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Long Term Performance of a Retainerless Bearing Cartridge With an Oozing Flow Lubricator for Spacecraft Applications

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LONG TERM PERFORMANCE OF A RETAINERLESS BEARING CARTRIDGE WITH AN OOZING FLOW LUBRICATOR FOR SPACECRAFT APPLICATIONS

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ABSTRACT

A hybrid bearing cartridge using two retainerless bearings and an oozing flow lubricator was operated in vacuum for 440 days. Conditions included: 12 000 rpm, room temperature, 31 N (7 lbf) preload, 0.52 GPa (75 ksi) mean Hertzian stress, 0.01 Pa vacuum level. Bearing races were made of 440C stainless steel and bearing balls were Si_3N_4. The lubricant was a polyalphaolefin (Nye 176 A). Bearing health was monitored by measuring shaft temperature, run down time, and power supply drop out voltage. Only minor changes in these parameters occurred throughout the test duration. Post test visual analysis of the bearings confirmed their excellent condition. About 2/3 of the original lubricant charge remained in the reservoir. Based on previous flow calculations and measurements, the estimated life for this cartridge is >12 years of nominal operation.

INTRODUCTION

Most spacecraft utilize momentum wheels, reaction wheels or control moment gyros for attitude control. These devices typically use conventional caged ball bearings operating at speeds of several thousand rpm. With the advent of longer duration missions (1), these devices must operate reliably for many years, since replacement is usually not an option. In addition, the progression toward smaller spacecraft, smaller attitude control systems operating at higher speeds may be necessary.

One factor which affects the reliability of these devices are problems associated with the cage or retainer (refs. 2 to 4). These include: retainer squeal, instability, pocket wear and accompanying wear debris, retainer fractures, and retainer lubricant impregnation problems. Removing the retainer eliminates these potential problems which can greatly improve reliability. In addition, the vacated space left by removing the retainer allows for more balls to be placed in the bearing. This increases both the stiffness and load capacity of the bearing and reduces the running torque.

Of course, conventional bearing theory (5) implies that a high speed ball bearing must have a retainer to prevent the balls from contacting each other and thereby causing failure. This results from the fact that the ball to ball contacts have a lubricant entrainment velocity of zero. However, it has been shown that high speed bearings can be operated successfully in the retainerless mode (refs. 6 to 9). Obviously, some kind of lubricant film must exist at these contacts.

The objective of this work was to demonstrate the long term performance of a high speed ball bearing cartridge operating in the retainerless mode. Other conditions included: 12 000 rpm, room temperature operation (~23 °C), 31 N (7 lbf) preload, 0.52 GPa (75 ksi) mean Hertzian stress, and 0.01 Pa vacuum.

EXPERIMENTAL

Bearing Test Facility

The test facility for the bearing cartridge is shown in figure 1. It consists of a vacuum housing, pump, bearing cartridge and low inertia wheel, two hysteresis motors, power supplies and associated electronics. The housing and bearing cartridge are shown schematically in figure 2. The bearing cartridge consists of a duplex pair of 101 size bearings having a pitch diameter of 21.56 mm (0.845 in.). Race material is 440C stainless steel. It utilizes (14) 4.76 mm (0.1875 in.) diameter Si_3N_4 balls in each bearing. Lubrication is supplied by an external lubricator (oozing flow lubricator). The oil reservoir and inner diameter (ID) of the outer races comprise the lubricator. This design is described in reference 6.

The hysteresis motors are powered by two constant frequency, single phase Powertron power supplies (one for each phase). Although each unit could supply up to 250 Vrms, a supply voltage range of 0 to 38 Vrms was found to be optimal.

Bearing Lubricant

The bearing lubricant is a polyalphaolefin (NYE 176A). Properties of this lubricant appear in table I. It is formulated with an antiwear additive (tricresyl phosphate) and a hindered phenol antioxidant.

Oozing Flow Lubricator

As indicated, the lubricator is incorporated into the cartridge. It feeds lubricant to the outer race groove by centrifugal force (caused by the lubricator rotation) through a controlled interface. The flow rate is a function of the interfacial roughness, lubricant viscosity at test temperature and shaft speed. There is no flow during storage. Capacity is ~8 cc (~6.9 g).

