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Predicting the Abrasive Wear of Ball Bearings by Lubricant Debris

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Abstract

Solid debris particles in a lubricant can become entrained into the contacts of ball bearings. The particles damage the bearing surfaces. This can lead to rolling contact fatigue failure or material loss by three body abrasion. This work concentrates on modelling the later process for brittle debris materials. A brittle particle is crushed in the inlet region and the fragments are entrained in to the contact. Rolling bearing contacts (because of the high degree of conformity) are subject to contact microslip. When this slip takes place, the trapped particle scratches the bearing surfaces. Repeated scratching by many particles results in substantial material removal. Although this failure mechanism is usually not as rapid as debris initiated fatigue, it is frequently important in mineral handling or desert environment rolling bearing applications. A simple model has been developed which considers the wear as the sum of the individual actions of each particle. The number of debris particles is determined by considering the volume of oil entrained into the bearing contacts and an empirically derived 'particle entry ratio'. The abrasive action of each particle is determined by the volume of material displaced during sliding and another empirical factor for the proportion of this removed as a wear particle. The predictions are compared with some experimental results. The correlation between bearing wear and the debris particle size is encouraging.

Introduction

Machinery lubricant systems usually contain some quantity of solid particulate debris. This debris may have been ingested from the surroundings (e.g. minerals in a conveyor system), left over from component manufacture (core sand, weld spatter, grinding debris etc.) or generated during operation (wear debris). Studies [1,2,3] have shown that lubricant borne particles can enter into rolling and sliding contacts; indeed in some cases the contact may act to 'concentrate' particulates [5].

Rolling element bearings are designed to operate in the elastohydrodynamic regime. The film of lubricating oil between the elements and the raceways is typically less than a micron in thickness. Much of the debris found in the lubricant will be larger than this and thus some form of surface damage occurs. The extent of the damage depends on the size, shape, and materials of the debris particle [4].

Large hard or tough debris particles cause severe surface damage which subsequently acts as a stress raiser and initiates rolling contact fatigue. Much attention [6,7,8] has been paid to this as it is a common cause of premature failure in rolling bearings used in contaminated environments. Filtration systems are important in removing these large particles, and longer lives are often observed with fine filtration [9,10].

However, lubrication systems also contain many small particles which are frequently impractical or even impossible to remove. These small particles will still, if larger than the lubricant film thickness, damage bearing surfaces. But in instances where this damage is not severe enough to lead to fatigue initiation a different failure mechanism occurs. Repeated damage by these small particles leads to gross material removal from the bearing surfaces. It is the process of material removal by small particles in a lubricant which is the subject of this work.

Background

Particle Deformation and Fracture

Lubricant borne particles are usually large compared with the thickness of typical lubricant films, but small compared with the size of the contacting bodies. They therefore do not carry enough load to force the elements apart but undergo size reduction before passing through the contact. Ductile particles are rolled into platelets, brittle particles are crushed into fragments [4]. The damage to the bearing surface is controlled by the size of these deformed particles. This in turn will depend on the hardness of the ductile particle or the toughness of the brittle particle.

The size of the brittle particle which 'survives' passing through the contact is related to the critical crack size in the contact stress field. A particle cannot be crushed to below this threshold. Similarly debris particles which are already below this size will pass through the contact undamaged. Thus in earlier experiments [4] 5 μm silicon carbide particles passed through the contact undamaged, whilst 60 μm glass microspheres were crushed down to fragments of the order of 1 μm is size.

In this paper we are largely concerned with the abrasion caused by the entrainment of these small size 'uncrushable' particles. In the model we consider the wear caused by diamond abrasives. These particles, although still considerable bigger than the lubricant film thickness, pass through a contact undamaged by imbedding into rolling element surfaces. This then allows us to relate material removal to particles of known size (rather than having to estimate the size of the abrading particle previously fractured or flattened).

Although diamond particle abrasion is somewhat divorced from the practical contamination problem, it still provides a useful analogue. The reasoning is as follows; large ductile particles and large tough ceramic particles will cause deep surface dents and lead to contact fatigue. Brittle particles will fracture to smaller fragments. These fragments will not be large enough to cause fatigue but lead to surface abrasion. Thus, it has been reported [9,10,11] that rolling bearings run with gear box wear debris or ceramic grinding grit failed by fatigue at approximately 10% of their rated life, whilst those run with sand debris showed no fatigue failure but a dramatic increase in bearing clearance by abrasion of the surfaces. Modelling the wear caused by 'fracture fragments' helps predict the wear which would be caused by brittle debris materials.

Material Removal by a Debris Particle

The debris particle becomes entrained into the ball/raceway contact. The load on each particle is high so it is immediately imbedded into the contacting surfaces. There is no disruption of the elastohydrodynamic film, even at very high particle concentrations [2]. It is of note that increasing the load carried by the bearing, therefore, does not increase the depth of penetration of the particle. This is in contrast to a two body abrasion process (e.g. cutting by a grit attached to a grinding wheel). Increasing the load increases the depth of penetration of the cutter and therefore the wear rate. The lubricant debris wear mechanism is essentially geometry rather than load controlled. This is an advantage, since it is not necessary to know the magnitude of the load on each abrasive particle. This presents a problem in modelling two body wear processes where it is often not known how many abraders are in contact.

