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THE BALL BEARING AS A RHEOLOGICAL TEST DEVICE

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SUMMARY

An angular-contact ball bearing provides an easily obtainable, precise mechanical system for rheological tests on thin fluid films under high pressure. The test conditions are by definition similar to those found in practice. Accessible independent variables include size, pressure, bulk temperature, roughness, adsorbed surfactant, fluid type, fluid quantity, fluid supply rate, film thickness, entrainment velocity, transit time, and combined strain. Easily measured or inferred variables include slip, changes in film thickness with time (transients), strain rate, lubricant elastic modulus (thin film, high pressure), tractive force, lubricant chemical degradation rate, and lubricant degradation product. Methods for setting and obtaining these quantities in a bearing are discussed, together with experimental limitations on them.

INTRODUCTION

Many machine elements, including ball bearings, depend on separation by an elastohydrodynamic lubrication (EHL) film to prevent seizure and wear. The rheologic and chemical response of the lubricant within high-pressure, high-shear EHL contacts is sufficiently complicated (refs. 1 and 2) to justify an experimental analysis. Bench tests in single-, four-, and five-ball and crowned roller test devices (ref. 3) may not reproduce the strains, pressures, volumes, and rates in real bearings. The test devices also require specialized precision parts that are easily damaged and hard to replace. As an alternative, an off-the-shelf bearing, slightly modified, can be used to investigate viscous, tractive, elastic, and chemical effects in lubricants. The rolling conditions are authentic, and if an experiment should damage the bearing, another is quickly available.

This paper presents a collection of formulas from various sources that allow quick calculation of test conditions. These include formulas for Hertz contact parameters and for bearing and contact kinematics and dynamics. Also, some of the rheologic experiments with bearings that have been made (and could be extended) are briefly described in an attempt to indicate the versatility of a ball bearing as a rheological test device.

TEST CONFIGURATION

For rheological testing the bearing is run in counterrace rotation at zero ball orbit rate under pure axial load. No ball retainer (cage) is fitted but

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a full complement of balls is provided. Lubricant is supplied in small, known amounts to the balls by withdrawal from a dilute solvent dip, the races being left dry after cleaning. The result is a more or less uniform lubricant film on the balls that is thin enough to be immobile in the operating centrifugal field. Its thickness can be established by weighing and is less than the calculated fully flooded classical thickness, thus insuring controlled starvation. Thickness can be varied by changing the solvent concentration.

BEARING GEOMETRY

Geometry is shown in figure 1. Nominal properties available from the bearing manufacturer are

β	Contact angle
C_O, C_I	Crossrace curvatures
d	Ball diameter
E	Pitch diameter
Y	Young's modulus
μ	Poisson's ratio

The maximum number of balls in a full complement is

$$N = \mathrm{int}\left[\pi/\sin^{-1}\left(\frac{d}{E}\right)\right] \tag{1}$$

HERTZ PARAMETERS

The test bearing carries an axial load L. Resulting Hertz dimensions and stress at the ball-race contacts (from the Brewe-Hamrock approximations (ref. 4) written in terms of bearing parameters) are

Effective curvatures

	Rolling direction	Crossrace direction
Inner race	$R_R = \dfrac{d(E - d)}{2E}$	$R_X = \dfrac{C_I d}{2C_I - 1}$
Outer race	$R_R = \dfrac{d(E + d)}{2E}$	$R_X = \dfrac{C_O d}{2C_O - 1}$

(2)