Test Conditions

Test conditions include: 12 000 rpm shaft rotation, room temperature (~23 °C), shaft temperature (~31 °C), a preload of 31 N (7 lbf) yielding a mean Hertzian stress of 0.52 GPa (75 ksi). Lubricant flow rate at these conditions averaged ~100 μg/hr for each bearing over the duration of this test.

Test Parameters

Shaft temperature and speed are monitored continuously. Periodically, the drop out voltage and run down time are measured. Typically, three measurements of each are taken. The average of the three is reported. The drop out voltage is analogous to having an iron mass, m, held at an angle θ by a magnetic force, F_m, produced by a voltage, V, across a coil. As long as $F_m \geq mg \tan\theta$, the system is in equilibrium. As V is reduced, eventually F_m becomes less than ($mg \tan\theta$) and equilibrium is lost. The voltage at this point is deemed the “drop out voltage.”

The drop out voltage is measured by slowly reducing the supply voltage (≅3 V/min) until the motors drop out of synchronization. The corresponding voltage is the drop out voltage. After this voltage has been determined, the voltage (and current) is increased until the bearing cartridge is again in synchronization. After 1 min has passed, this process is repeated.

After three measurements of drop out voltage have been made, the run down time is measured. The run down time is the time it takes for the bearing cartridge to drop to half of its initial speed (12 000 rpm) after the supply power is switched off. After this time has been determined, the power supplies are turned back on and the voltage (and current) is increased until the bearing cartridge is again in synchronization. Measurements are repeated after one minute has passed. Both of these techniques are considered to be an indirect measurement of the bearing torque.

RESULTS

The system operated nominally for over 440 days until the test was voluntarily shut down. The run down time, drop out voltage and shaft temperature as a function of time appear in figure 3. No major changes were noted in these parameters. Pretest and post-test low speed dynamometer traces appear in figure 4. A minor increase in post-test torque was observed (18.5 compared to 16 in-oz). The post-test trace showed more torque variation. Teardown of the cartridge was performed. Optical photographs are shown in figures 5 to 9. Figure 5(a) is of side-A before disassembly. Lubricant is obvious on the balls but no meniscus appears at the ball/race interface. Side-B is shown in figure 5(b). Again, lubricant is obvious on the balls and a ball/race meniscus is present. Higher magnification photos of Side-B appear in figures 6(a) and (b). In figure 6(b) there is an indication of some residue on the outer race.

After disassembly, photos of balls from each row were taken and appear in figures 7(a) and (b). The circle in the center is a lighting artifact. A side-A ball appears in figure 7(a) and side-B ball in figure 7(b). Both balls were covered with a thin film of lubricant. Specimens were then solvent rinsed and photos from the outer and inner race appear in figures 8(a) and (b), respectively. On the outer race a series of residue tracks are apparent, but no discernible damage. Similar observations were made on the inner race (fig. 8(b)). Some tracking was apparent on the solvent rinsed balls from side-A (fig. 9), but none on side-B balls. Approximately 4.6 g of lubricant remained in the oil reservoir (~2.3 g were consumed) (i.e. expelled into the housing).

DISCUSSION

Long term performance (440 days) of a retainerless cartridge using an oozing flow lubricator has been demonstrated. Post-test analysis has indicated that the bearings were in excellent condition at test conclusion.

Lubricant Flow Rate Required for Steady State Operation

Kingsbury (ref. 10) has calculated and experimentally verified the amount of lubricant needed for stable operation under starved conditions for this size of bearing. These results indicated that only about 0.2 μg of lubricant/hr is actually needed for steady state operation. Based on the flow rate calculations of Singer (ref. 6), an estimated lubricant flow rate as a function of time for the current test cartridge appears in figure 10. As lubricant is consumed, the centrifugal head is reduced and flow rate decreases. However, it is clear that a flow rate in excess of 10 μg/hr will still exist even after 12 years of operation. Therefore, the life expectancy of this test combination is >12 years.

SUMMARY OF RESULTS

1. Retainerless operation with a hybrid bearing (440C races and Si_3N_4 balls) at 12 000 rpm was successfully operated for 440 days in vacuum.
2. Based on the amount of lubricant remaining in the reservoir, an average flow rate of about 100 μg/hr was estimated during the life test.
3. Based on this flow rate calculations, a projected life >12 years can be estimated for this system.

