Abrasive wear of rolling bearings by lubricant borne particles

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Abstract: Damage caused by lubricant borne particles in rolling/sliding contacts can severely reduce the operational life of machine elements such as cam mechanisms, roller bearings, gears, and pumps. Lubricant supplies frequently contain such contaminating particles, either generated from within the machinery itself or entrained from the surroundings. The particle can be entrained into a lubricated contact and damage the bearing surfaces. Many such individual abrasive actions can lead to significant change in the surface profile of the rolling elements.

In this work, a series of experiments has been carried out to investigate the mechanism of this surface damage and abrasion process when the contaminating particles are small and hard. The tests show, how particles are entrained into the contacts, the form of the scratches they produce, and the resulting surface profile changes. On the basis of these observations, a model of the abrasive wear process has been developed. The prediction of abrasive wear compares qualitatively well with observed form change on the bearing surface.

Keywords: simulation, abrasive wear, roller bearings, debris particles, three-body abrasion

1 INTRODUCTION

The oil in lubrication systems always contains some level of particulate contamination. The oil supply may be contaminated at its source. Particles may be entrained into the oil system from the environment through filler caps or breather holes. In addition, particles may be generated from within the machinery by wear or corrosion processes. During the run-in period of rolling and sliding components, particle generation can be very high even for a clean system with clean components [1]. It is therefore important that filtration be very efficient during this period. Experimental results show that 1 h of filtration with a 3-μm filter during the run-in of a roller bearing can reduce both the wear and the number of self-generated particles by a factor of 10 [2].

However, such filtration is costly and may not be the best economical solution [3]. Rolling-element bearings are particularly susceptible to debris damage caused by lubricant borne particles. This is because, they rely on smooth surfaces and thin separating oil films to function correctly. The debris particles are usually much larger than the oil film; so, when they become entrained into the contact, they indent both surfaces. This can lead to a fatigue initiation site or excessive abrasive wear.

The experimental results of Nilsson et al. [4] show that abrasive wear caused by hard particles can significantly change the surface profile of both washers and rollers in roller bearings.

Particles can be divided into those that are harder than the contact surfaces and those that are not. Particles that are harder than the contact surfaces may cause wear, whereas those softer than the contact surfaces may cause indentation [5]. Abrasive wear is usually divided into two types: two-body and three-body abrasions. In two-body abrasion, the particle embeds into the softer surface and scratches the harder surface. When compared with two-body abrasion, three-body abrasion is much more common and more complex [6, 7]. There are two-movement patterns for the particles in three-body
abrasion, and they are rolling and sliding. When the particle rolls or tumbles through the contact, a series of indentations is found, resulting in lesser wear damage. Alternatively, the particle can indent both surfaces, rotate until it reaches an equilibrium condition, and then scratch both surfaces. Dwyer-Joyce et al. [8] showed how these two mechanisms can occur simultaneously in a contact. Fang et al. [9] demonstrated that the situation where both surfaces suffered scratching simultaneously could not be achieved experimentally.

The modelling of the abrasive wear process by hard particles is complex. An account must be taken of the entrainment of particles into the contact and how they individually and collectively remove material. Williams and Hyncica [10] developed a geometrical model of a particle trapped between two surfaces and showed how an equilibrium particle orientation may be reached. Dwyer-Joyce [5] used a similar approach to predict debris abrasion in ball bearings using a series of empirical factors to model particle entrainment and groove formation.

The present article considers two aspects of this problem. First, an experimental study has been performed using both long- and short-duration wear tests. The long duration tests are used to obtain wear data; whereas the short-duration tests are used to investigate the entrapment of particles, their motion, and how they created abrasive scratches. Secondly, a model of this wear process has been developed on the basis of these observations.

2 EXPERIMENTAL APPROACH

2.1 Test specimens

Two spherical roller thrust bearings (model SKF 29412 E) were used in the abrasive wear studies (Fig. 1). This type of bearing inherently has large sliding motion and so is very susceptible to debris damage and wear. The first bearing was tested for a long duration in order to obtain a measurable wear distribution across the contact surfaces. The second bearing was tested for a short duration. This test was performed to study the entrapment of particles, the behaviour of entrapped particles, and the formation of abrasive scratches.

To generate abrasive wear, Micron+ MDA synthetic diamonds from Element Six Limited were added to the lubricating oil. The lubricant is pumped to circulate it around the bearings. The diamonds were 6–8 μm in size and of a blocky cubic shape. These debris particles were chosen as a standard contaminant for two reasons: firstly, because they are available in a finely graded size, range, and shape and secondly, the particles remain intact as they pass through the contact [8] and are not crushed down. It is then possible to relate any wear marks to a particle of known size and shape passing through the contact.