Curvature sum

$$\frac{1}{R} = \frac{1}{R_R} + \frac{1}{R_X} \tag{3}$$

Ellipticity (ratio of major to minor contact dimensions)

$$k = a/b \approx 1.0339(R_X/R_R)^{0.636} \tag{4}$$

Elliptic integral

$$\mathscr{E} \approx 1.0003 + \frac{0.5968}{R_X/R_R} \tag{5}$$

From these the semimajor Hertz width is

$$a_{B-R} = \left[\frac{6k^2\mathscr{E}R(1 - \mu^2)}{\pi Y}\right]^{1/3}\left[\frac{L}{N \sin B}\right]^{1/3} \tag{6}$$

and the average contact pressure is

$$P_{B-R} = \left(\frac{L}{N \sin B}\right)\frac{1}{\pi ab} \tag{7}$$

At a ball-ball contact carrying load ℓ the area is circular with radius a_{B-B}, where

$$a_{B-B} = \left[\frac{3\ell d(1 - \mu^2)}{8Y}\right]^{1/3} \tag{8}$$

and the average contact pressure is

$$P_{B-B} = \frac{P}{\pi a_{B-B}^2} \tag{9}$$

KINEMATICS

A reasonable figure of merit for a rolling contact is surface velocity; in a ball bearing this is given by

$$V = \pi\dot{\delta}d \tag{10}$$

where $\dot{\delta}$ is the ball spin angular velocity.

For any combination of race rotations (mode), including counterrace rotation, ball spin rate is (ref. 5)

$$\dot{\delta} = \rho S \tag{11}$$

where

$$\rho \text{ (basic speed ratio)} = \frac{E^2 - d^2 \cos^2 B}{2Ed} \tag{12}$$

$$S \text{ (total speed)} = \dot{\gamma}_0 - \dot{\gamma}_I \tag{13}$$

ρ is a constant bearing characteristic number, and S is the algebraic difference in race rates.

All modes (including inner, outer, or counterrace rotation) at the same total speed give the same ball spin rate; they are kinematically equivalent. This allows counterrace-rotation rheological experiments to be related to the more common operating modes on the basis of total speed and surface velocity:

$$v = \pi d \rho S \tag{14}$$

Specification of v specifies S; for counterrace rotation nominal race rates become

$$\left.\begin{aligned} &\text{Outer-race rate} && \dot{\gamma}_0 = \frac{S}{2}\left(1 - \frac{d \cos B}{2}\right) \\ &\text{Inner-race rate} && \dot{\gamma}_I = -\frac{S}{2}\left(1 + \frac{d \cos B}{2}\right) \end{aligned}\right\} \tag{15}$$

BALL-RACE SLIP

There are two kinds of slip in a bearing: kinematic slip, which is dictated by the motions and shapes of the balls and races; and an additional rheological slip introduced by the lubricant. Examples of the former are Heathcote, pivoting, and transverse precessional slip. These would be present without any lubricant. They are the main tribological reason for lubricating a ball bearing, and they can be calculated without knowing anything about the lubricant.

The most important kinematic slip is produced by pivoting between balls and races. (Pivoting is the relative angular velocity component of two bodies in contact normal to their plane of tangency. It is often loosely referred to as spin.) The sum of the inner- and outer-race pivotings is constant in a ball bearing, independent of the position or motion of the ball spin vector (ref. 6):

$$\sum \mathscr{P} = S \sin B \tag{16}$$

The associated pivoting slip u is maximum at the edge of the Hertz area

$$u = 2\pi a_{B-R} \mathscr{S} \tag{17}$$

and zero at its center. Pivoting slip has been suggested as the source of mechanically activated lubricant degradation reactions in starved bearings (see below).

Drag torques produced by orbit and spin dissipations of the ball set cause an additional rheologic slip in the lubricant. Its magnitude depends on viscosity and film thickness in the EHL contacts. Since the thickness is set and known, this slip can be used to study high-pressure viscosity in situ. Drag slip is always in the rolling direction and cannot presently be calculated. One method of measurement is to measure the ball and race rates (ref. 7) and then to infer slips (normalized with respect to v) from kinematic equations connecting running geometry and rates:

Ball - outer race

$$\xi_O = \dot{\gamma}_O/\dot{\delta} \left(\frac{E_A}{d_A} + \cos B_A\right)$$

Ball - inner race (18)

$$\xi_I = \dot{\gamma}_I/\dot{\delta} \left(\frac{E_A}{d_A} - \cos B_A\right)$$

Actual pitch and ball diameters E_A and d_A and contact angle B_A in these equations are not the manufacturer's nominal values. Instead they must be measured in an auxiliary experiment at low speed and with thin film so as to minimize drag slips. For zero drag slip they are given by

$$\left.\begin{aligned} \cos B_A &= \left(\frac{\dot{\delta}}{2\dot{\gamma}_O\dot{\gamma}_I}\right)\left(\dot{\gamma}_I + \dot{\gamma}_O\right) \\ E_A &= \left(\frac{\dot{\delta}}{2\dot{\gamma}_O\dot{\gamma}_I}\right)\left(\dot{\gamma}_I - \dot{\gamma}_O\right) d_A \end{aligned}\right\} \tag{19}$$

The actual ball diameter d_A is easily measured directly. Once established, these parameters can be used in equation (18) for running conditions when the slips are not small.

An artificial rheologic slip can be induced in the counterrace rotation mode by applying an external torque to the ball group. Race rates are set to give a not-quite-zero "free" orbit rate. A pin inserted axially between two balls and connected to a straingage dynamometer forces zero orbit rate (fig. 2). This technique allows a study of tractive effects within EHL films and also of ball-to-ball load carrying capacity (ref. 8).

BALL DYNAMICS

Bearing balls are forged from drawn wire and usually have an anisotropic structure and an axis of figure (ref. 9). An artificial axis of figure can be produced with a small diametral hole. Occasionally a bored ball is observed to precess (ref. 10) such that the hole axis traces out a cone of half-angle θ at rate p, with the ball simultaneously spinning around the axis at rate s. Figure 3, taken from (ref. 10), shows stroboscopic photographs of a precession; θ, s, and p can all be obtained from figure 3. Conditions leading to precession are not yet completely understood, but if it is present there must be a kinematic transverse precessional slip at the ball-race contacts (ref. 11) given by

$$\mathcal{S}_T = \frac{d}{2} (s \sin \theta \sin pt) \tag{20}$$

Both s and θ are measured to be small but p is found to be large ($\cong \dot{s}$). Thus this slip is of small amplitude but high frequency. It has been suggested (ref. 2) that the slip is accommodated by elastic strain within the lubricant. If so, precession provides an experimental method for studying elastic behavior in bearing lubricants.

The bored ball rapidly spinning about a fixed point (its center in counterrace rotation) is a classical gyroscope. Its motion can be used to infer the resultant torque acting on the ball through the EHL films (ref. 12)

$$M = p \sin \theta \left[I_H s + (I_H - I_E) p \cos \theta \right] \tag{21}$$

Here I_H is the moment of inertia of the ball about its axis of figure and I_E is the moment of inertia about a normal axis, both easily calculated.

CHEMICAL EFFECTS

If the EHL film is too thin, the (constant kinematic) pivoting slip induces too much shear energy in the lubricant and oxidation or polymerization reactions are activated. The fluid lubricant is changed irreversibly into a solid (ref. 13). Figure 4 shows lubricant degradation products on a race. It has been suggested (ref. 2) that the rate of these reactions is given by

$$R = (\text{const}) \exp (-Ah/u\tau T) \tag{22}$$

where A is an activation energy, u is the pivoting slip velocity, h is film thickness, τ is a limiting shear stress and T is contact transit time. As the film thins, the degradation rate increases exponentially, without any metallic film penetration being required.

It is possible to increase the rate at constant thickness by treating the metal parts with a surfactant (ref. 13). This would have the effect of decreasing A.

CONCLUDING REMARKS

These various tests have been made on bearings ranging in pitch diameters between 1 and 5 cm with basic speed ratios between 2 and 6. Races and balls of chromium steel and stainless steel have been used. Stainless steel parts can be final cleaned with concentrated H_2SO_4 to remove all organic contaminants. This step is necessary if adsorbed surfactants are to be evaluated. So far only race finishes as received from the manufacturers have been run. There is no reason why artificially roughened (etched or sandblasted) surfaces could not be used.