CONCLUSION

The use of hybrid retainerless bearings for spacecraft applications is an attractive option for future missions.

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TABLE I. SELECTED PROPERTIES OF NYE SYNTHETIC OIL 176A

Kinematic viscosity @ 210 °F, cs	40
@ 100 °F, cs	423
@ 0 °F, cs	30 000
Viscosity index	142
Density @ 60 °F, g/ml	0.85
Flash point, °F	575
Pour point, °F	–45
Evaporation, 6-1/2 hr @ 350 °F, percent	1.0
Neutralization number, mg KOH/g	0.02

Figure 1.—Test facility.

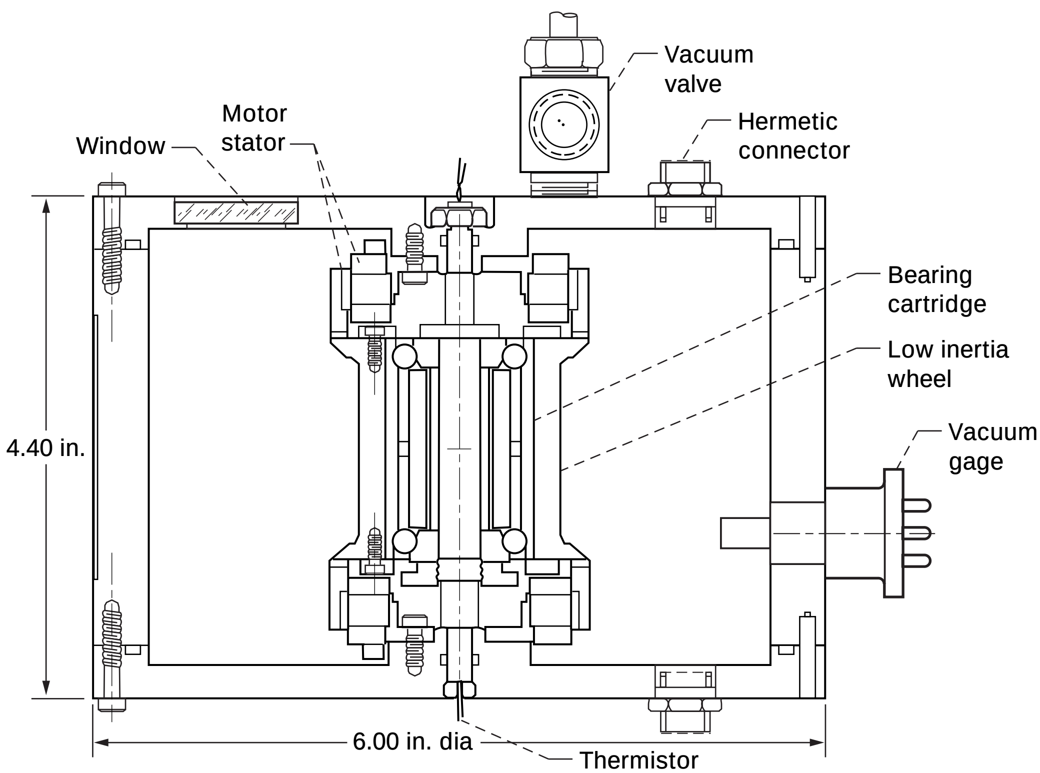

Figure 2.—Vacuum housing, bearing cartridge, and low inertia wheel.

