HOWEY, F.I.Mar.E., asked if the author would please comment on the condition of the bearing shown in Fig. 33 of his paper and kindly give his opinion as to whether or not this bearing might give any further useful service.

DR. J. M. CONWAY-JONES said that during the discussion the Chairman had asked how low the water content in the oil had to be to avoid corrosion of the bearings, and he would ask whether this referred to the corrosion of whitemetal or lead bronze bearings?

The Chairman then replied that he was referring to lead bronze lined engine bearings, and Dr. Conway-Jones continued by saying that it was well known that the tin oxide corrosion of tin based whitemetals was often associated with a water content of around one per cent in the oil. In diesel engines water in the oil was only one of several factors which were considered to be possible causes of corrosion of the lead phase in lead bronze bearings, and hence it was not possible to state a lower limit. The factors were:

i) presence of organic peroxides caused by oil degradation;
ii) presence of organic acids, again due to oil degradation;

iii) presence of sulphur compounds in the oil, possibly caused by blow-by exhaust gases past the pistons or leakage of fuel into the lubricating oil;

iv) presence of water, particularly if it contained chemical additives such as antifreeze products.

Once initiated, corrosion damage did tend to accelerate fairly rapidly due to the entrapment of stagnant degraded oil.

Correspondence

MR. G. CROMBIE, wrote that it was rather surprising that in his section "The Bearing" the author had made no reference to the Camella restricted clearance bearing. This might be because he was not aware that this bearing was still available following the closure of its designer's and manufacturer's works. However, the writer's company now held the exclusive world rights for the sales, design and manufacture of these bearings.

If an attempt to fit this bearing into Mr. Hill's Fig. 5 were to be made we might allot the following ratings, 4:4:4:4:5, making it as good as, if not better than, any of the other designs shown. Regarding the remarks column it might be stated that the bearing

1) could be made in any size (bearings up to 625 mm diameter were in service);
2) could accept loads in any direction since the bores were all truly circular;
3) was very simple and cheap to manufacture since it merely required the machining of circular bores;
4) could accept rotation in either direction.

For those not familiar with this type of bearing, it was shown in Fig. 39 and might best be described as consisting of

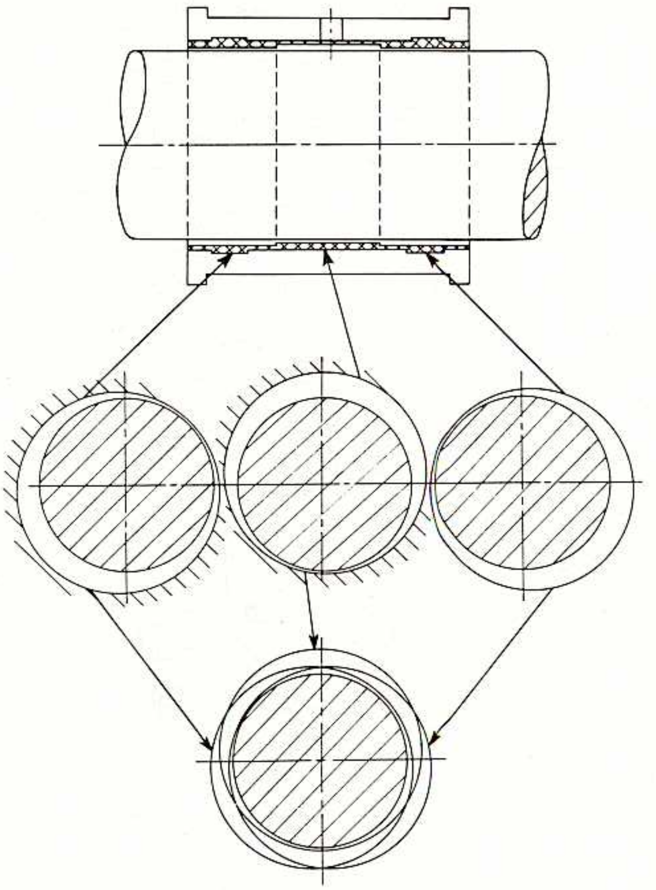


FIG. 39

a series of completely plain, round bores each placed eccentrically to its adjacent partners. There could be any number of sections and they need not be of equal lengths. The number of sections and their lengths would depend on the degree of radial restraint required and on the directions of the forces acting on the journal. It could be observed from Fig. 39 that the bearing gave a very restricted clearance for shaft restraint but, at the same time, offered a very large oil clearance. The large major (i.e. oil) clearance led to lower frictional resistance thereby giving lower bearing losses than conventional bearings. Around 20 per cent reduction was possible on turbine and gearbox bearings.

