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Wave Fluid Film Bearing Tests for an Aviation Gearbox

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WAVE FLUID FILM BEARING TESTS FOR AN AVIATION GEARBOX

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ABSTRACT

An oil-lubricated wave journal-thrust bearing assembly was successfully tested at conditions found in general aviation engine gearboxes. The bearing performed well at both steady state conditions and in start-stop tests. It ran stably under all loading conditions, including zero load, at all speeds up to 16 000 rpm. The bearing carried 25 percent more load than required for the gearbox application, supporting 8900 N (94 bars average pressure), and showed very good thermal stability. 450 start-stop cycles were also performed, including 350 cycles without oil supply during starting and stopping. Test results and numerical predictions were in good agreement.

INTRODUCTION

Transmission noise levels can be substantially reduced (up to 10 dB) if journal bearings are used in place of rolling-element bearings (Drago, 1990). A low noise level means less wear and longer gear life, and increased comfort for the transmission users. Journal bearings have been used successfully in planetary aero-transmissions for more than 30 years. Measurements of planetary gear vibration showed the damping effect of fluid film bearings (Botman, 1980). A review of this technology was presented by Badger et al. (1994) for transmissions of 400 to 1200 kW capacity. The planet gears of these transmissions were supported by plain journal bearings.

Wave bearing technology, developed since 1990 mainly for gas lubrication, (Dimofte, 1993, 1995; Dimofte, et al., 1994, 1995, 1996) has been incorporated in a new planet gear bearing design. These wave fluid bearings were designed to replace the rolling element bearings used in a 600 kW general aviation gearbox in support of the NASA General Aviation Propulsion (GAP) Program. The new bearing design is a combination journal-thrust bearing assembly that can carry a radial load equal to or greater than the gear load and also support a thrust load up to 10 percent of the radial load. The wave bearing technology was used to stiffen and better lubricate the bearing. Hard materials were selected for both the shaft and the bearing sleeve to maintain the bearing geometry under high loads and to simplify the design. This design eliminates the use of a sleeve pressed into the planet gear. Coatings were applied to all bearing surfaces to prevent seizure and ensure a low wear rate.

The objective of the present work is to verify the analytical predictions under conditions corresponding to those in a turboprop gearbox, and thus determine the suitability of wave bearings for this application. The journal-thrust wave bearing was analyzed and its performance was experimentally evaluated. The analysis was based on the Reynolds lubrication equation similar to that presented by Dimofte (1993, 1995). Turbulent flow, viscosity-temperature relations, centrifugal growth of the bearing diameter, and the deformation of the bearing under gear loads were also taken into account. Following a description of the test apparatus, results of the radial load, axial load, and the start-stop tests are presented.