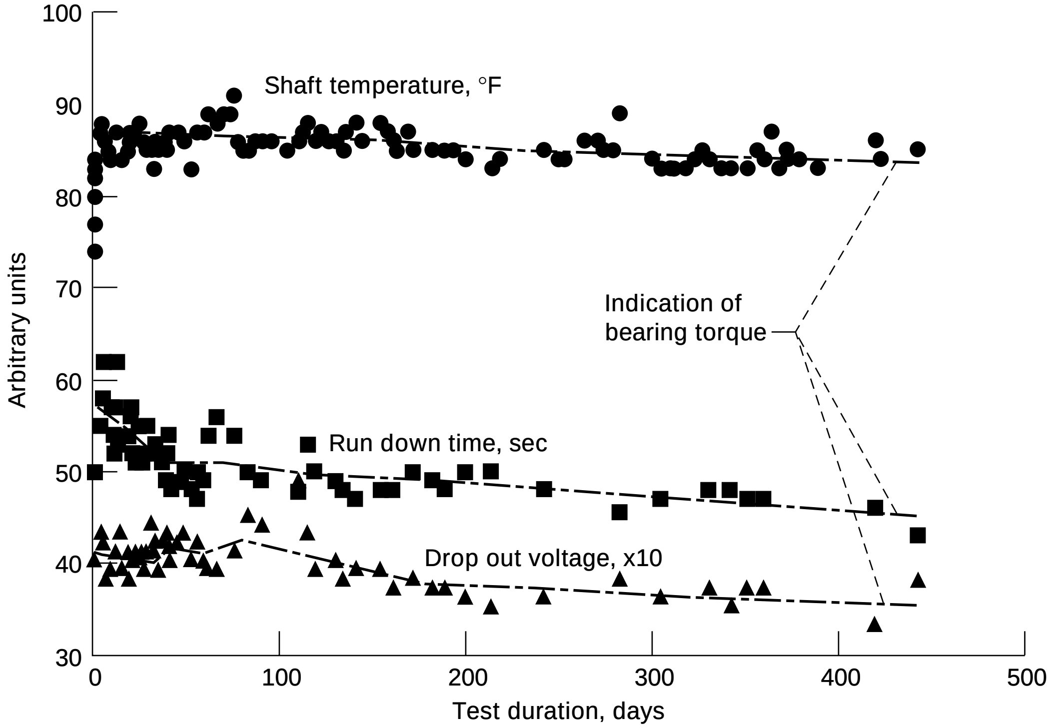

Figure 3.—Life test of retainerless bearing pair (12,000 rpm, 0.01 Pa vacuum).

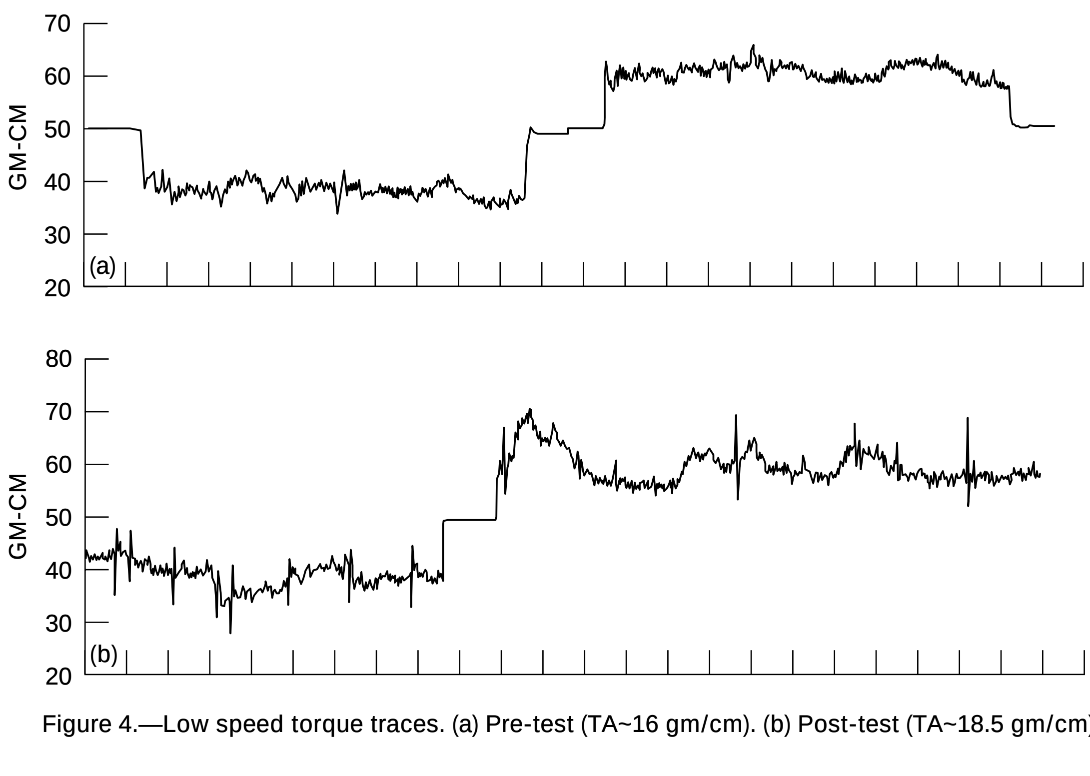

Figure 4.—Low speed torque traces. (a) Pre-test (TA~16 gm/cm). (b) Post-test (TA~18.5 gm/cm).