Typically in a bearing the rolling elements are slightly harder than the raceways. This has little effect on the relative size of the indentations (i.e. the debris particle is pressed almost equally between the two counterfaces). However, this hardness differential has a large effect on the material removal process. The contact is subject to microslip (known as Heathcote slip) so within the contact, one surface moves with respect to the other. The trapped particle must accommodate this slip. Interestingly, it does so by remaining held in the softer surface whilst scratching the harder surface. The soft surface shows an indentation whilst the harder surface is scratched [12].

One would expect therefore, there to be proportionately more material removed from the balls than the inner and outer raceways. The experiments (to be described later in this paper) show that this is indeed the case; figure 1 shows the proportions of material removed from the ball and raceways from a plastic caged deep groove ball bearing. In reference [2] it was demonstrated that this high level of ball wear is not caused by the ball/cage contact. Close inspection of the cage showed little wear damage had taken place, and changing the cage material had little effect in the breakdown of wear between raceways and balls.

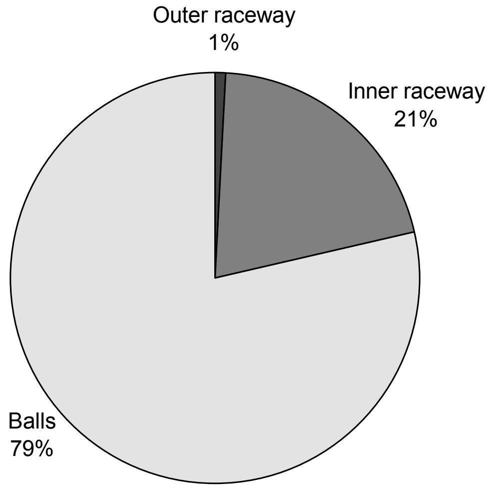

Figure 1. Proportion of mass loss from the balls, inner and outer raceways of a deep groove ball bearing run with lubricant contaminated with 2-4µm diamond abrasive particles.

This phenomenon has been noted in journal bearings [13] where in a contaminated environment the harder of the journal/bush pair shows increased wear. And likewise in a foil bearing arrangement [14] where particles were seen to score the harder surface whilst pitting the softer. Again, this is in contrast to two body wear processes where the softer surface is expected to wear faster than the harder surface.

The particle, once it has created a groove, is expelled from the contact. Close inspection of the surfaces after testing showed no evidence of any particles remaining imbedded in the surfaces. Further, subsequent running of a bearing with clean oil (but without cleaning the bearing surfaces) resulted in no further wear.

Formulation of the Bearing Wear Model

Independently Acting Particles

There are many millions of particles in a contaminated lubricant. For example 1 g/l of 5 μm debris (equivalent to about a teaspoon full in the sump of a car) is equivalent to about 10^9 particles per litre. However the volume of a lubricated contact is also very small, and one would not expect many particles to be present in a contact at any one time; this has been observed to be true in earlier work using an optical method [2]. The particles may therefore be assumed to act independently as abrasives.

Moreover, the secondary metal wear debris particles are smaller than the lubricant film thickness [12], so they do not lead to further wear. These two pieces of experimental evidence allow us to deduce that the total material removal is the sum the abrasive wear actions of each particle. To a first approximation then, provided the kinematics of the contact are unchanged, the wear rate should remain constant throughout a test. Again experimental observations on a ball/disk contact have shown this to be the case [12].

The abrasive particle is pressed into the surface and creates an impression of cross sectional area, A perpendicular to the direction of motion. If the particle then slides by a distance d; then a groove of volume Ad must be displaced. Clearly not all of this groove will be accommodated by the formation of a wear particle. Some will be absorbed by elastic or plastic deformation of the surrounding material. The proportion of the displaced volume removed as a wear particle is denoted f.

Then the total volume of material removed is given by;

$$V = \sum_{i=1}^{n} A_i d_i f_i \qquad (1)$$

The successful prediction of the mass loss from a rolling bearing relies on determining these quantities;

(i) the number of particles which take part in the abrasive process,

(ii) the cross section of the indentation each particle produces (or 'cutting area'),

(iii) the distance each particle slides, and

(iii) the proportion of a groove removed as wear.

The second and third of these may be obtained relatively easily from the geometry of the particle and the kinematics of the ball/race contact. The first and fourth are complex tribological processes (frictional particle entrainment and metal cutting). In this study they have been approached empirically.

Cross Section of a Groove

It is known that the particles become fully imbedded between the two surfaces separated by an oil film of thickness, h (as shown in figure 2). If we assume a rigid particle of known geometry we can readily determine the cutting area. In this formulation, since cubic shaped diamond abrasives, side length δ, were used as a bench mark particle, this geometry has been chosen. Then the 'cutting area', is given by:

$$A = \left(\frac{\delta}{\sqrt{2}} - \frac{h}{2} \right)^2 \tag{2}$$

This assumes that the particle is equally imbedded between the two rolling element surfaces. This is true if the surfaces are of equal hardness. In most rolling bearings, the balls are slightly (about 10%) harder than the raceways; however the two values are close enough for the respective indentations to be of the same depth. This assumption was checked by comparing diamond particle induced dents, measured by a profilometer, on the raceway and disk surfaces.

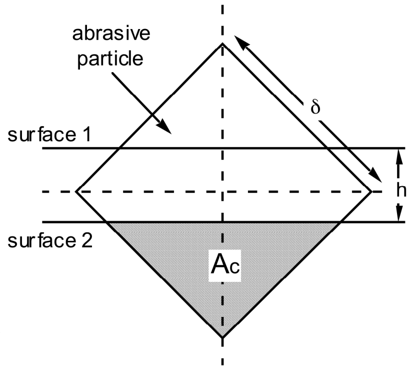

Figure 2. Diagram of an abrasive particle trapped between two surfaces separated by an oil film of thickness, *h*. The direction of motion of the microslip is into or out of the page. The cutting area is shown shaded.

