TRIBOLOGY IN SLOW ROLLING BEARINGS

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ABSTRACT

Wear has been found in the Hertz tracks of fluid lubricated, high quality angular contact ball bearings which spin too slowly to generate an elastohydrodynamic lubricating film. Often a taper section or profile of the track shows a "W" or a "U" shaped groove. A mechanism is proposed to account for these shapes.

This wear has been detected by destructive long term life tests in expensive bearings. A bench test is described which gives accelerated results on easily rejuvenated plates. Connections between the ball on plate tester and any angular contact bearing are outlined.

Ball on plate tests using two perfluorinated and one synthetic hydrocarbon lubricant are interpreted in terms of a slow buildup of an electrically insulating, chemically derived film on the Hertz track. Details of the film buildup and its wear protection ability are specific to each lubricant.

INTRODUCTION

Occasionally there is a low speed ball bearing application which requires zero wear over an extended design life. If speeds are too low to support an elastohydrodynamic lubricating film, boundary lubricated wear at high pressure and very low sliding speed may occur. Long term life tests in expensive bearings are necessary to demonstrate reliability; sometimes the tests show that wear grooves have been formed. This paper considers boundary lubricated wear of ball bearings in three related parts: a description of the wear grooves and a model to explain their shape, some background information on an accelerated bench test with simple rolling geometry designed to mimic conditions in a real bearing, and some experimental results obtained with a synthetic hydrocarbon and two perfluorinated fluids.

Track profiles

Figure 1 shows a "W" shaped groove having a central peak height of about 30 microinches. It is a taper section across the inner race of an angular contact ball bearing, made after several thousand hours of low speed running. Figure 2 shows a profilometer trace across a "U" shaped groove about 40 microinches deep, from the outer race of a second similar bearing. Both types of groove have been found on both types of race; Figures 1 and 2 are presented as examples. The lubricant in both cases was a perfluorinated polyalkylether, chosen because good boundary lubricity was expected.

The finish in the contact tracks indicates that boundary lubrication was almost good enough. There was no galling, no prow formation, and no large wear particle formation. To account for the groove shapes, suppose the wear rate at each point in a Hertz contact to be proportional to the product of the local pressure and the local sliding velocity, i.e. to local friction energy dissipation. This assumption underlies most of the wear formulations found in the literature [1]. Total wear would be the integral of this product at a given location across the track over the time required for a complete contact to roll past it.

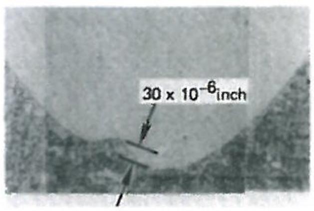


Figure 1. "W" taper section.

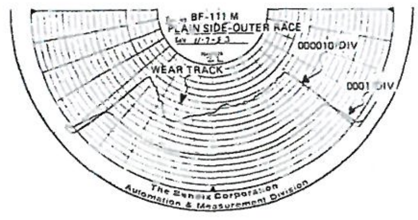


Figure 2. "U" profile.

Local pressure in a Hertzian contact is ellipsoidally distributed. Local sliding velocity has two sources: kinematic slip due to pivoting (the component of relative angular velocity between ball and race normal to the contact, often loosely referred to as spin); and rolling drag slip due to drag torques on the ball set. Kinematic slip can be calculated at each contact point and time, based only on the geometry of the contact [2]. Rolling drag slip is uniform over the contact.

Local pivoting slip vectors are given by the cross product of the local radius vector from the footprint center and the pivoting vector. If only pivoting slips are present, a "W" profile should be generated with the central peak of the W on the track centerline, since the pressure-velocity product goes to zero at the center of the footprint. The peak should be below the edges of the groove since points on the track centerline spend most of each contact off the footprint center, where neither pressure nor slip is zero, therefore some wear takes place.

Local drag slip vectors are all the same length, always in or against the rolling direction. Thus, as drag slip increases from zero, the central peak in the groove lowers and moves toward that side of the track where drag slip and the circumferential component of pivoting slip add; the groove takes on a "U" shape. On this basis we expect a W track if drag is relatively small, a U track if it is large. Large drag would result from poor retainer geometry - tight pockets, land interferences, etc.