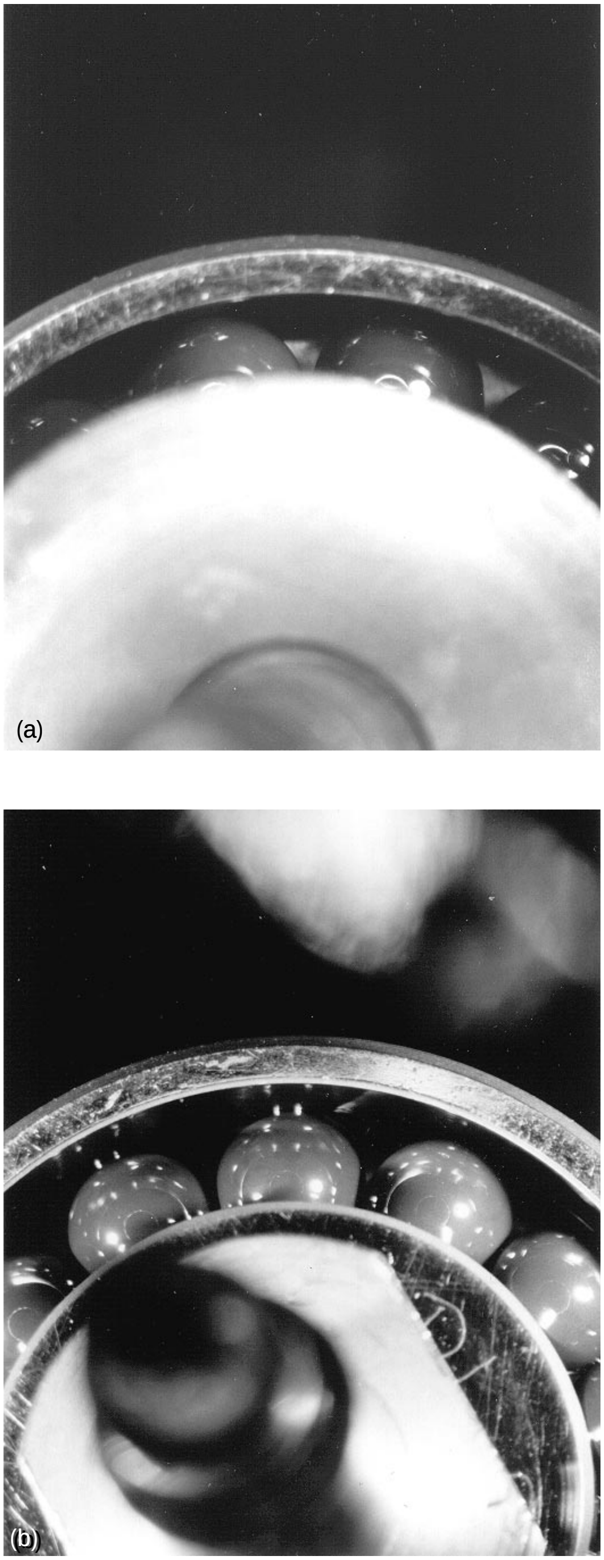

Figure 5.—Post test (assembled). (a) Side A. (b) Side B.