Sliding Distance in a Ball/Raceway Contact

The distance a particle slides, d_i in equation (1) is obtained from the kinematics of the contact. For the case of the ball bearing contact the sliding originates from the microslip. Johnson [14] gives an expression for the amount of sliding within the contact patch, ξ; the so-called 'Heathcote' slip,

$$\xi(y) = \frac{(\gamma b)^2 - y^2}{2R^2} \tag{3}$$

where R is the radius of the ball, b is the contact width in the transverse direction, and y=γ is the position of the lines of no slip. The contact is made up of regions of stick and positive and negative slip. Reference [15] details how the variable γ (the location of the pure rolling region) may be determined. The rigorous solution requires an iterative technique. In this instance, where friction is low and degree of osculation high, it is sufficient to use a simplified approach. It is assumed that complete slip occurs at all points off the pure rolling lines (this was Heathcote's assumption) and that there is no net tangential force. This yields a value of γ=0.35.

The actual distance a particle slides will depend on the location, y at which the abrasive particle enters the contact (see figure 3). Then the sliding is the product of the slip ratio and the contact width at this location;

$$d(y) = |\xi| 2a \sqrt{1 - \frac{y^2}{b^2}} \tag{4}$$

Note we use the magnitude of the slip ratio since for the determination of the wear volume it makes no difference in which direction the particle is sliding (either the same or opposite to the direction of rolling).

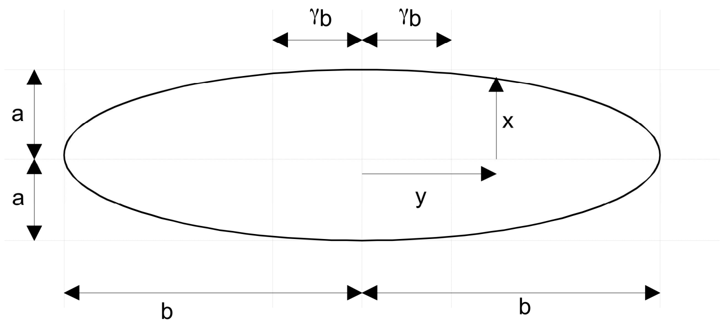

Figure 3. Schematic representation of an elliptical contact patch. A particle is entrained into the contact at position, *y* where the contact width in the direction of motion is 2*x*.

It is the last two expressions which are plotted in figure 4. Beneath these curves (and plotted on the same scale) is a profile of the wear surface from a rolling bearing which has been run with a sand contaminated lubricant. The profilometer has traced from one side of the contact patch to the other in the transverse direction; the groove radius has then been subtracted from the profile. The resulting profile then shows how much wear has taken place. Note how the regions of maximum material removal correspond to the areas of maximum particle sliding, *d*. The regions of least wear correspond to the lines of pure rolling, $y=\gamma b$.

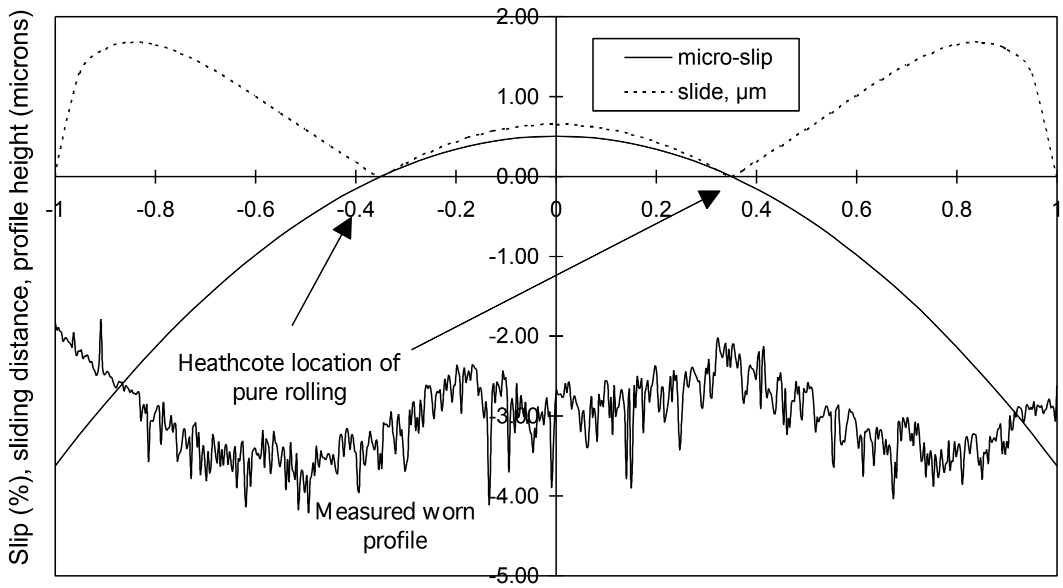

Figure 4. The variation of slip and particle sliding distance within a ball bearing contact in the transverse direction compared with a worn inner raceway profile. The profile has been recorded in the transverse direction, demonstrating that regions of greatest material removal correspond to areas of maximum sliding.

The material removal process is caused by the action of many particles entering the contact at a variety of locations *y*. The total wear volume is the sum of these actions. It is convenient to express this summation as *n* particles sliding by a mean distance;

$$d_{av} = \frac{1}{2b}\int_{-b}^{b} d(y)\,\mathrm{dy} \qquad (5)$$

It is this value, determined by numerical integration, which is used in the determination of wear volume in equation (1).