Figure 3 shows theoretical wear profiles calculated according to the energy dissipation assumption, for drag slips of 0, 1/10, 1/2 and 1 times maximum pivoting slip. Major Hertz width is four times minor. Each curve in Figure 3 is normalized with respect to its maximum value; the track centerline is in the center of the plot. The features mentioned above are present.

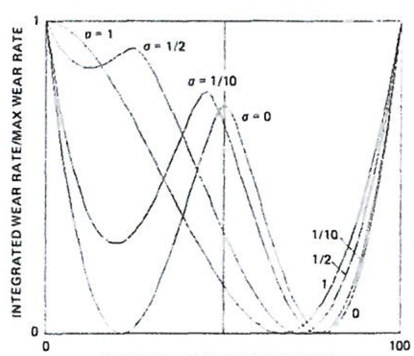


Figure 3. Pressure-slip velocity integrals.

Ball on Plate tester

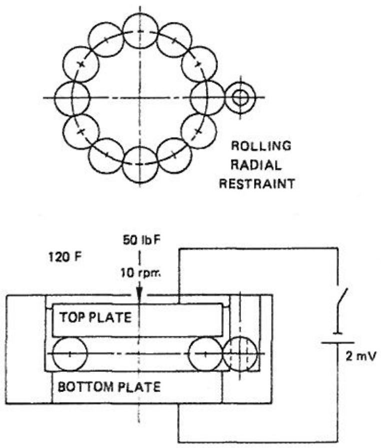
The W and U wear grooves were generated in expensive bearings over several thousand hours. An accelerated bench test whose internal mechanics can be adjusted for any bearing, but which uses a ball-between-planes geometry, has been developed to substitute for real time tests with real bearing parts. Important parameters affecting wear evaluation are entrainment velocity (used in elastohydrodynamic calculations of film thickness), Hertz pressure, individual and total Hertz area, number of contact passes in a given time, maximum pivoting slip speed, temperature, lubricant, and material. All these can be controlled in a ball-between-planes geometry. It is impossible to match an actual bearing exactly, but comparisons and tradeoffs can be made.

The angular contact ball bearing of Figures 1 and 2 must run at least 2000 hours submerged in lubricant at 120°F, without grooving. Parameters averaged for the two races of the bearing are given in the first column below. They are approximated in a group of 12 0.5 inch diameter balls rolling in a 2 inch circle between parallel plates and carrying 50 lbf. Ratios between the parameters (ball on plate/bearing) are given in the second column for a plate spin rate of 10 rpm.

Entrainment Velocity	0.023 in/s	23.61
Mean Hertz Pressure	69,000 lbf/in^2	1.46
Major Hertz Semidiameter	7.60×10^{-3} in	
Minor Hertz Semidiameter	7.62×10^{-4} in	
Individual Hertz Area	1.82×10^{-5} in^2	2.26
Total Hertz Area	6.01×10^{-4} in^2	1.64
Max Pivoting Slip Velocity	4.22×10^{-4} in/s	9.41
Ball Passes per 2000 Hours	870,000	0.83

The increased entrainment velocity is too small to give a significant elastohydrodynamic film thickness (formally calculated at 1/100 microinch), a sliding velocity of 0.004 in/sec will not give a contact temperature (estimated at 1/4 °F) much larger than one of 0.0004 in/sec, and the increased pressure and area are compensated for by the reduction in ball passes. Thus this configuration is a reasonable substitute for the actual bearing.

There are however some complications in the ball on plate test. A ball will not roll in a circle when loaded between spinning parallel planes without an inward radial force, no matter how small the orbit velocity. This is due to an asymmetrical distribution of pivoting friction lockup and microslip in the ball-plane contacts [3]. Without the radial force the ball will roll in a spiral. For these tests radial restraint is provided by a single ball mounted on a vertical pin outside the test group, to nudge each test ball in turn back into its original track. Between nudges its spiral has a pitch of order 0.0003, and varies between lubricants. No ball retainer or cage is used since balls of slightly different sizes under slightly different loads have equal orbit rates in the ball-between-parallel-planes geometry. Thus test balls touch only the two plates and the lubricant most of the time, giving the simplest possible rolling conditions at the ball-plate contacts. The positioning ball is free on its pin so that mostly rolling contact exists in the ball-ball contact as well (Figure 4). However, during the nudges there is increased sliding at the ball-plate contacts. In these tests each ball experiences its repositioning contacts less than 2% of the total test time. A drill press provides the test stand so that the bottom plate is stationary and the top plate rotates. It is driven by a spindle held in the drill chuck, and loaded by weights on the drill feed. Balls and plates are hardened 440C steel, the same material as the actual bearing. New balls are used in each test. Plates are surface ground and lapped between tests to remove any existing track. They are mounted in a heated, covered, 440C tank for temperature and lubricant control. The only materials present in the test are 440C and lubricant. Approximately 40 cc of lubricant completely fills the ball group free volume. The tank is insulated from the drill press so that electrical resistance across the ball-plate contacts can be measured. A photograph of the rig is shown in Figure 5.