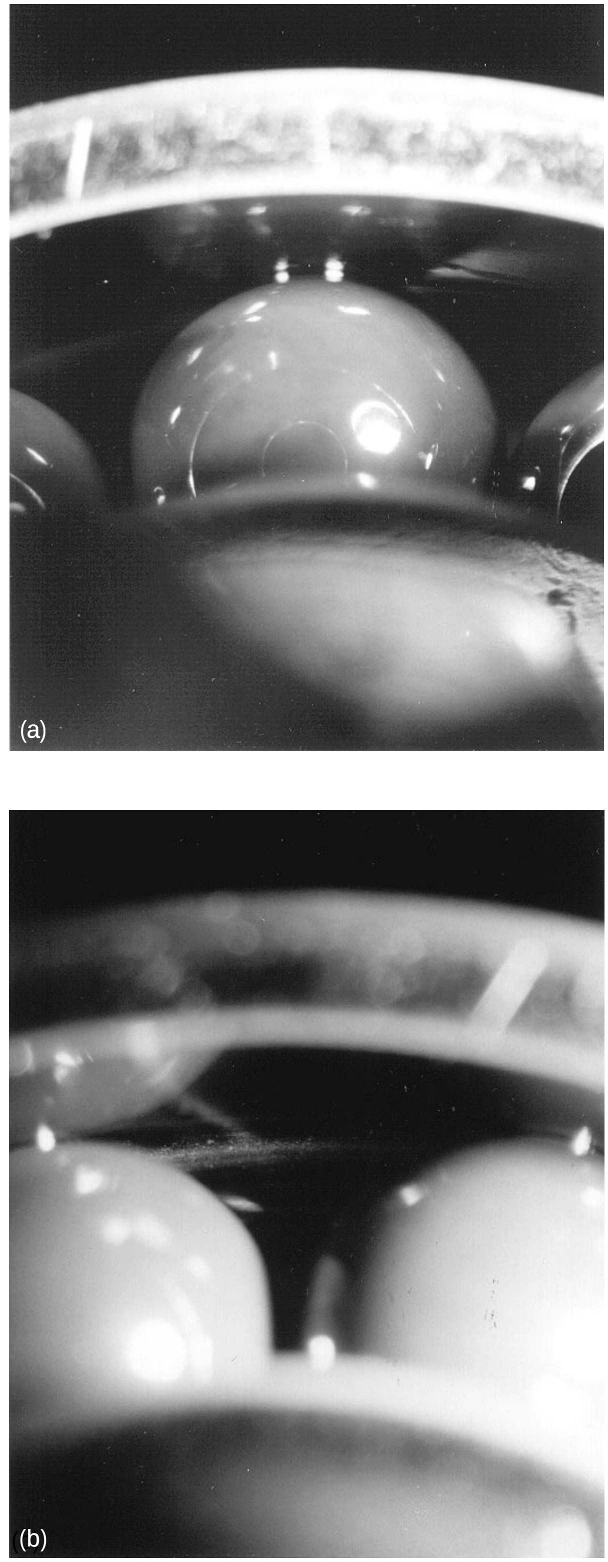

Figure 6.—Post test (assembled). (a) Side B, Ball. (b) Side B, outer race.