Material Removal Function

The abrasive particle displaces a groove of volume *Ad*. Not all of this material will be removed as wear. Some of the particle will be accommodated elastically, and some of the displaced material will plastically flow into the surface or to form raised shoulders.

We can obtain an estimate of this fraction of material removed *f,* from measurements of wear scratches caused by particles of known geometry. A section through an idealised scratch is shown in figure 5a, if we assume the cross sectional area of the wear particle is given by the area of the residual groove, A_1 minus the area of the raised shoulders, A_2 then;

$$f = \frac{A_1 - A_2}{A_1} \qquad (6)$$

A surface profilometer was used to measure individual scratches on bearing steel surfaces caused by particles of known size (graded diamond particles). Figure 5b shows some typical groove cross sections.

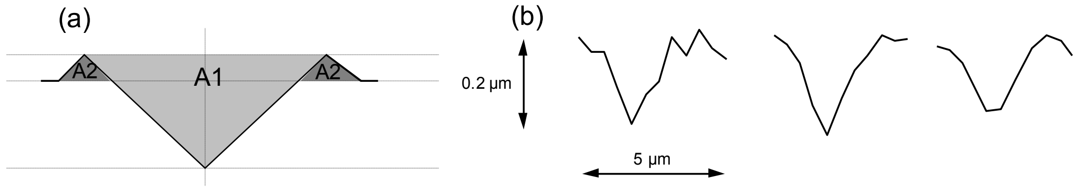

Figure 5(a). Section through an idealised abrasive scratch, (b) Profiles of typical scratches caused by diamond abrasives.

Approximations for *f* have been determined for 2-4µm and 3-6µm sizes of diamond abrasive, obtained by averaging 15 scratch profiles (again there is a high scatter with deviations ±100% of the mean). It was found that when a particle ploughs through the surface, approximately 10 to 15% of the particle's cross section is removed as wear, the rest is either accommodated elastically or redistributed plastically. For particles smaller than 2 µm the surface profilometer does not give a reliable estimate of scratch geometry since the stylus tip is too large.

It is instructive to compare these removal factors with those determined by other workers. Larsen-Basse [16] carried out a similar analysis for abrasive paper action and predicted 15% of the groove volume was removed as a chip. Buttery and Archard [17] studied how the material removal factor depended on the hardness of the abraded material and found proportions ranging from 30 to 75%. Kato et al [18] have measured the grooves cut by fixed cutting tools. The material removal is very dependent on the tip geometry and the attack angle.

The abrasive cutting process is clearly complex and the proportion of material removed is dependent on a number of factors. In this work we use the empirical data for diamond grits abrading a bearing steel surface (i.e. $f = 0.1 - 0.15$ depending on particle size). The process is strictly neither a three body abrasive wear process (since the particles are essentially held in one surface whilst they scratch the counterface) nor a two body process (since the particles are largely free to move). It would be dangerous to extrapolate this data for use in any other applications.

Number of particles entering the contact

The number of particles which take part in this material removal process must be estimated. Previous work [5] has shown that it is not safe to assume that the concentration of particles in the contact is the same as that in the bulk. Experimental data shows how the contact can have a particle concentrating effect which varies with particle size and contact speed. This is expressed in terms of a ratio ϕ, defined as the concentration of particles in the contact divided by that in the bulk.

First, it is necessary to determine the volume of oil (mixed with particles) which is rolled over by a ball as it passes through the loaded region of the bearing, in both the inner race/ball and outer race/ball contacts. Figure 6a shows the load distribution in a ball bearing and figure 6b, the resulting locus of contact patches between the ball and the inner and outer raceways (for an ideal bearing of zero clearance). The maximum ball load position is at $\psi = 0$ and the load on the ball falls to zero at $\psi = \pm\pi/2$. Notice for the outer race there is also a region where a film is generated by the centrifugal loading of the ball.