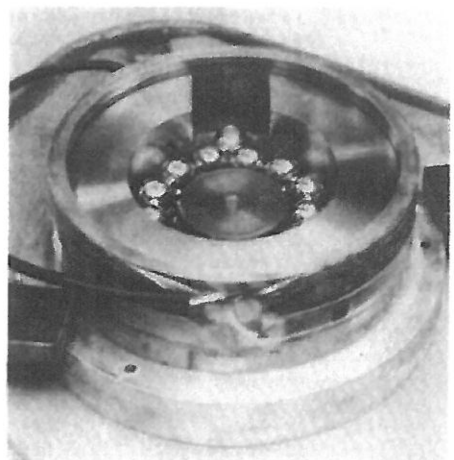


Figure 4. Ball on plate geometry. Figure 5. BOP tester (top plate removed).

DC voltage drop across the ball set and wear are the measured parameters in these tests. A 2 mv open circuit potential, chosen to minimize electrical effects on any lubricant derived film, is applied across the ball set. The voltage is present only during a measurement, about 10 seconds per reading. Typically, voltage is applied over less than 0.1% of the whole test. Top plate electrical contact is made with an alligator clip on the spindle by winding a lead around the spindle for a few revolutions, to avoid brush resistance problems. Wear profiles are taken on the plates using a standard profilometer at a vertical magnification of up to 50,000X and resolution of 1 microinch. Horizontal magnification is 100X.

Ball on Plate test results

Three fluids were tested: a polyalphaolefin synthetic hydrocarbon, 440 cSt; a perfluorinated polyalkylether, 11 cSt; and a perfluorinated tertiary amine, 1.5 cSt. For at least the first several hundred plate revolutions all three fluids showed zero contact resistance, confirming absence of any hydrodynamic lubrication under the test conditions, and eliminating ordinary viscosity as a significant parameter. But after that, each test showed an increase in the voltage drop across the ball set over time, different for each fluid.

Figures 6 and 7 plot the voltage drop against time for the perfluorinated ether and synthetic hydrocarbon fluids. The initial slopes are 0.15 and 0.67 mv/hr. Supposing the contact resistance to be caused by an insulating film of friction polymer gradually building over time, we expect better wear protection from the hydrocarbon since its film is complete after about 10 hours, compared with 40 hours for the fluoroether. Also, the latter film shows signs of penetration towards the end of its test.

Figure 8 shows three profiles taken across the track from the bottom plate of the hydrocarbon test; there is no difference in the surface finish inside and outside the track (in the center of each trace). Some preliminary

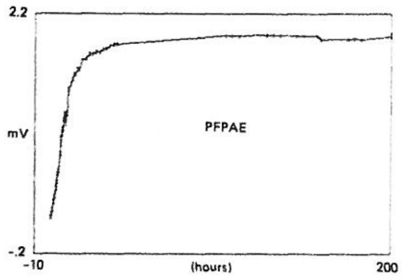


Figure 6. Voltage drop across balls vs running time.

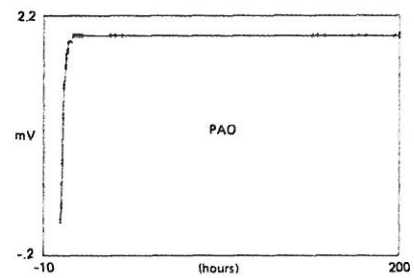


Figure 7. Voltage drop across balls vs. running time.

attempts to detect and identify films in the hydrocarbon track by Fourier transform infra red spectroscopy have been unsuccessful. Figure 9 shows profiles from the fluoroether test. There has been grooving and surface disruption, consistent with the contact resistance data and general experience with synthetic hydrocarbons and fluoroethers.