2.2 Test procedure

Both bearings were tested in the boundary lubrication regime; the axial load was 70 kN and the rotational speed was 5.4 r/min. Before testing, the bearing was cleaned in an ultrasonic bath. In addition, the circulating oil was filtered through a filter with a capacity of $b_3 > 200$. After filtration, the filter was disconnected and 4 mg of diamond particles per litre of oil were added. The system contained 6 l of oil and the flow was 4 l/min. The oil temperature was kept at 40 °C. A drawing of the test apparatus can be found in reference [2].

The surfaces of unused washers are relatively rough; it would not be possible to distinguish abrasive scratches from the grinding marks on the surface. For this reason, the short duration test bearing was initially run for 150 h with 4 mg/l of 0–5 μm air cleaned fine test dust, [11]. This had the effect of smoothening the washer surfaces [1]. The oil was then continuously filtered for a further 50 h of running-in. The procedure was interrupted to record surface micrographs and profiles. Finally, the bearing was mounted in the test rig for an additional 25 h of running-in with continuous filtering before the actual test.

The long-duration test bearing was for eight consecutive 23.5 h load cycles. The short-duration test bearing was for a single 8 h load cycle.

2.3 Specimen metrology

Before and after the test, the shape of the contacting surfaces of both the bearings were measured at the same position with the stylus instrument, so that the wear depth could be calculated. The surface was marked with Vickers indents (outside the contact region) as a means to assist in relocation of a surface profilometer or as a datum for the measurement of wear features.
A scanning electron microscope (SEM) was used after the test to examine the contacting surfaces on the short-duration test bearing. These examinations were used to count and measure the length of the scratches. To measure the shape of individual scratches, an atomic force microscope (AFM) was used.

All measurements and calculations were oriented according to Fig. 2. The \( y \)-axis that indicates the measured and calculated positions on the components and originates from where the roller centre line crosses the bearing centre line.

3 ROLLING BEARING KINEMATICS

In the case of house washer–roller contact, both the sliding between the surfaces and the surface separation have been calculated. The method of Olofsson [12] has been used to calculate the normal load distribution, the normal approach, analysis of tangential contact, and tangential creep. The minimum oil film thickness was calculated according to the method of Dowson and Higginson [13].

In both the house washer–roller and axle washer–roller contacts, sliding occurs within the nominally rolling contact. Within the contact, there are two points of pure rolling contact, as shown in Fig. 1. These points are located at the rolling cone of the roller. On either side of the rolling points, there will be sliding in one direction, whereas, between them, there will be sliding in the opposite direction.

A schematic of the entrainment of a particle into a washer–roller contact is shown in Fig. 3. A particle of size \( d \) first becomes trapped by the rolling elements at half contact width denoted by \( a_e \). It can also be seen in that figure that the entrapment area is dependent on the deformation of the surfaces. The deformation was approximated by moving the rigid surfaces of the contacting components, a distance equal to the approach against each other.

If a particle is entrained into the contact, it will experience the relative sliding of both the washer and roller. The particle will, therefore, slide with respect to one or the other of the surfaces. The magnitude of that sliding distance will depend on the lateral position at which the particle was entrained.

The sliding distance of the contact surfaces was calculated as \( s = 2a|\xi| \), where \( a \) is the half contact width and \( \xi \) is the creep. The sliding distance of an entrapped particle, \( s_e \), was calculated as \( s_e = 2ae|\xi| \). Figure 4 shows the calculated sliding distances for the house washer–roller contact of an SKF 29412 E bearing. The sliding distance also depends on which surface the particle adheres to. If the particle sticks to the faster surface, it will spend less time in the contact and so create a smaller scratch on the counterface. The maximum difference in sliding distance between the roller and the house washer is 7 per cent, at the inner and the outer edges of the entrapment zone. For positions in between, the difference is significantly less. Therefore, for the
calculations that follow in this article, because it is not always clear to which surface the particle adheres, the sliding distance on the house washer alone is used.

The number of contact events per shaft revolution, \( n_c \), for each rolling element was determined from the number of rollers \( (n_r = 15) \) and the radii at the inner pure rolling points \( (r_a, r_r, \text{ and } r_h, \text{ for the axle washer, the roller, and the house washer, respectively}) \) (Fig. 5). These radii were obtained by measurements and are given in Table 1 together with the results of these calculations.