The lower frictional loss combined with the large oil clearance led to much reduced operating temperatures; for example, reductions of 17 to 28°C had been obtained after replacing conventional bearings with these restricted clearance bearings. On a test bearing at 10 000 rev/min, 3·45 N/mm^2 pressure and 0·076 mm clearance an oil drain temperature of 72°C was recorded, compared with 88°C for a conventional bearing with a clearance between 0·30 and 0·38 mm at the same rev/min and bearing pressure.

The ability to vary the disposition of the segments was particularly helpful when shaft operating speeds were near to the critical speed. If the inboard bearing segments were made the principal load carrying members, then the shortest possible unsupported shaft length was obtained thus giving the corresponding highest possible critical speed without otherwise changing the rotor in any way. This could be used to advantage in high speed applications such as turbochargers.

The bearing offered inherent anti-whirl properties, since the more restricted the clearance was the higher were the reactive forces which tended to centre the shaft in the bearing. It was also extremely effective for reducing noise and vibration.

It should not be imagined that this bearing was restricted to small or high speed applications, however. A number had been supplied for stern tubes and for split stern bearing applications in sizes up to 625 mm diameter. The greater length to diameter ratio of stern bearings meant that a large number of segments could be incorporated thus enabling restraint and support to be supplied where most needed. It was obvious that the restricted clearance and vibration damping qualities could be invaluable in stern bearing applications. Its major advantage for stern bearings, however, was that no oil ways were required. On conventional bearings these were of massive dimensions and were positioned on the horizontal centre line port and starboard. The writer had measured the attitude of tailshafts within stern bearings and had found that the shaft frequently ran into the area covered by the oil ways. Oil film breakdown with its associated effects was, therefore, not impossible. The completely plain circular bores of the Camella bearing eliminated this problem altogether.

A useful side effect was the fact that the segments could be arranged so that the bearing acted as a pump. A self-contained lubricating oil system, whereby the bearing was gravity fed to one end, and itself pumped the oil back up to the header tank from the other end, could thus be used. This property was also extremely useful for vertical shaft applications.

With regard to the low speed operation of stern bearings the following facts might be of interest. Several 533 mm diameter Camella bearings were fitted on land based machinery for rolling plastic floor tiles and similar materials. Loads of 101·6 tonnes giving bearing pressures of around 14 N/mm^2 at 10 to 25 rev/min were common. This type of load and speed could be likened to that of a stern bearing operating under slow speed conditions. The restricted clearance bearings for this duty had given a life of between four and eight times that of conventional bearings.

The value of the Camella bearing being able to accommodate loads in any direction could be appreciated by studying Mr. Hill's Figs. 12 and 13 and the relevant text. Using this bearing there was no problem of instability, and expensive profile bore or tilting pad bearings were unnecessary. There was also no need to worry about the position of oil grooves since only a small oil hole was required.

MR. J. F. BUTLER, M.A., F.I.Mar.E. thought that the author was to be congratulated on condensing such a vast amount of information on bearings into a paper of acceptable length. So complete was the coverage it appeared that the only bearings not dealt with, and rightly so, were those with oscillating motion using elastomers to absorb the relative motion and piston rings.

In Fig. 10 the minimum film thickness for a steadily loaded bearing of 500 mm diameter was given as 20 microns. It was known the bearings in diesel engines with cyclical loading would run satisfactorily with half that figure. Piston rings had quite a long life with a minimum film thickness of only 0·5 microns. Did the author think that there was any possibility of devising a formula correlating minimum film thickness with the proportion of cycle time during which it occurred.