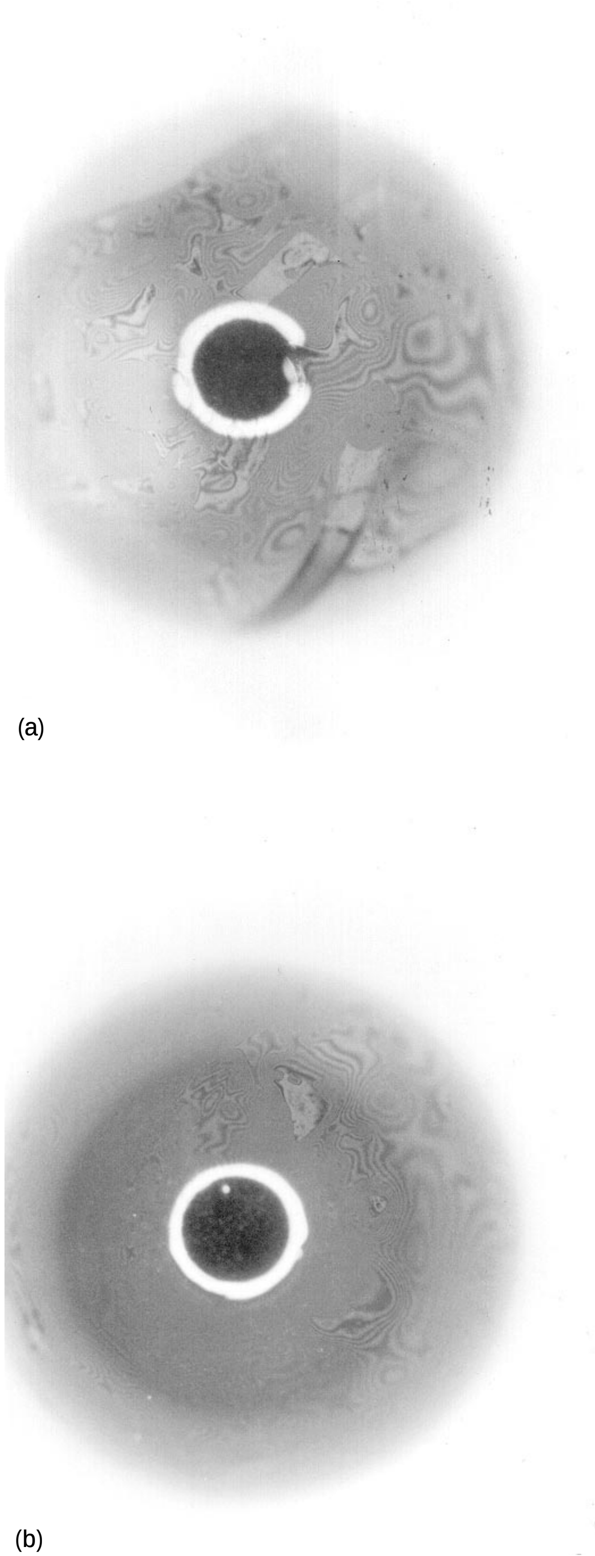

Figure 7.—Post test (wet). (a) Side A ball. (b) Side B ball.

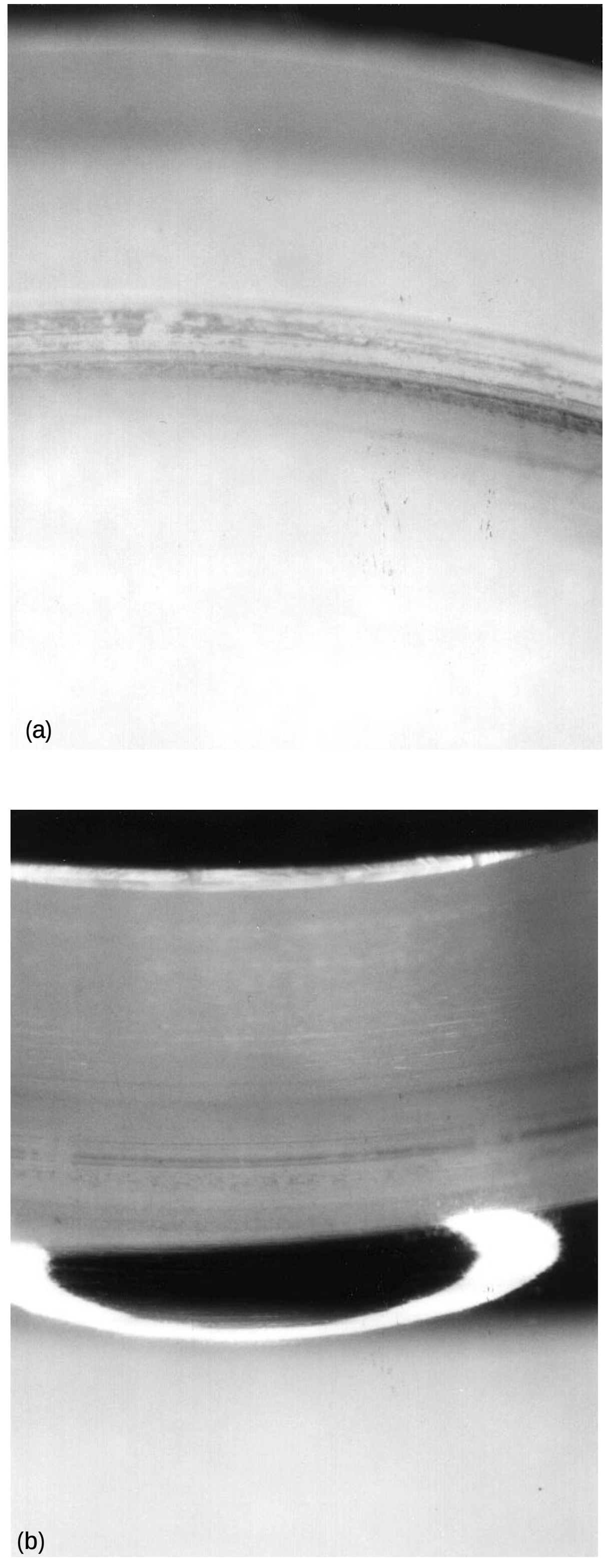

Figure 8.—Post test (after solvent rinse). (a) Side A outer race. (b) Side B inner race.