For the axle washer, roller, and house washer, respectively, the number of contact events are given by the expressions

\[
\begin{align*}
  n_c &= \left(1 - \frac{r_a}{r_a + r_h}\right)n_r \\
  n_c &= \frac{2r_ar_h}{r_r(r_a + r_h)} \\
  n_c &= \frac{r_a}{r_a + r_h}n_r
\end{align*}
\]

(1) (2) (3)

4 EXPERIMENTAL RESULTS

4.1 Wear distribution

Material removal in the long-duration test is shown in Fig. 6. These plots are the subtraction of the transverse surface profiles before and after testing and thus, showing a profile of the wear depth. Generally, the extent of the worn region is greater than the contact patch width, but corresponds closer to the width of the entrapment area. At the points of pure rolling, the wear depth is small, whereas, at the regions of maximum sliding, the wear is the greatest. Similar distributions are observed for all three elements.

The wear profile on the roller is anomalous. It is surprising that the wear on the outer side of the contact is approximately five times than that on the inner side (whereas, the expected sliding distance at those two locations is the same). There is a corresponding drop in wear on the house washer at the outer location. This suggests that a greater proportion of the abrasion has taken place on the roller. As the scratch length plots described later (Fig. 11) show similar length of the scratches on both the roller and house washer, this may therefore depend on the number of particles entering the contact, how much material a particle removes, or if the particles may imbed in the opposing surface.

It can be observed that the outer side of the wear profile for the house washer better correlates to the contact area than the entrapment area. Furthermore, it can also be seen that the outer side wear profile for the axle washer better correlates to the wear profile for the roller. This may indicate that particles have a tendency to embed on the house washer surface and then scratch the roller. In reference [13], it is reported that embedded particles occurred very sparsely on the axle washers and not at all on the rollers. On the house washer, embedded particles were more common, especially on the outer side of the outer pure rolling point.

Figure 7 shows the wear depth profiles from the short-duration test. The purpose of this test was to count and characterize abrasion scratches. It was

| Table 1 Data and results for calculation of number of contact events |
|----------------------|----------------|----------------|
| Component | Axle washer | Roller | House washer |
| Radius (mm) | \( r_a = 37.6 \) | \( r_r = 8.3 \) | \( r_h = 48.2 \) |
| Number of contact events, \( n_c \) (-) | 8.4 | 5.1 | 6.6 |

Fig. 6 Wear depth on the axle washer, roller, and the house washer for the long-duration test

Fig. 7 Wear depth on the axle washer, roller, and the house washer for the short-duration test
designed therefore to be a test where the surface is not obscured by further wear. The figure shows that the wear is minimal on the roller and house washer. The axle washer shows relatively high wear. Therefore for this element, it is not possible to use a scratch counting method to determine particle entrainment (as many of the scratches will have been worn away subsequently).

4.2 Scratch morphology

Figure 8 shows SEM images of the abraded surface at three locations on a roller surface from the long-duration test. The regions correspond to different levels of sliding within the bearing contact. In each of the regions, the length of the scratches is largely constant. This indicates that all the particles are undergoing, approximately, the same motion. This is as expected, as the size range of the added diamond particles is narrow. The length of the abraded scratch is generally commensurate with the distance that the particle slides in the contact (as presented in Fig. 4). However, in Fig. 9(c), there are also some very long scratches. These are much longer than the calculated sliding distance. These scratches may originate from diamond particles trapped in the roller–cage contact or from repeated contacts with permanently embedded particles on either of the washers, assuming that the roller is fixed in lateral position. As permanently embedded particles were found on the house washer surface, the latter scenario is possible.

Several scratches on the roller and the house washer from the short-duration test were measured with an AFM. The chosen scratches were those that appeared to be the largest at that lateral location. Therefore, they should be caused by the larger diamond particles, and also, not to have been altered by subsequent wear. Figures 9 and 10 show the AFM images for three locations on the roller and the house washer, respectively.

The cross-sections of all the scratches are approximately V-shaped. The scratches tend to be deeper towards the end of the travel. The particles appear to be embedding deeper into the surface as they
abrade (i.e. they are not finding a new equilibrium position of lower abrasive profile). The width of scratches is approximately equal to half the size of the diamond particles. In all cases, the grooves are shallow (less than 1 μm deep). This is surprising, as they have been created by a 6–8 μm sized particle. The relationship between the measured cross-section area and the particle size is analysed further in section 5.3.