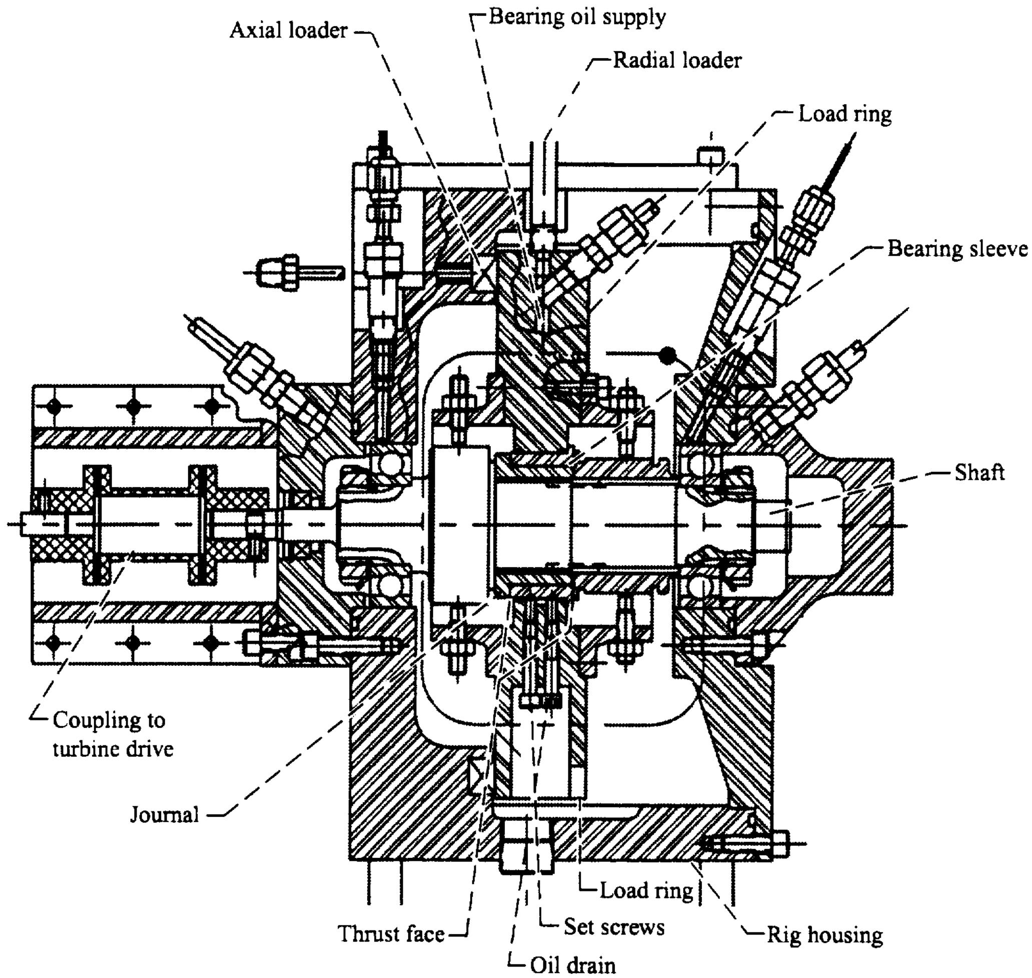


Figure 1.—Wave oil bearing rig cross section.

Test Apparatus

The oil wave bearing test rig was designed, fabricated and assembled at NASA Glenn Research Center. As the cross section in figure 1 shows, the rig has a shaft supported by ball bearings at both of its ends with the test wave journal-thrust bearing located between them. The shaft is elastically coupled to a turbine drive (not shown) and can be run to speeds up to 20 000 rpm. The rig housing is supported at its mid-horizontal plane to eliminate thermal misalignment between the rig shaft and the turbine drive shaft as the rig heats up.

The test bearing sleeve is mounted in the load ring. Set screws in the load ring are used to deform the bearing sleeve to generate the wave profile on the bearing sleeve inner diameter. The shoulders on the journal and the thrust bearing sleeve provide the faces for the wave thrust bearing. All test bearing parts were manufactured from carburized Carpenter Pyrowear-53 specialty alloy and were initially coated with Balzers Balinit WC/C (tungsten-carbide/carbon) coating. However, the rotating thrust bearing surfaces were re-ground after the rotor was assembled to adjust the thrust bearing clearances, which removed the coating from these surfaces. The test bearing dimensions are shown in table 1. The journal bearing profile was measured at both journal bearing edges; the measured profile can be seen in figure 2.

A pneumatic cylinder was used to apply radial load to the wave bearing through a rod connected to the load ring. Axial load was applied to the thrust wave bearing by an air-bladder between the rig housing and the load ring. Radial and axial loads up to 8900 N (2000 lb) and 890 N (200 lb), respectively, can be applied. Oil (MIL–L–23699) is supplied through the load ring to the center of the wave journal bearing and drains out the bottom of the rig housing. The oil supply system can deliver oil at pressures up to 6.2 bars (90 psi) and temperatures up to 150 °C (300 °F). Hot oil was also trickled over the outside of the load ring to heat the load ring and keep its temperature constant.

To determine bearing performance the following measurements were made: shaft speed, radial load, axial load, bearing inlet oil temperature and pressure, journal bearing exit oil temperature and pressure, thrust bearing exit oil temperature, oil flow rate, and metal temperature of the bearing sleeve. Due to noise problems with the proximity probes, data to determine the relative position of the shaft to the bearing was not obtained. Data was recorded at the start and finish of each period of time at a given test condition and at each change point of run condition during the start-stop cycles (as many as 5 per cycle).