It was agreed that top end bearings of slow speed two cycle engines had always been difficult. The use of precision thin shell bearings in conjunction with oil pressure enhanced sufficiently shortly before maximum load occurred to lift the pin from the bearing had greatly improved reliability. A further device used by the writer's company was an oil feed arrangement in which oil supplied through the crosshead pin had to flow along axial grooves in the bearing to feed the bottom end. Apart from ensuring an ample supply at the grooves this presumably provided some local cooling. A combination of these factors had increased top end bearing life by an order, and permitted the pin surface roughness to be relaxed from 0·075 microns to 0·15 microns. This relaxation made the nitrided pin surface much less sensitive to corrosion.

The author's statement that because of minimal misalignment and deflexion the load intensity acceptable on the test rig bearing was two to three times that in most practical applications was most important and borne out by Fig. 19 and Fig. 20.

In many machines and particularly in diesel engines, angular shaft deflexions under load produced changes in journal positions at the ends of the bearing which were large compared with minimum oil film thickness. It followed that even if the original machining could be perfect, it was essential to design sufficient compliance into the mating parts to keep the specific loading at the two ends of the bearing within, say, plus or minus 30 per cent of the mean.

Older lightly loaded engines with long slender connecting rods achieved this condition automatically for the bottom end bearings but in modern stiff engines special provision must be made. It was the writer's opinion that this factor became even more important as bearing materials with increased strength came into use and he would be very grateful for the author's view on the subject.

There was so much valuable information in the paper that it was likely to be used as a reference guide for many years. Could it perhaps be reprinted in book form with hard covers to withstand continual use?

MR. J. H. MILTON., F.I.Mar.E., thanked Mr. Hill for his very comprehensive and informative paper which he was sure would be retained by many engineers for future reference purposes.

To all marine engineers — especially those who have "felt round" a steam reciprocating job prior to taking over the 12 to 4, bearings would always be a topic of interest whether they were thin shell 40 per cent tin-aluminium, or white metalled "sphericals."

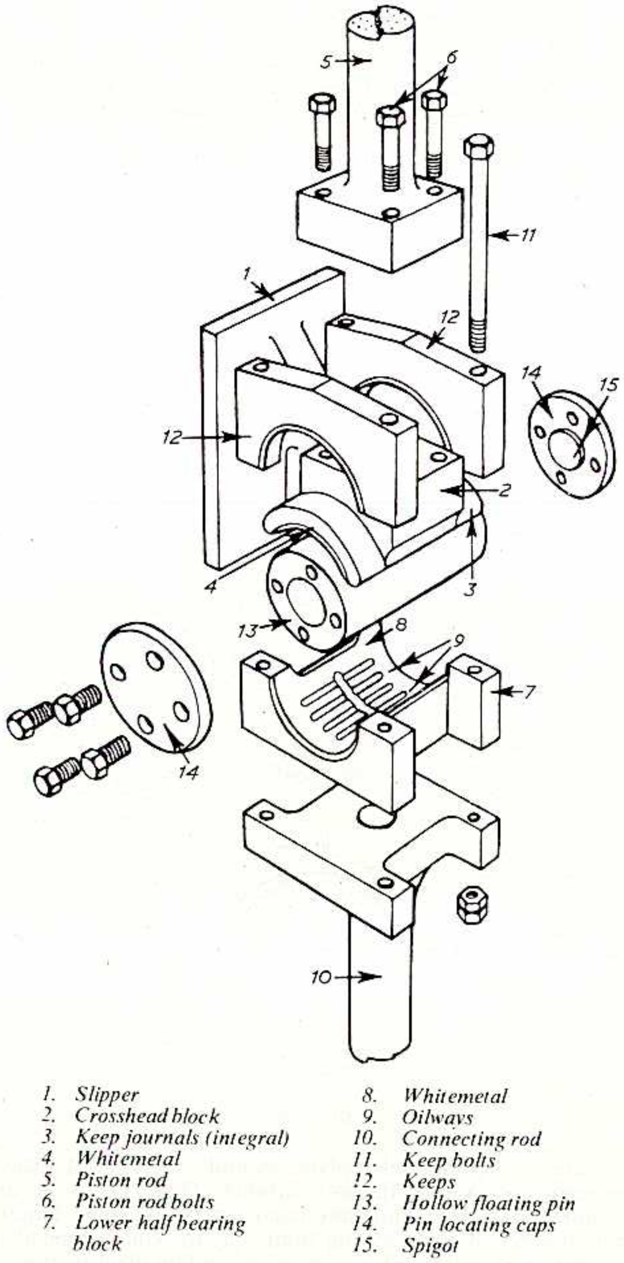
The fourth paragraph of the Introduction to the paper stated "a section on bearing damage diagnosis has been included, since this is not only of interest to the operator, but is of great importance to the designer of the machinery. It is unfortunately still true that much more is to be learnt from failures than from their absence". From this statement one could assume that bearing makers were inclined to blame machinery designers for their bearing failures.