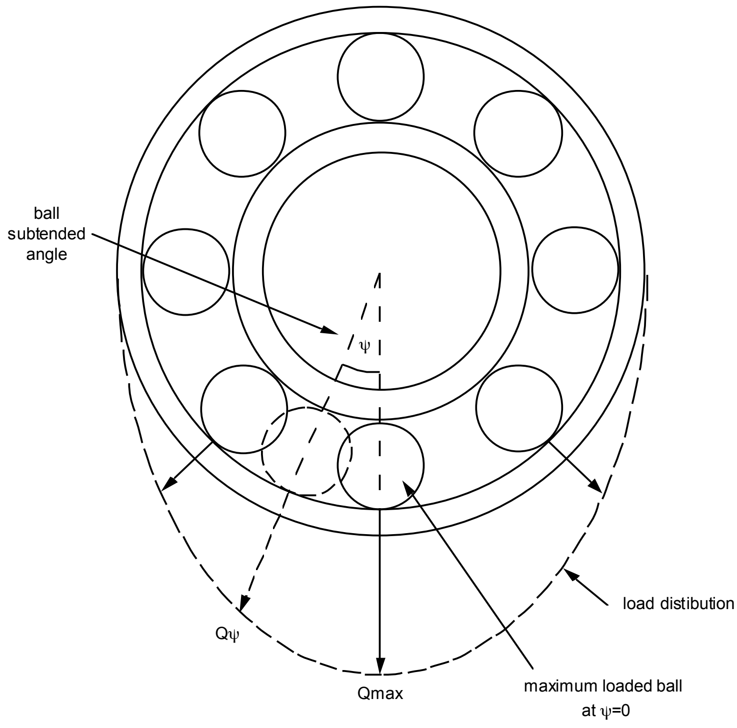

Figure 6(a). Sketch of load distribution in a zero clearance ball bearing

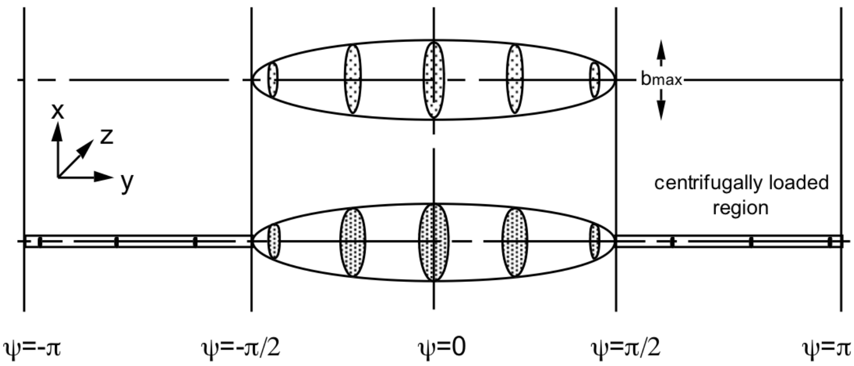

Figure 6(b). The locus of contact areas in the loaded region for the ball/inner race (top) and the ball/outer race (bottom).

The load distribution in a zero clearance ball bearing is given by [19];

$$Q_\psi = Q_{\max}\left(\cos\psi\right)^{\frac{3}{2}} \tag{7}$$

The dimensions of the ball/race contact patch can then be determined at any location from elastic Hertzian analysis. The lubricant film thickness (also a function of ψ) at any location can be determined from Dowson and Higginson's equation [20]. The volume of oil swept through the contact is an integration of the product of the local contact area and the film thickness over the bearing circumference. Full details of this numerical integration procedure can be found in reference [2].

This volume of oil swept by a ball as is passes around the bearing is denoted V_s which has two components one for the ball/inner race and for the ball/outer race. Then the number of particles which will be present in this volume is;

$$\frac{V_s \phi x}{\rho \delta^3} \tag{8}$$

where ρ is the density of the debris material and x is the bulk concentration (particle mass per unit oil volume). The particles are assumed to be of cubic shape. Relation (8) will give the number of particles trapped during a single pass of a ball. The number of ball passes during N bearing revolutions, for the inner and outer raceway contacts respectively, is given by [19];

$$0.5z\left(1+\frac{2R}{d_m}\right)N \quad \& \quad 0.5z\left(1-\frac{2R}{d_m}\right)N \tag{9}$$

The particle entry ratio ϕ is approached empirically. A separate study [5] has been carried out for this purpose. The apparatus consisted of a steel ball loaded onto a flat steel disk and run in pure rolling with a lubricant mixed with diamond particles of known size. The number of particles entrained into the contact after a set number of revolutions was determined by counting the number surface dents produced. Checks are made to ensure that neither the particles breakdown nor imbed in the surfaces to create further indentations. One entrained particle then causes one indentation. The particle entry ratio is readily calculated from the number of dents, the particle concentration, and the volume of swept lubricant. Figure 7 shows how this ratio varies with particle size and contact speed.

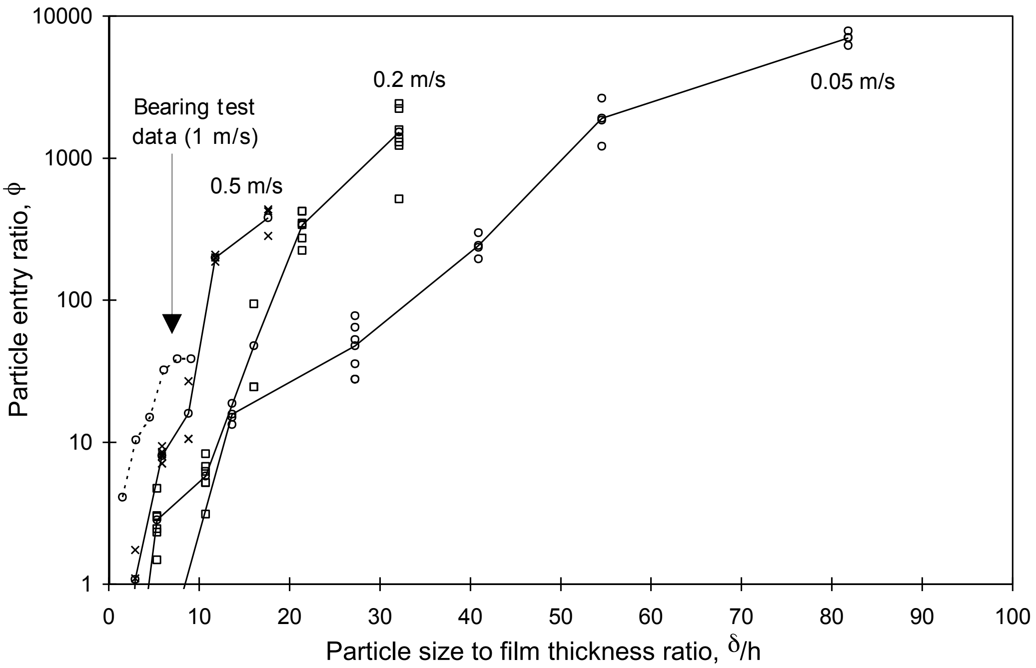

Figure 7. Plot of particle entry ratio, ϕ against the mean particle size to film thickness ratio for three contact rolling speeds. Particle concentration 0.15 g/l.