Table 1.—Wave bearing dimensions.

Journal diameter, mm	45
Journal length, mm	21
Wave amplitude, μm	5.5
Average Radial clearance, μm	19
Thrust OD, mm	56
Thrust ID, mm	48
Thrust side clearance, μm	24
Thrust wave amplitude, μm	6.7

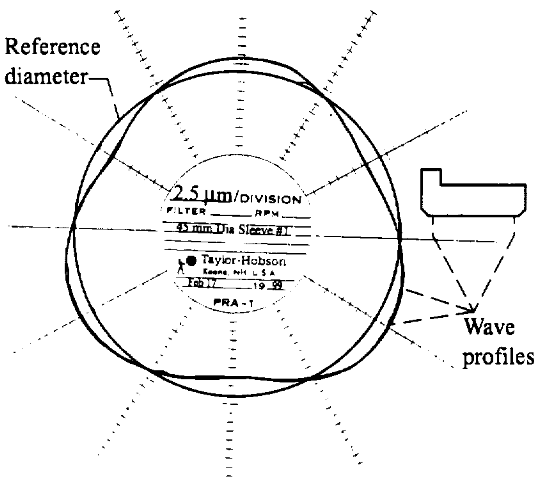


Figure 2.—Measured profile at both edges of the bearing sleeve inner diameter prior to test.

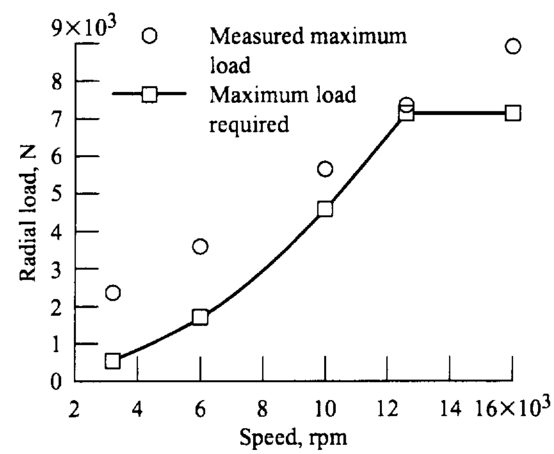


Figure 3.—Measured and required maximum radial loads.

Radial Load Tests

The wave bearing was tested sequentially at five speeds (3184, 5970, 9550, 12 600, and 16 000 rpm) under radial loads from 0 to 8900 N (2000 lb). Dwell time at each condition was about 10 minutes to allow the temperature to reach steady state. The oil supply pressure varied with speed to simulate the oil supply available in the gearbox application. The oil temperature was nominally 120 °C (250 °F), but varied by about 28 °C (50 °F) due to changes in flow rate. The bearing ran stably under all test conditions. The maximum load applied to the bearing at each speed is plotted in figure 3 as are the required maximum loads for the general aviation gearbox. The maximum load applied was determined as follows: at the lowest test speed (3184 rpm) load was increased until the shaft speed started to drop (due to increasing friction at high bearing eccentricity). Maximum loads applied at other speeds were then determined by linear extrapolation. The predicted minimum film thickness for all maximum load cases was between 2.8 and 3.1 μm (110 and 122 μ-inch). It can be seen that at 16 000 rpm the wave bearing can carry a load 25 percent greater than required (8900 N actual max load vs. 7120 N required) and the maximum average pressure was 94 bars (1366 psi).

Comparisons between measured and predicted oil temperature rise across bearing, oil flow rate, and friction loss at maximum load tested can be seen in figures 4, 5, and 6, respectively. These figures show a good agreement between measured and predicted data. The oil temperature rise does not exceed 30 °C (54 °F). The maximum flow rate of 900 ml/min (0.24 gpm) occurs at 16 000 rpm. The friction loss increases with speed as expected and is 842 W (1.13 hp) at 16,000 rpm. The experimental friction loss is based on the measured oil flow rate and temperature rise. The bearing performance under radial loads is acceptable for the gearbox application.

A visual inspection was made after testing at 9550 rpm. Both the journal and bearing sleeve surfaces were found in good condition, indicating that the bearing and the coating worked very well.