The size of the shoulders of material on either side of the scratch is small, at most 20 per cent of the groove depth. Material ‘piles up’ around the edges of an indentation if the material does not work-harden [14]. If hardening occurs, then the plastic flow spreads remote from the indent causing ‘sinking-in’. It would seem that the latter is closer to the case occurring with these scratches. There is also no piling-up at the end of the groove. The particle seems to have cut material from the surface without causing the formation of a wedge of plastically flowed material ahead of the particle prow. This cutting behaviour is characteristic of sharp square indentors [15], like the abrasive diamond particles.

4.3 Transverse distribution of scratch length

Figure 11 shows a plot of the scratch length recorded at a range of transverse positions on the roller and washers. These results were obtained by the use of the SEM images. The scratches measured were those that appeared to be unaffected by subsequent wear. Typically, this meant recording the longer scratches with a pronounced clear cross-section. The scratches were also chosen to be surrounded by scratches of similar length, hence avoiding scratches created by extreme wear particles. In all cases, the length of the scratch is related to the amount of sliding at that location. At the lines of pure rolling, the scratches are at their shortest, shorter even than the size of a single particle. Scratches occur outside of both the contact and the entrapment area on the house washer. This may be explained by larger particles than the 7 μm used for the calculation entered the house washer–roller contact. Or alternatively, the contact

![AFM images and profiles of wear scratches at three different positions on the house washer in the short-duration test](image)

**Fig. 10** AFM images and profiles of wear scratches at three different positions on the house washer in the short-duration test. The distance between horizontal lines is 50 nm for the image profile at position 66 and 68 mm. For the image profile at position 74 mm is the distance 200 nm.

![Length of scratches on the axle washer, roller, and the house washer in the short-duration test](image)

**Fig. 11** Length of scratches on the axle washer, roller, and the house washer in the short-duration test.
and sliding calculation deviates from the real roller–house washer contact.

Scratches occur on all the three bearing elements. It is, therefore, not simply the case that the particles stick on one surface and scratch another (in three body wear situations, the particle usually sticks to the softer surface and scratches the harder).

4.4 Transverse distribution of number of scratches

Figure 12 shows plots of the number of scratches recorded at various transverse locations on the roller and the house washer in the short-duration test. The aim of this measurement was to determine whether particular locations on the bearing favour the entrainment of particles.

The data is only used as a guide to the entrainment mechanism as it is possible that some scratches are worn out or obscured by subsequent damage. For this reason, the counting was not carried out on the axle washer, as this had worn to a level deeper than that of the measured depth of an individual scratch (Fig. 6). The house washer also shows some wear in the short-duration test. The most reliable results were obtained for the roller.

There is not a marked variation in entrainment over the contact width. It is important to note that it is not just the contact area that becomes abraded but that surface damage occurs also in the entrainment area (Fig. 3). It is interesting that the number of dents on the house washer seems to be slightly lower on the lines of pure rolling. This is unusual because one might expect a rolling contact to be more likely to entrap particles [16, 17] than a rolling and sliding contact.

5 ANALYSIS OF EXPERIMENTAL RESULTS

5.1 Particle entrapment

Both the particle concentration and the number of contacts experienced by each point on the rolling element surfaces are known. On the basis of this and the experimental results presented in section 4.4, it is possible to calculate an entrapment height, which describes an oil film thickness containing the number of particles that are drawn into the contact. This is described in terms of an entrapment ratio, \( r_e \). Thus, all the particles contained in a layer of oil of thickness \( r_e d \) are drawn into the contact (shown schematically in Fig. 13).

The number of scratches per area unit, \( I_A \), is then given by

\[
I_A = c_v r_e d N_c
\]  
(4)

Where \( c_v \) is the number of particles per unit volume and \( N_c \) is the total number of contact events. The mass of an individual diamond particle was determined by assuming that the particle is cubic of diagonal length, \( d \). Then

\[
c_v = \frac{c_m}{\rho_d (1/3)^{3/2} d^3}
\]  
(5)

where \( \rho_d \) is the density of diamond, and \( c_m \) is the mass concentration of particles. During one revolution of the shaft, all the surfaces experience \( n_c \) numbers of contact events. The total number of contact events during the test, \( N_c \), is then the product of the \( n_c \), the rotational speed, \( \omega \), and the test duration, \( t \).

From this, the entrapment ratio, \( r_e \), can be calculated as

\[
r_e = \left( \frac{1}{3} \right)^{3/2} \frac{I_A \rho_d d^2}{c_m n_c \omega t}
\]  
(6)

By using equation (6) and calculating the mean number of indents per area unit for the roller and the house washer, on the basis of results given in Fig. 12, it is possible to calculate a mean entrapment ratio for each component. For the roller and house washer, the data and results for such a calculation are given in Table 2.