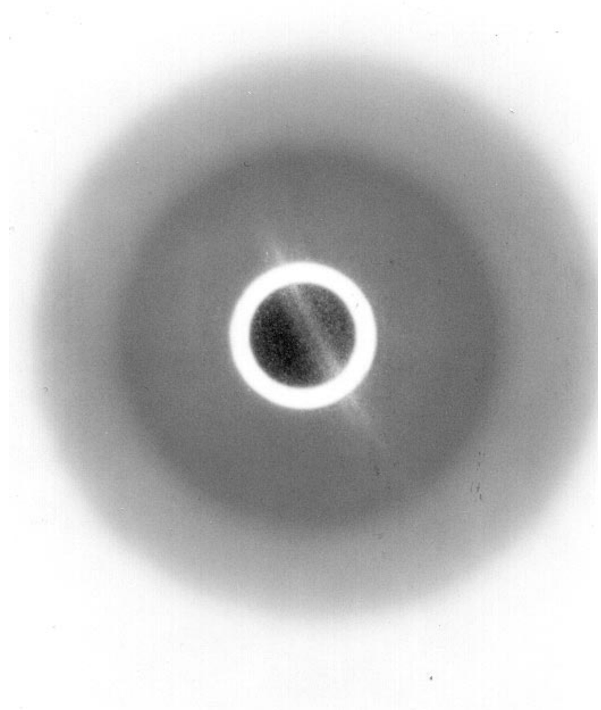

Figure 9.—Post test (after solvent rinse) side A ball.

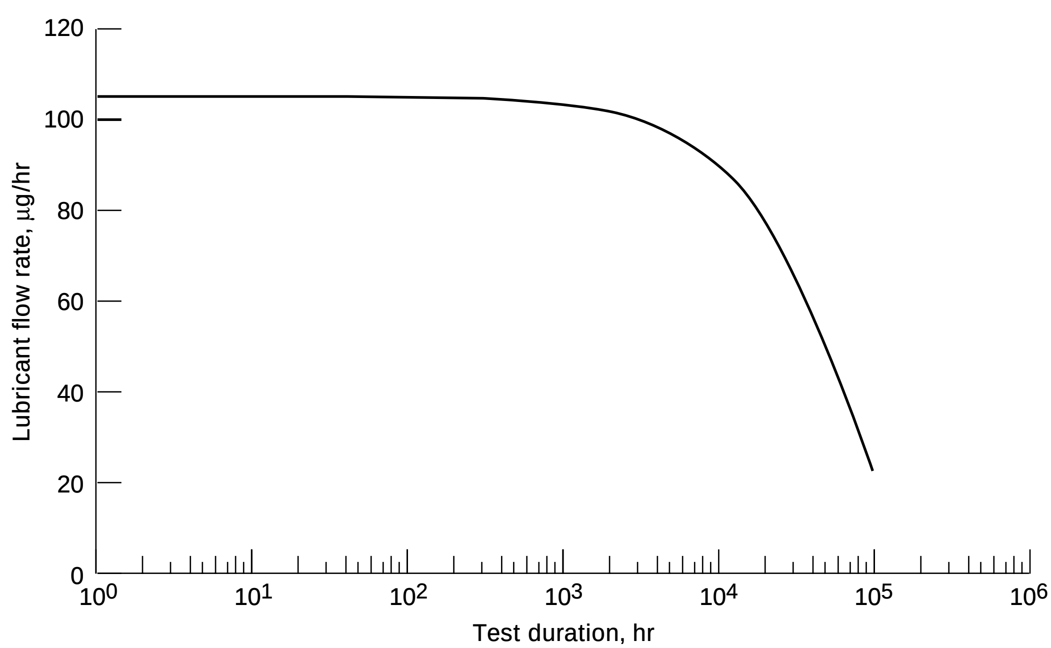

Figure 10.—Calculated lubricant flow rate per bearing as a function of test duration.

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