In the writer's present position with Lloyd's Register, he was privileged to see machinery defect reports coming in from all over the world, and, apart from stating that troubles with crosshead bearings were not so prominent as they were say ten years ago, maybe due to the larger percentage of 4 stroke medium speed engines now being installed, he would much appreciate the author's comments on the following:

1) Thin shell bearings were normally prevented from turning in their housings by a snub on one side, and nip of the housings — although some engine makers appeared to rely entirely on the nip. If such bearings were opened up for survey and then reassembled using correct bolt torque, was it safe to assume that the required nip would still be present?

2) Cases had occurred of thin shell crankpin bearings being wrongly assembled, top for bottom, thus blanking off piston cooling oil and causing a crankcase explosion. Likewise main bearing shells had revolved in their pockets with serious consequences. Was it not the bearing makers responsibility to ensure that his bearings were so designed that the top and bottom halves, if different, were not interchangeable, and in addition always had means other than the actual nip to prevent them turning in their housings?

3) One engine builder recommended that when two thirds of the white-metal overlay was torn off his thin shell copper lead bearings it was time for their renewal — presumably on account of their low seizure resistance had the author any comment on this and would he agree that in these days of non-scraping it was not advisable to renew one such bearing



1.	*Slipper*	*8.*	*Whitemetal*
2.	*Crosshead block*	*9.*	*Oilways*
3.	*Keep journals (integral)*	*10.*	*Connecting rod*
4.	*Whitemetal*	*11.*	*Keep bolts*
5.	*Piston rod*	*12.*	*Keeps*
6.	*Piston rod bolts*	*13.*	*Hollow floating pin*
7.	*Lower half bearing block*	*14.*	*Pin locating caps*
		15.	*Spigot*

FIG. 40—*Assembly of proposed "rigid" crossheads with fully-floating pin*

on a crankshaft without the others?
4) Cases had occurred of spare bearings which had never been in service developing blisters on their bearing surface, which, it was understood, had been caused by hydrogen liberated from the steel backing. Would the author state his experience of this and make comment.
5) In the case of crosshead bearings for slow speed diesels, Mr. Milton asked if the author could inform him which engines had adopted tin-aluminium and would he enlarge on the statement made in the paper that "for difficult applications such as the crosshead bearings in slow speed diesel engines it had been found advantageous to overlay plate even whitemetal" — presumably this was electro-deposition of lead-tin on to the whitemetal bearing surface?
6) Finally, as the author was aware, at one time the writer had advocated a floating pin crosshead for slow speed two stroke diesels, as illustrated in Fig. 40. If it could be demonstrated that in service such a pin would rotate, due to the different bearing pressures on power and compression strokes, would he agree that this might go a long way towards overcoming the lubrication problem which caused failure of these bearings?

Author's Reply

In reply, Mr. Hill expressed his thanks to all who had contributed to the discussion, giving the benefit of their views and experience. He felt that the value of these discussions was often as useful to the profession as the author's papers.

Unfortunately it was only too true that bearings did suffer damage, and more frequently than anyone would wish. Sometimes the fault lay in the bearing design or manufacture, but more often the cause was extraneous, and the bearing damage only the visible result — bearings could be maltreated by dirt in the system, by incorrect assembly or perhaps by misalignment or flexure in the machine, to name only a few possibilities. In some cases the cause could be diagnosed and cured; in others only the palliative of shorter periods between inspections and servicing was possible. It was in the interests of all concerned to discuss these problems openly so that avoiding action in one form or another could be initiated.