The entrapment ratio is of the order of unity. This means that the assumption, that all particles in an approaching oil layer of thickness \( d \) are entrained into the contact, holds. This is encouraging, because these measurements are difficult to take and can only be considered as approximate.
However, the entrapment ratio should be the same for both the components, as a particle entering a contact should create an indent on both the components. This calculation assumes that the entrapment conditions are the same for the axle washer–roller contact, as well as for the house washer–roller contact. As the house washer has been worn more than the roller (Fig. 7), it is more likely that the result for the roller agrees better with the reality than the result for the house washer.

The number particles per unit volume in the circulating oil is $17 \times 10^9$ m$^{-3}$. The contact area between the roller and the house washer is $4.3 \times 10^{-6}$ m$^2$ and the entrapment area is $33.7 \times 10^{-6}$ m$^2$, calculated according to section 3. Using the calculation method described in this section, the number of entrapped particles at any given instant is determined. The results are given in Table 3 for entrapment ratios, $r_e$, of 1.0 and 1.8, respectively.

If we assume that the entire bearing load is carried by these trapped particles, then the normal contact pressure would be at least 10 000 GPa. This is well in excess of the hardness of the bearing steel and so indicates that the particles would be fully pressed down into the metal. At these particle concentrations, the load is, therefore, almost all that is carried by the oil film and it is not possible for the particles to cause a change in the oil film thickness.

### 5.2 Sliding distance

When a particle is trapped in a contact, the relative sliding of the surfaces can be accommodated by one of the following means: the upper surface is grooved, the lower surface is grooved, the particle rolls through the contact, the particle fractures, and the particle shears. In this experiment, there are scratches on all contacting surfaces, indicating that grooving takes place on both the roller and the house/axle washer. Clear signs of particles rolling through the contact have not been found, but there is evidence of a few fractured particles.

Figure 14 shows the sum of the lengths of the scratches on the roller and the house washer compared with the theoretical calculation of the sliding distance for that contact.

There is reasonable agreement, although the scratches are longer (especially in the central region) and have occurred over a greater transverse width than predicted. This is probably because the rollers do not always run centrally on the washers. There is some axial play, which means that the lines of pure rolling are not always in the same location. Some of the measured scratches on the roller will have been caused by the contact with the axle washer, but the sliding distribution at this contact is of a similar order and so similar scratch lengths would be expected.

It is also instructive to examine the lengths of the scratches on the two elements. At the contact sides, the scratch lengths on the roller and the washer are approximately the same. In the centre, the roller scratches are longer. The roller is slightly harder than the house washer. It has been observed that the particle usually sticks to the softer surface and scratches the harder surface [10, 18], and this is also true when there is only very small hardness difference [8].

It is unlikely that sliding would occur simultaneously on both surfaces (the limiting shear stress on one surface would always be lower than on the other). Possibly, the particle slides for some distance on one surface, and then, finds a new orientation or imbeds deeper into the material, which then means it is more favourable to stick in that way.
position and scratch the opposing surface. This would imply that the groove starts off shallow and then gets deeper (until it reaches a point when it becomes energetically more favourable to slide on the other surface). Examination of the AFM groove profiles (as shown in Fig. 10) show that this does appear to be the case.

5.3 Material removal

Ten individual scratches on each of the roller and the house washer from the short-duration test were analysed. The width and depth were determined from AFM measurements. Table 4 shows the mean and standard deviation scratch depth, width, and calculated cross-section area.

The cross-section area of a scratch is estimated as \( \frac{1}{2} \) the width times the depth. These scratches are created by a particle pressed equally into each of the surfaces which are separated by an oil film of thickness, \( h \). This means then as a first approximation the transverse area by which the particle intrudes into each of the surfaces is given by \( A_c = \frac{(d^2 - dh)}{2} \). In Table 4, the ratio of the scratch area to this particle intrusion area is also shown.

The scratch area is small when compared with the intrusive area of a particle. It appears that most of the area disturbed by the particle is either accommodated elastically or by the plastic redistribution of material. The proportion removed, as wear is relatively small.

5.4 Abrasive wear model

An empirical approach can be taken to estimate the abrasive wear using the measured data for scratch length, scratch area, and particle entrainment. The expected wear can then be compared with the measured wear.