The particle entry data is at first sight surprising; larger particles are more likely to enter the contact. These larger particles when they approach a contact are trapped in the inlet region at a distance remote from the contact. At this location the fluid drag forces on the particle are relatively low. Smaller particles are subjected to much higher fluid drag forces as they progress further down the inlet region before becoming trapped by the closing surfaces. Thus smaller particles tend to get swept around the sides of the contact; whilst the larger ones are gripped by the closing surfaces and become entrained. This mechanism is described in further detail in reference (5).

It is tentatively assumed that the particle entry ratio determined from a ball on flat disk contact may be applied to the elliptical contacts in a ball bearing. Clearly the best we can expect from this approach is an estimate. Lubricant properties, contact geometry, particle shape, concentration, and possibly several other parameters may control this complex process.

The number of particles *n* to be used in equation (1) is then the product of (8) and (9).

Bearing Wear Experiments

A few bearing wear tests were carried out to help verify the modelling approach. Deep groove ball bearings (SKF 6203) were run under purely radial load in a dead weight loaded test machine. The bearings were lubricated with a mineral base oil (Shell Turbo T68) mixed with 10 g/l diamond abrasive particles (deBeers MDA synthetic diamond). The diamond particles are available ready graded into size ranges (0-0.5µm, 0-1µm, 0.5-1µm, 1-2µm, 1.5-3µm, 2-4µm, 3-6µm, 4-8µm). These particles were used because they were known not to breakdown during the test; wear can then be directly correlated with particle size. The bearings were thoroughly cleaned and weighed before and after test. Plastic cage C3 (increased clearance design) were found to be particularly useful since they could be disassembled and reassembled for easy cleaning and weighing.

The experimental data presented in figures 8 and 9 exhibits a high degree of scatter (±50%). This is most likely attributable to the difficulty in maintaining a uniform particle concentration throughout the lubricant. Particles drop out of suspension quickly and continual mixing and oil circulation is maintained during the test.

Features of The Model

The approach used here predicts that material removal will take place at a constant rate. The mechanism is the steady state entrainment of particles and their scratching the bearing surfaces. Figure 8 shows how weight loss is predicted to vary with the size of the abrasive particle. The correlation between theory and experiment is encouraging, particularly when one considers the approximations in the empirical approach.

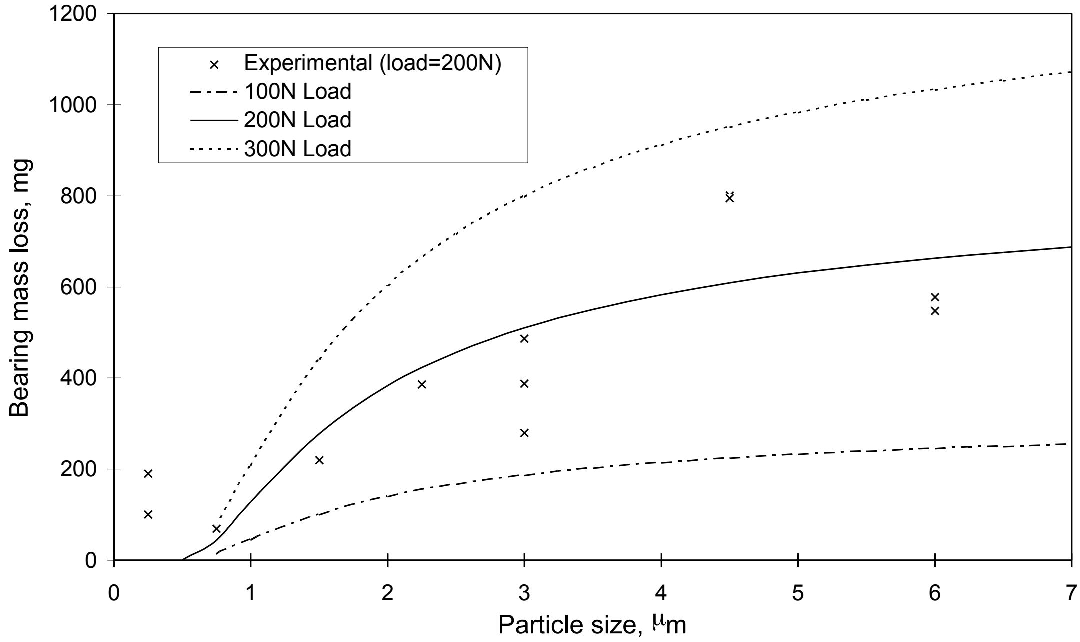

Figure 8. Bearing mass loss against particle size. Results of the model are compared with some experimental data. The test case is a 6203 bearing run with 10 g/l diamond particles of various size. Load 200N shaft speed 1425 rpm.

Remember, here we are concerned only with small size particles (i.e. the fragments of larger particles which have been crushed in the contact inlet region). For this test case the lubricant film thickness (on the most heavily loaded ball) was calculated to be 0.61 μm. The model predicts that particles smaller than this film thickness cause no wear. However, some wear was observed for the 0-0.5μm particle range. This is probably due to either the larger particles in the mix, or wear occurring at locations of lubricant starvation or thin film within the bearing.