Typically, the discussion showed that tin-oxide corrosion was still very much a problem, but one which had come to be accepted as inevitable and therefore it no longer received much publicity.

Several contributors had asked about advice and instructions for the assembly of thin walled bearings. The author had to confess that he was not aware of this felt need among operators, and therefore had not included any notes in his paper. The subject was rather long to include in this reply but consideration would be given to the production of a suitable publication in the near future.

Mr. Mack had raised some very important issues and the author was not sure that sufficient was known at the present time to answer him in full.

Many of the problems were directly related to the pursuit of ever increasing output from diminishing size, with the result that we were beginning to approach natural barriers. The law of diminishing returns operated here, as elsewhere, so that greater efforts were needed to make smaller gains and inevitably the margin of safety was steadily eroded.

The fortified lubricating oils mentioned were a case in point, emphasized by the relatively new phenomenon of Bacterial growth and which would no doubt result in a new additive whose side effects were still to be realized.

Filtration too, became more important when thinner oil films and harder materials were used. To obtain filtration of desirable fineness with full flow element filters would require them to be of quite uneconomic size. Therefore in practice, people tended to make do with a coarser full flow filter, backing it up by a fine by-pass filter, hoping that all dangerous — i.e. too large — particles would be trapped ultimately. To provide improved by-pass filtration the author's company had introduced the small centrifuge, driven by the oil itself, which was being increasingly fitted to diesel engines by the makers.

Mr. Mack's remarks on monitoring bearing performance were very useful. Unfortunately there was, as yet, no reliable method of providing an alarm that the bearing condition was deteriorating to danger point. All the methods he mentioned were helpful, and if the author stated that they were all too slow in action to enable machinery to be stopped in all circumstances before a bearing failed, that was not to say that it was useless to take those precautions.

Experience had shown that the most reliable monitor was a thermocouple with its junction in metallic contact at, or just below, the bearing surface. But if it was situated even 3mm below the surface it was possible for a bearing to wipe before the thermocouple registered any increase in temperature. Fortunately, in most cases, the temperature increased more slowly, giving the thermocouple time to react, although even then the total time to failure could be all too short.

Mr. Galloway had spoken of the importance of the feed back of information from users to manufacturers, and the author was entirely in agreement. Often a damaged bearing was simply replaced by a new one and the incident forgotten. Generally only serious or recurring damage was reported back to the maker.

Certain causes of damage could only be diagnosed in the early stages; later on the damage progressed to the point where the evidence was lost. It was possible that a report of the first incidence of damage, with the return of the bearings for examination, might enable the maker to diagnose the cause correctly and so prevent a subsequent, and perhaps much more serious, occurrence. Failures could raise sensitive issues but it was surely of benefit to all concerned to hold frank, if confidential, discussions.

Mr. Beckett had instanced a problem with gearbox bearings and queried whether a higher viscosity oil during running in would be beneficial.

The answer to this depended very much upon the application. A higher viscosity oil would tend to provide a slightly greater minimum film thickness, thereby reducing the risk of scoring by foreign matter in the film. But the increase in film thickness would only be marginal since film thickness did not vary directly with viscosity, whilst the increase in operating viscosity would be reduced by the higher temperature in the bearing caused by the higher viscosity feed. In practice, it was unlikely that film thickness could be increased sufficiently by this means to avoid damage by foreign particles, whilst the higher running temperatures would increase the risk of seizure following scoring.

The danger from scoring was that the raised metal at each side of the furrow caused local heating which could initiate seizure. The safer practice was to run-in at a lower speed to minimize heating effects.

As for foreign particles, the only safeguard known to the author was to avoid them, by the exercise of rigorous cleanliness, particularly during final assembly. He admitted that this was easier said than done.

The author was grateful to Mr. Duncan for his remarks on crosshead bearing problems and for showing the results of the survey of failures in the period 1960-71. The author could recall many of the problems experienced in the early 60's with direct lined bearings.

The bearings were invariably scraped to obtain a good "bed". For an oscillating application a properly scraped surface should give good results but the author had seen bearings where the scraping was so badly done that the contacting surface was less than 10 per cent of the designed surface. It was not surprising that fatigue cracks developed. The lead-in tapers at the grooves were usually too large and reduced the load bearing surface still further. The presence of keying grooves provided stress raisers which further reduced the load capacity of the whitemetal.