The empirical wear depth for any slice on the actual component in the long-duration test \( h_L(y) \) was calculated according to

\[
    h_L(y) = n_S(y)l_S(y)A_S(y)S
\]

The term \( n_S(y) \) the number of scratches per unit area, \( l_S(y) \) the length of the scratches, and \( A_S(y) \) the approximate cross-section area of a scratch for the short-duration test. \( S \) is a scaling factor equal to the duration of the long test divided by that of the short test. The data for \( n_S(y) \) is given in Fig. 12 and for \( l_S(y) \) is given in Fig. 11. The cross-section area of a scratch was taken as the mean data given in Table 4. The results of the calculation for the roller and the house washer are shown in Fig. 15.

6 DISCUSSION

The three lobed profiles shown on the washer surfaces are characteristic of rolling element bearing abrasive wear as well as mild wear when the rolling element rolls in a groove, as in references [1, 5, 12]. The main difference between the mild and the abrasive wear is that, in the mild wear case, the form change is not measurable on the harder rollers, but significant abrasive wear due to two-body abrasion can be noticed on the roller in Fig. 6. For both the mild wear and the abrasive wear, the wear depth is small at the points of pure rolling, whereas at the region of maximum sliding, the wear is the greatest. This indicates that the sliding distance in the contact is a major factor causing two-body wear.
abrasive wear and this is also in good agreement with the observations of the length of the scratches presented in Fig. 11.

At the inner and outer sides of the pure rolling points, the sum of the scratch length on the house washer and the roller correlates reasonably well with a calculated sliding distance (Fig. 14). However, at positions of maximum sliding, the scratch lengths are greater than the sliding distance. Scratches are also generated at positions outside the entrapment area. These scratches are the result of larger particles in the abrasive sample or of some deviation in the contact model from real conditions.

Between the pure rolling points, the sum of the scratch length on the house washer and the roller are longer than the calculated sliding distance. One contributing reason could be that different particles stick to both the roller and the washer and scratch the opposite surface. It is these longer scratches that have been measured. When adding the length of the scratches on both the surfaces, the sum is then too high. Usually, we expect the particles to stick on to the softer surface and scratch the harder. This is not the case here as scratches occur on both the surfaces.

As can be seen in Table 4, the proportion of the removed material is very small when compared with the particle intrusion area. Also when studying the AFM images in Figs 9 and 10, it appears that the scratches created on the surfaces are shallow compared with the size of the diamond abrasive particles. Some part of the indentation by the particle has been elastically and/or plastically absorbed and redistributed.

The counting of the number of scratches on a surface can only be used as an indication of the entrainment of particles into the contact. It is a difficult measurement to take and it is also possible that scratches are obscured or worn away as testing continues. The roller provides the best test specimen for this measurement, as it suffered very little wear in the short-duration test. Comparing the particle count to the concentration of particles in the oil, and the number of revolutions indicates that the contact concentrates particle, as is also the case with point contacts [16]. Any particle in an incident oil layer, of thickness of the order of the particle size, is trapped by the converging rolling elements. All these particles are then entrained into the contact while the surrounding oil is expelled.

The empirical calculation of roller wear between the pure rolling points agrees reasonably well with measured wear (comparing Figs 6 and 15). Inside the inner pure rolling point, the deviation between calculated and measured wear is quite small. The large deviation between measured and calculated roller wear outside of the outer pure roller point may be because of abrasion caused by particles embedded in the washer surfaces.

The empirical calculation wear of the house washer deviates distinctly from measured wear. This may be due to large amount of wear generated by wear debris, which is not taken into account when performing the empirical calculation. As the surfaces of the washers are softer than the surfaces of the rollers, it is more likely that generated wear debris causes wear on the washer surfaces than on the surfaces of the rollers.

7 CONCLUSION

A series of experiments has been carried out to study the wear of a spherical roller thrust bearing by lubricant borne hard particles. The form of the worn profile and the length of the wear scratches correspond closely to the sliding within the contact. Measurements of the geometry of individual scratches show that only a very small proportion of the material disturbed by an abrasive particle is removed as wear. A count of the number of wear scratches on the rolling element surface indicates that the contact concentrates particles. This occurs because, as soon as a particle gets trapped by the rotating elements, the friction forces pull it into the contact. Therefore, all the particles in an incident layer of oil, of similar thickness to the particle itself, are drawn into the contact. The empirical data for scratch length, cross-section area, and particle entrainment have been assembled to predict the abrasive weight loss. Comparison of the simulation results with the experimental results shows a qualitative agreement for the form change of the washer surfaces.

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