Of importance here is the effect of the particle entry ratio ϕ. Large particles will remove more material than small ones. But a given mass contains many more particles if they are only a cubic micron in size. As the particle size, δ is increased, the number of particles decreases with δ^3 whilst the cutting area increases with δ^2. It might be expected therefore that the huge number of small particles present outweighs the fact that they are potentially less damaging. However, the entry study [5] has shown that these smaller particles are less likely to get into a contact (essentially they are carried by the fluid stream around the sides of the contact). The end result demonstrates that wear does increase with particle size, as borne out by the experimental evidence.

Another way of looking at this data is to consider what particle entry ratio would be required to give a wear model which agrees with experimental data. For the test data this ratio is shown as the dotted curve on figure 7. The curve is consistent in both form and magnitude with the data derived from the entry study.

The effect of changing the load on the bearing is shown in figure 9. Increasing the load does not cause further penetration of the abrasives. However, it does cause an increase in the size of the contacts and the number of particles which can become entrained. Thus wear is predicted to increase with load. Interestingly, the predicted increase is steeper than that observed in the experimental data. The reasons for this are not fully clear.

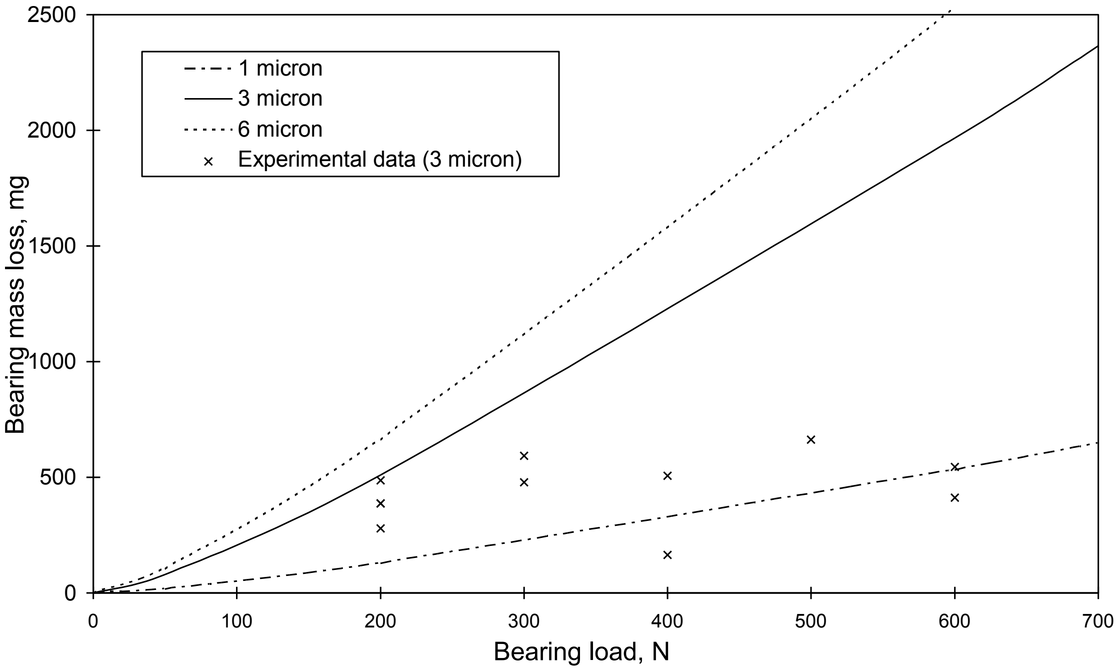

Figure 9. Bearing mass loss against bearing load. Results of the model are compared with some experimental data. The test case is a 6203 bearing run with 10 g/l 2-4μm diamond particles of various size. Shaft speed 1425 rpm.

Discussion - Limitations of the Model

The model presented here is useful for predicting wear of ball bearings by brittle debris particles. The fracture fragments behave essentially as rigid bodies and plough material from the bearing surfaces. The model relies heavily on empirical data for predicting how many particles will enter the contact and the abrasive action of each one. The particle entry process is the hardest to estimate. Here some previous test data has been tentatively extrapolated to obtain an approximation. This is probably the weakest point in the modelling process. The effect of inlet and contact shape, particle concentration, and lubricant properties on particle entry have not been investigated.

By modelling the action of small size brittle fragment particles, problems of the deformation of the particle in the bearing contacts are avoided. This approach will not work for ductile debris particles; the wear caused by a flattened platelet entering the contact has not been studied yet. It is interesting to speculate whether the relative sliding between the bearing surfaces would be accommodated by the particle shearing or by one or both of the surfaces being ploughed. Clearly the flattened particle in the contact will be much larger than a brittle debris fragment but it may not have the same ploughing capacity.

The effect of the bearing cage has also not been considered, experimental evidence suggests that the cage contacts do not play a large role in ball wear with these small particles. Changing from a riveted steel to a moulded plastic cage had little effect. It was also not possible to find any particles imbedded in the cage materials. For a fuller understanding more experimental investigation is needed in this area.

An further important process which has not been considered here is the settling of the debris particles in the lubricant supply. All modelling (and testing) has assumed a concentration of particles which does not vary with time. In practice the particles fall out of suspension quickly. In a real bearing system, estimating the number of particle delivered to any given contact inlet is difficult.

In principle this approach could be used for any rolling/sliding elastohydrodynamic system. Gears and other bearing geometries could be studied with the same approach. The contact kinematics can be used to obtain the particle sliding and swept lubricant volume. The particle entry process is likely to be dependent on the inlet region geometry (line contact, edge effects etc.) and therefore some alternative testing would have to be done for these cases.