In the early 60's the author's company made strenuous efforts to persuade the engine builders to fit insert bearings in order to obtain a marginal gain in whitemetal lining quality and to open the way for further improvements in materials, for example tin-aluminium. Samples were made and trials carried out with all leading makes of engine but at that time most of the builders were reluctant to make such a radical change, partly no doubt because it would increase the first cost.

However, the builders had not been idle and several improvements were made. Keying grooves were omitted and generally higher standards of machining, fitting and alignment were intoduced. Some had changed the crosshead design to the 'inverted T' type used earlier by Gotaverken, whilst the others had adopted some form of conjugate deflexion.

The conjugate deflexion principle was illustrated in Fig. 41. The inevitable deflexions in a traditional crosshead were

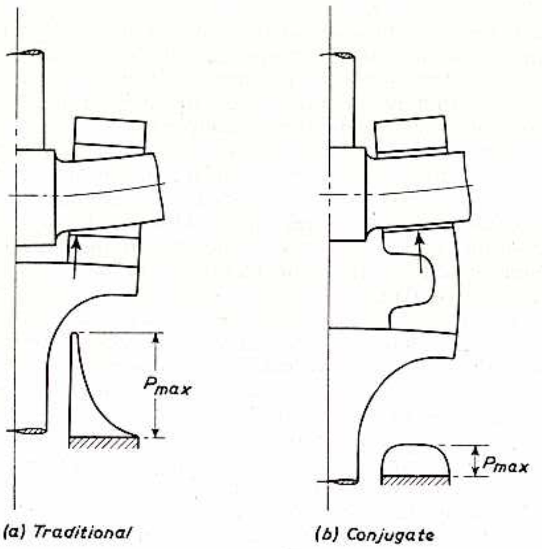


FIG. 41—*Deflexions in typical crossheads (exaggerated for clarity)*

as shown at (a), and resulted in severe specific loading at the inner ends of the bearings and consequently extremely thin oil films. A design using conjugate deflexion was as shown at (b) where the natural deflexion of pin and bearing caused them to remain in line, resulting in a lower, and more uniform, specific load.

One maker at least had adopted much larger pin diameters which gave the advantages of less deflexion, lower specific load and higher sliding velocity.

However, it was probable that these improvements had only kept pace with the uprating of the individual engines and there had been little gain in safety margins.

With the acceptance of insert bearings between 1965 and 1970, according to maker, it had not only become possible to provide a more consistent lining where whitemetal was still used but the way had been opened for stronger materials such as the 40 per cent tin-aluminium.

The application of an electrolytic overlay of lead-tin had been found to be beneficial. It was originally applied to direct lined bearings after they had been hand scraped. The soft overlay filled the undulations left by the scraping and enabled the high spots to "bed-in" more quickly. With precision boring this aspect was no longer necessary but the overlay was still beneficial in accommodating slight geometric inaccuracies and improving the bedding-in generally. It became more advantageous when harder lining materials than whitemetal were used.

None of these changes, however, had dealt with the fundamental problem, which was that the oil film was extremely thin, because of the oscillating motion and because there was no load reversal to encourage the entry of oil, to provide a squeeze film during the high load part of the cycle.

This had been tackled by the builders, one using a pump mounted at the crosshead to inject high pressure oil into the surfaces. Another had increased the lubricating oil pressure to the crosshead to a level where it caused separation of the surfaces during the low load part of the cycle, providing a squeeze film during the high load period. Yet another had developed a spring mechanism for the same purpose but had abandoned this in favour of twinned bearings which cooperated with a pin having bearing surfaces slightly eccentric to each other so that the load was transferred from one to the other during the cycle. It was the author's view that once thicker films were formed by using one of these methods, crosshead bearing life would be considerably extended.

The author agreed with Mr. Duncan that test and development programmes were needed. Unfortunately a rig test, to be effective, had to be carried out at a fairly large size and would be very expensive. Shipowners were understandably reluctant to have trials carried out on their engines but even when they had agreed it proved very difficult to obtain information on results. Moreover, one could not take their bearings to failure point, and this was an essential part of any development. The right way was for the interested parties to get together and provide funds for large scale rig testing to be done.