In the initial test at 16 000 rpm the turbine did not have enough power to drive the shaft when the load reached 7560 N (1700 lb) and rotation stopped suddenly under that load. Subsequent testing at 16 000 rpm was achieved by increasing the turbine power. After completing the radial load tests, polish marks were found on the journal bearing surfaces as shown in figures 7(a) and 7(b). These polish marks show that the shaft bent unsymmetrically under large loads and forced the bearing to run in a misaligned condition. However, the coating worked well and no damaged surface regions were found.

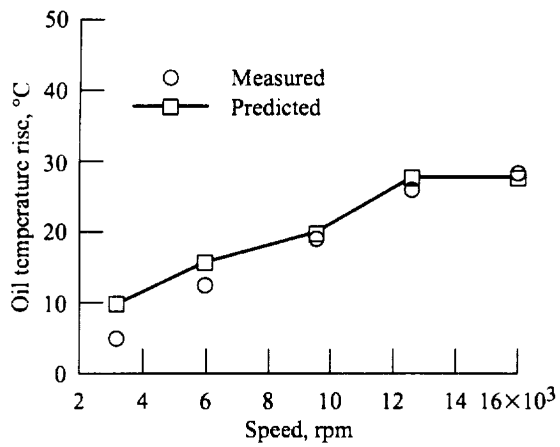


Figure 4.—Measured and predicted oil temperature rise across wave bearing for maximum radial load tested.

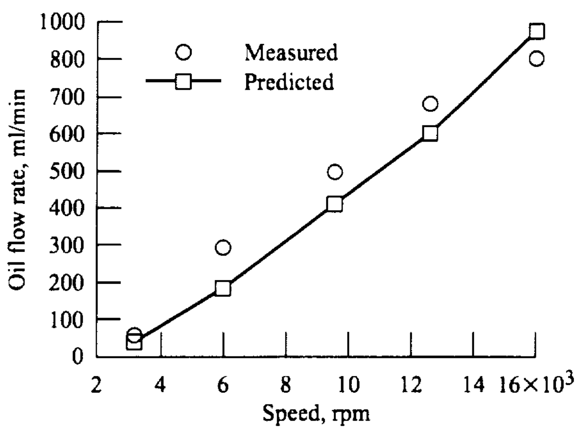


Figure 5.—Measured and predicted oil flow rate for maximum radial load tested.

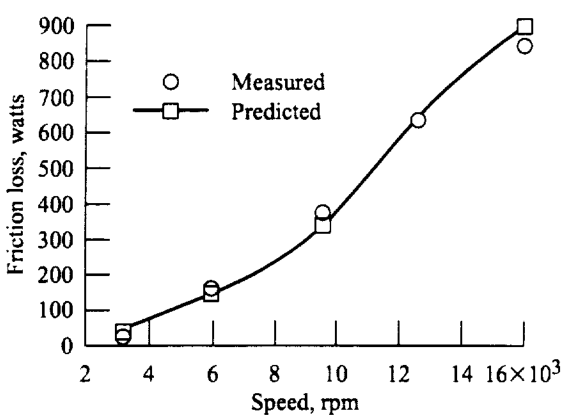


Figure 6.—Measured and predicted friction loss for maximum radial load tested.