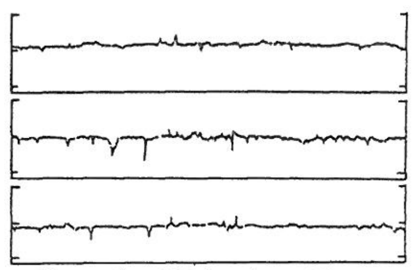


Figure 8. PAO track profiles.

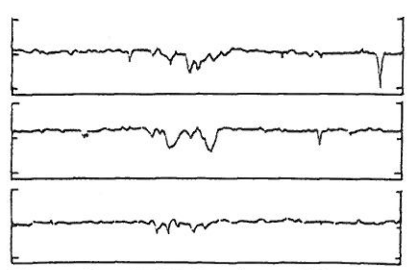


Figure 9. PFPAE track profiles.

Vertical scale - inches x 50,000
Horizontal scale - inches x 100

Figure 10 shows the voltage drop measured under fluoroamine. Initial slope is 0.80 mv/hr, suggesting quick film buildup, but after 30 hours the film is apparently disrupted. The cycle repeats 3 times over the test. Figure 11 shows the profiles for this material. Vertical magnification was reduced from 50,000X to 20,000X to show the complete grooves. Energy dispersive analysis by xray of orange colored wear particles showed oxygen, iron and chromium. The particles were identified as 440C rust. An insulating film of rust quickly built up under the fluoroamine fluid, but it was brittle and flaked off after 30 hours. Since it contained a substantial fraction of iron, considerable wear resulted.

The speculation that a protective film of friction polymer is formed from the synthetic hydrocarbon lubricant is based on results obtained at higher speeds with a thin, but complete elastohydrodynamic film of mineral oil [4]. In that case polymerizing reactions are thought to be activated by mechanical shear within the film, required by the pivoting in the contact. It is reasonable to suppose that similar reactions are activated in synthetic hydrocarbon trapped between solid contacts in the boundary lubricated case. A similar mechanical activation of rusting reactions might occur under the fluoroamine fluid. 440C does not rust without some external stimulation. Finally, it is unclear whether an unbiased eye will allow a claim for a W with a displaced peak in the profiles of Figure 11. Certainly the biased eye can see them.

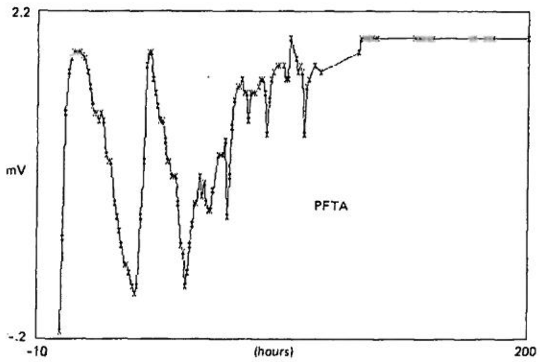


Figure 10. Voltage drop across balls vs running time.

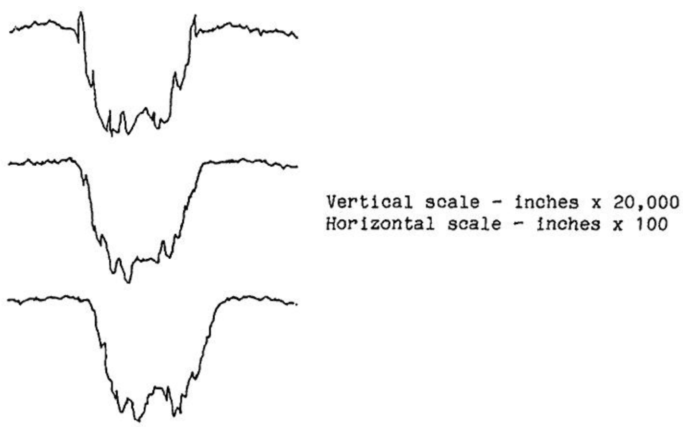


Figure 11. PFTA track profiles.

ACKNOWLEDGEMENTS

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