The author's company had maintained a close liaison with the engine builders and would be very willing to collaborate in the setting up and carrying out of a suitable programme of experimental work.

Mr. Duncan's suggestion of a further survey of performance was timely and perhaps could be set up in the near future by an independent body: B.S.R.A. perhaps?

Meanwhile, the author's company was continuing to work on the problem and reports on performance from owners would be welcomed.

Mr. Howey had asked whether the bearing shown in Fig. 33 had any useful life left or should it be rejected, but it was not possible to answer with a plain yes or no.

Yes, this bearing could be used for a further period and if no spares were available the author would not hesitate to put it back. But, as the overlay had been worn through over a substantial area in the crown, and penetrated by erosion near the groove, there was the likelihood of corrosive attack of the exposed copper-lead (see Fig. 29), which would lead to more severe wear and damage. So the bearing had a relatively short service life left. In the event the bearing was rejected and replaced by a spare.

Mr. M. V. Elliston's question regarding the acceptable level of water contamination had been answered by Dr. Conway-Jones who was more competent in that subject than the author.

In his communication, Mr. Milton had raised several points:

(1) Thin walled bearings were intended to be secured and prevented from turning solely by the interference fit, or "nip", in the housing. The snub or "nick" was only to assist in locating the bearing during assembly.

Normally it was safe to assume that there would be adequate nip when re-assembling a bearing into the housing to which it was previously fitted.

However, in some cases, particularly some of the more highly rated medium speed engines fitted with relatively thinner wall bearings, the steel backing was stressed very near to the yield point, and if there were signs of swelling near the joint, or of fretting, it would be a wise precaution to check that the nip was correct to the drawing when re-assembling.

(2) The author agreed heartily that bearings should be designed so that the halves could not be incorrectly assembled. Usually the nick could be so placed that it ensured correct assembly. This was clearly the responsibility of the bearing and engine designer, rather than the bearing maker, although the latter was always willing to advise.

In the large engine crossheads the half bearings were held in the housings by button stops at each side, primarily to prevent damage to machinery or personnel during handling and assembly. The stops were placed so as to prevent incorrect assembly but were not expected to prevent rotation of the bearing in the housing: that was ensured by the nip when the housings were bolted up.

(3) The engine builder mentioned might well be using the arc of overlay removed as an indication of the amount of wear.

The amount of wear permissible before a bearing was replaced, and whether the renewal of one bearing in isolation was advisable, depended very much on the engine and the operating conditions. Only the engine builder could advise. In some engines, particularly with corrosive or dirty oil, it might be advisable to replace bearings before one-third of the overlay had worn off.

(4) Hydrogen was dissolved in the steel during manufacture and subsequently diffused out, the rate depending on temperature.

It did cause blistering of the lining but this problem was usually confined to steel above about 25 mm thick. With thinner steel there was usually ample time for the hydrogen to diffuse out before the lining was applied. With thick steel the diffusion rate was slower and then it was normal practice to accelerate the diffusion by heat treatment before applying the lining.

(5) The 40 per cent tin-aluminium alloy was developed specially for the low speed engine bearings. It was first applied to production engines of Sulzer design built by I.H.I. in Japan, the first of which entered service in September 1971. Operating experience had been favourable. A full account was given in[25]. Similar developments had been on-going in Europe.

Overlay plating had been described in the reply to Mr. Duncan.

(6) Mr. Milton's proposals were known, and might offer some advantage. The critical aspect was whether the floating pin would rotate with certainty or in a haphazard and/or partial manner; if the latter then the lubrication conditions would probably be worse than in a conventional crosshead.

The contribution by Mr. Butler was very helpful. The author agreed that, under special conditions, much thinner films than shown in Fig. 10 were operating satisfactorily. Fig. 10 was for the guidance of designers and had all the demerits of being based on "average" conditions and including a fairly large factor of safety; perhaps ignorance was a better term. Moreover it was intended for use with steadily loaded bearings where sliding speeds and temperatures could be high — much higher than in diesel engines.