A variety of organic fluids have been tested. Mineral oils ranging in viscosity between 0.02 and 4 N s/m^2 have been deposited in films between 5×10^{-6} and 25×10^{-6} cm thick. A suitable inert solvent for mineral oils is hexane.

Ball-race contact pressures as high as 2×10^9 Pa have been used at surface velocities up to 500 cm/sec. Transit times of 1×10^{-5} sec are typical, with pivoting slips as high as 5 cm/sec.

It is easy to control bulk temperature for small bearings at typical service loads and speeds. Larger bearings tend to run above room temperature.

In summary, a ball bearing used as a rheological test device gives useful experimental access to the regime of EHL where physical and chemical effects in the lubricant blend together. The simple geometry and motions of the bearing parts allow easy determination of the relevant test parameters.

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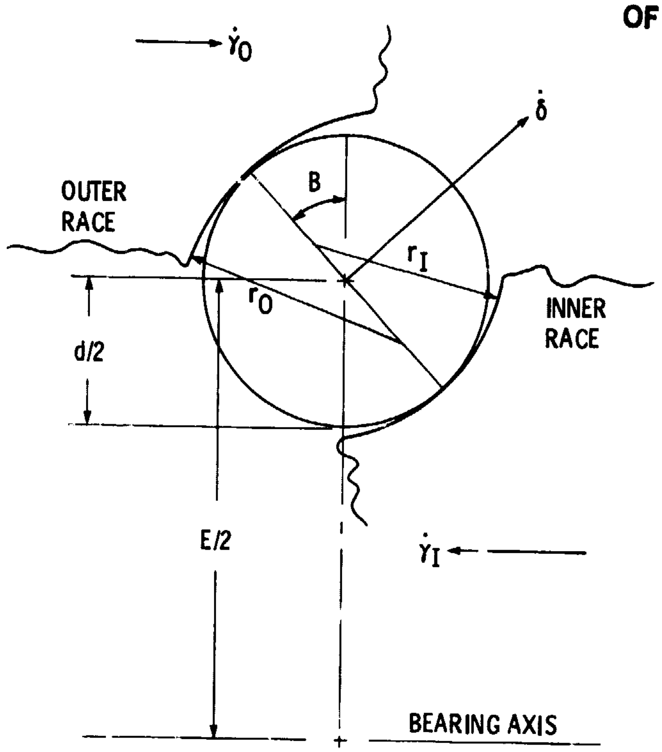

Figure 1. - Ball bearing geometry.

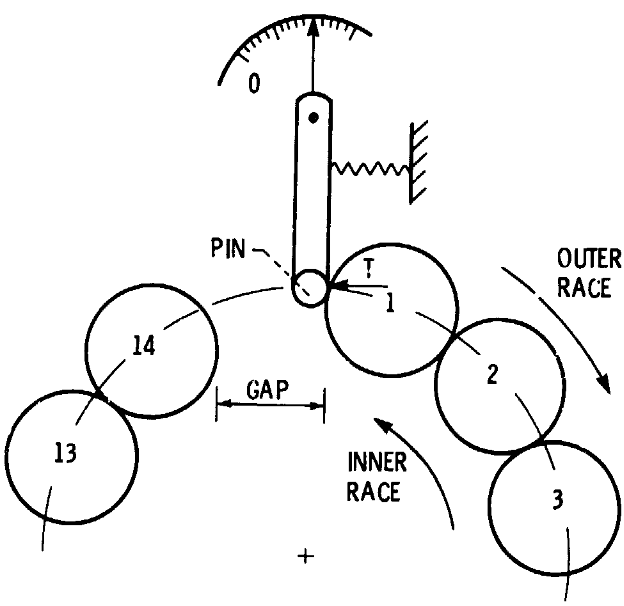

Figure 2. - Induced-slip apparatus.