Conclusion

1. A model of the abrasion of ball bearings by brittle lubricant debris materials has been developed. Brittle particles fracture in the inlet region and the fragments pass into the contact. The particles imbed between the two bearing surfaces. When relative sliding takes place the harder surfaces is scratched.

2. The kinematics of the contact can be used to determine the distance a particle slides in the contact. Profiles of worn bearing surfaces substantiate this approach.

3. The contact has been shown to concentrate debris particles. This has important implications for the prediction of bearing wear. The number of particles entrained into the contact is predicted by determining the volume of lubricant rolled over during bearing operation and multiplying by the particle concentration and an empirical particle entry ratio.

4. Because the particles are relatively dispersed in the lubricant they may be considered as acting independently. The resulting wear debris is too small to cause further material removal when it becomes entrained into the contact. Thus the wear process is essentially a steady state one.

5. Bearing wear increases with particle size. This is not because the larger particle removes more material but because it is more likely to be entrained into a contact.

6. Experimental data obtained using diamond abrasives agrees reasonably well with predictions.

References

[1] Wan, G. T. Y. and Spikes, H. A., (1987), "The Behaviour of Suspended Solid Particles in Rolling and Sliding Elastohydrodynamic Contacts", Transactions of the ASLE, Vol. 31, No.1, pp. 12-21.

[2] Dwyer-Joyce, R. S., (1993), "The Effects of Lubricant Contamination on Rolling Bearing Performance", University of London PhD Thesis.

[3] Sliney, H. E., (1978), "Dynamics of Solid Lubrication as Observed by Optical Microscopy", Transactions of the ASLE, Vol. 21, pp. 109-117.
[4] Dwyer-Joyce, R. S., Hamer, J. C., Sayles, R. S. and Ioannides, E., (1990), "Surface Damage Effects Caused by Debris in Rolling Bearing Lubricants with a Particular Emphasis on Friable Debris Materials", Rolling element bearings - towards the 21st Century, pp. 1-8, MEP Ltd, London.
[5] Dwyer-Joyce, R. S. and Heymer, J., (1996), "The Entrainment of Solid Particles into Rolling Elastohydrodynamic Contacts", Proceedings of 22nd Leeds-Lyon Symposium on Tribology.
[6] Beghini, E., Dwyer-Joyce, R. S., Ioannides, E., Jacobson, B. O., Lubrecht, A. A. and Tripp, J. H., (1992), "Elastic/Plastic Contact and Endurance Life Prediction", Journal of Physics D: Applied Physics, Vol. 25, pp. 379-383.
[7] Tallian, T. E., (1976), "Prediction of Rolling Contact Fatigue Life in Contaminated Lubricant: Part I - Mathematical Model", Transactions of the ASME, Journal of Lubrication Technology, Vol. 98, No.2, pp. 251-257.
[8] Webster, M. N., Ioannides, E. and Sayles, R. S., (1985), "The Effects of Topographical Defects on the Contact Stress and Fatigue Life in Rolling Element Bearings", Proceedings of 12th Leeds-Lyon Symposium on Tribology, pp. 121-131.
[9] Sayles, R. S. and Macpherson, P. B., (1982), "The Influence of Wear Debris on Rolling Contact Fatigue", ASTM STP 771, American Society Test & Materials, pp. 255-275.
[10] Loewenthal, S. H., Moyer, D. W. and Needelman, W. M., (1982), "Effects of Ultra-Clean and Centrifugal Filtration on Rolling Element Bearing Life", Trans. ASME, J. Lub. Tech., Vol. 104, pp. 283-292.
[11] Lorosch, H. K., (1983), "Research on Longer Life for Rolling Element Bearings", Transactions of the ASLE, Vol. 41, No.1, pp. 37-43.
[12] Dwyer-Joyce, R. S. , Sayles, R. S., and Ioannides, E (1994), "An Investigation into the Mechanisms of Closed Three Body Abrasive Wear", Wear, Vol. 175, pp 133-142.
[13] Ronen, A. and Malkin, S., (1981), "Wear Mechanisms of Statically Loaded Hydrodynamic Bearings by Contaminant Particles", Wear, Vol. 68, pp. 371-389.
[14] Williams, J. A. and Hyncica, A. M., (1992), "Mechanisms of Abrasive Wear in Lubricated Contacts", Wear, Elsevier, Amsterdam, Vol. 152, No.1, pp. 57-74.
[15] Johnson, K. L., (1985), "Contact Mechanics, 1st ed.", Cambridge University Press.
[16] Larsen-Basse, J., (1968), "Influence of Grit Size on the Groove Formation During Sliding Abrasion", Wear, Elsevier, Vol. 11.
[17] Buttery, T. C. and Archard, J. F., (1971), "Grinding and Abrasive Wear", Proceedings of the Institution of Mechanical Engineers, Vol. 185, No.43, pp. 537-551.
[18] Kato, K., Hokkirigawa, K., Kayaba, T. and Endo, Y., (1986), "Three Dimensional Shape Effect on Abrasive Wear", Transactions of the ASME, Journal of Tribology, Vol. 108, pp. 346-351.
[19] Harris, T. A., (1984), "Rolling Bearing Analysis, 2nd ed.", John Wiley & Sons Interscience, New York.
[20] Hamrock, B. J. and Dowson, D., (1981), "Ball Bearing Lubrication, the Elastohydrodynamics of Elliptical Contacts, 1st ed.", John Wiley & Sons, London.