It was unlikely that there would be any simple correlation between allowable film thickness and the proportion of cycle time during which it occurred. It would depend on whether the motion was rotating (main and bottom end bearings) or oscillating (crosshead). It was also important whether it occurred predominantly on one zone of the bearing or shaft, producing local heating or was distributed. A minimum film position which moved relative to bearing and shaft was more acceptable.

The author agreed that cooling was a very important factor in crosshead bearings. Without some device, such as bleed grooves, to provide adequate flow the already thin film would be reduced further by high temperatures. Mr. Butler's solution was excellent since it provided a higher flow of oil, none of which was wasted as occurred with bleed grooves. It was gratifying to learn that bearing life had increased by an order, and that the surface finish of the pin could be relaxed to 0·15 micron.

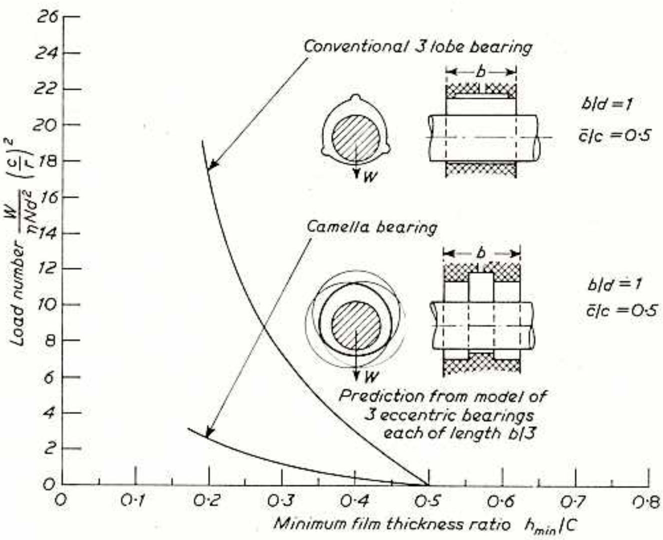
He was also in agreement regarding built in flexibility of the right kind. This certainly could become more important when harder bearing materials were used. It was also important to ensure that the load was applied at, or near, the centre of the bearing length.

Mr. Crombie had raised some interesting points in his communication. The Camella bearing was probably a special case of a 3 lobe bearing. Because it was a restricted clearance bearing it suffered the same problems common to restricted clearances although they may be mitigated by the enhanced cooling oil flow mentioned by Mr. Crombie.

Comparing it with the bearings in Fig. 5 the author would hesitate to ascribe to it the optimistic ratings allotted by Mr. Crombie and would consider 3:1:4:2:2 max. more appropriate. This type of bearing had been seriously considered from time to time but finally discarded for two reasons. The total length was usually three times, but at least twice, the length of the load carrying section. By comparison a multilobe or tilting pad bearing was shorter, and minimum length was often a designer's requirement.

It was true that the bearing span could be minimized by arranging the load carrying sections to be innermost but this was only true if the direction of load vector only varied within a narrow arc. If the direction of load vector changed by a large angle, as in reversing gearboxes for example, then the effective bearing span changed also. In high speed machines which might have dynamic load vectors the varying span would introduce a further variable into an already complex system; with what result the author could not say.

Unfortunately the test results quoted did not provide sufficient data to enable any comparisons to be made. However, Fig. 42 had been prepared to provide a comparison, in non-dimensional terms, between a Camella bearing and a normal 3-lobe bearing of the same length.



c is the hydrodynamic radial clearance (i.e. bearing bore radius — shaft radius)

$\bar{c}$ is the assembled radial clearance (i.e. the smallest clearance between bearing bore and shaft when the shaft is at the geometric centre).

FIG. 42

This indicated that, for any given minimum film thickness, the 3-lobe bearing had a higher load capacity. In practice, of course, the 3-lobe bearing would be much shorter, probably between b/2 and b/3.

Published for THE INSTITUTE OF MARINE ENGINEERS by Marine Media Management Ltd. (England Reg. No. 1100685) both of 76 Mark Lane, London EC3R 7JN. Printed by St. Stephen's Bristol Press Ltd., Bristol BS15 5JD. (England Reg. No. 757432).