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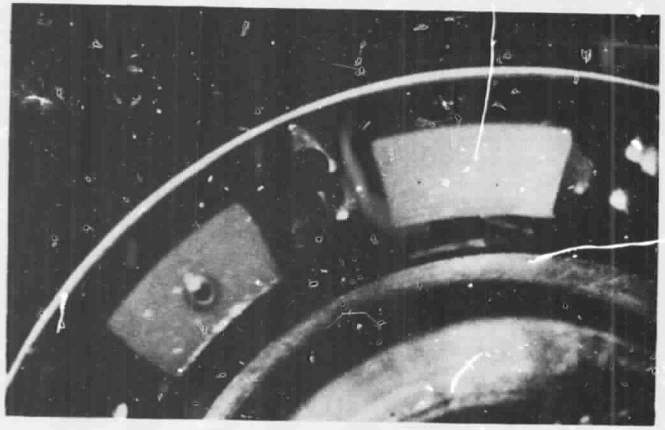
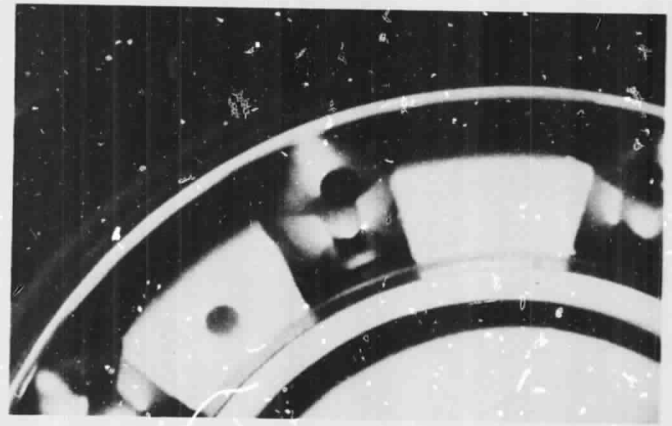
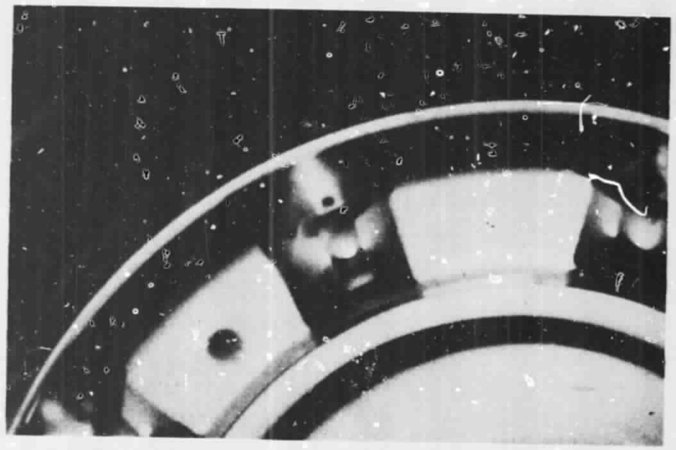


(a) Bearing stationary.



(b) Ball precessing, stroboscopically illuminated at p.



(c) Ball precessing, stroboscopically illuminated at s + p.

Figure 3. - Ball precession.

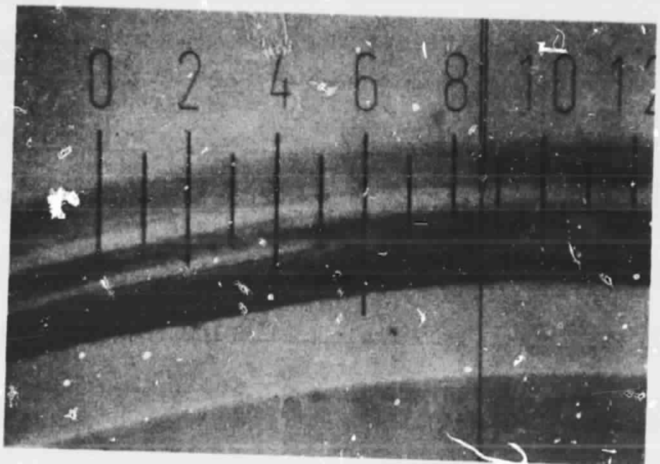


Figure 4. - Solid-lubricant degradation products.