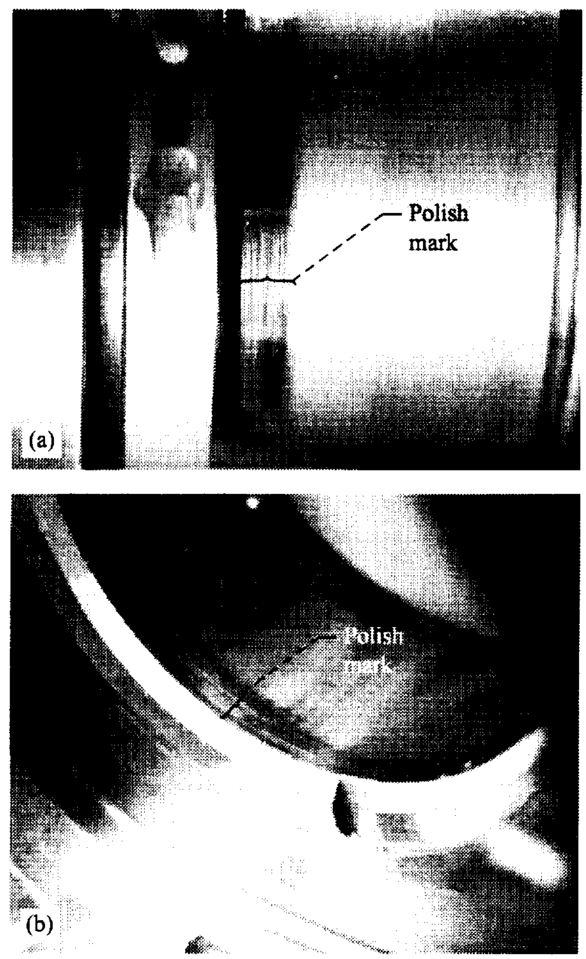
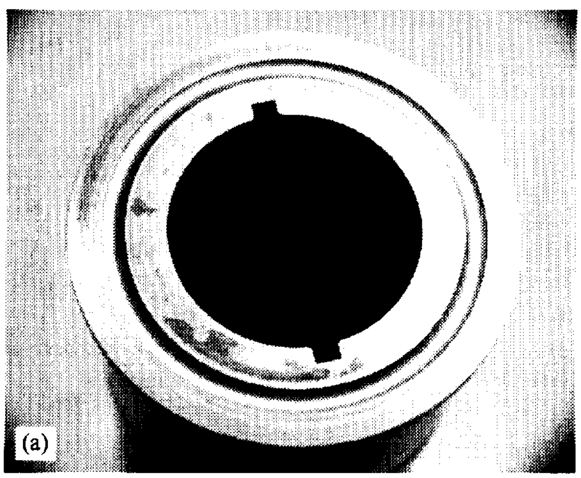


Figure 7.—Journal bearing surfaces after 8909 N (2002 lbf) radial load at 16 000 rpm and an accidental stop under a radial load of 7565 N (1700 lbf) from 16 000 to 0 rpm. (a) Journal surface. (b) Sleeve surface.

Axial Load Tests

Axial loads up to 667 N (150 lb) were applied successfully at 12 600 rpm. However, a direct contact between the thrust bearing surfaces occurred at a slightly higher axial load of 690 N (154 lb) due to misalignment of the rig axial loading system and the sharp edges of the thrust rotating surfaces. The turbine could not maintain speed and the rig stopped under 690 N axial load. Some damage occurred at the thrust stationary face and rotating surface (figures 8(a) and 8(b)) and demonstrated that a coating on only one surface cannot provide adequate protection. After this incident, bearing tests under axial load were stopped to conserve the assembly for start-stop tests.

The double thrust wave bearing concept was successfully demonstrated but it appears that all bearing surfaces must be coated to properly protect the bearing. The bearing edges should be rounded to preclude edge contact.



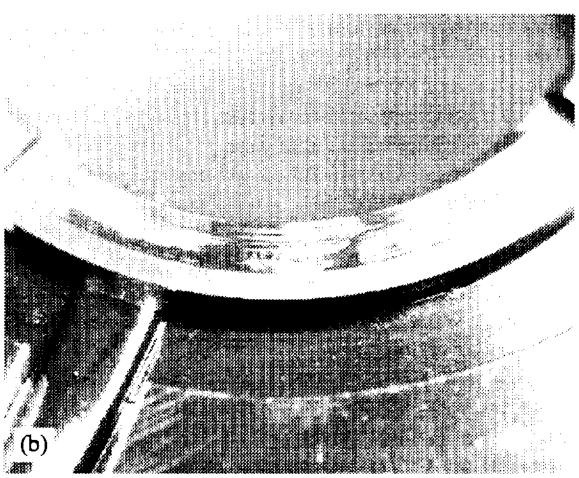


Figure 8.—Thrust bearing surfaces after an accidental stop under an axial load of 685 N (154 lbf) from 12 638 to 0 rpm. (a) Thrust bearing rotating surface. (b) Thrust bearing stationary face.

START AND STOP CYCLES

Start-stop tests were performed between 0 and 3200 rpm (the anticipated idle speed of the gearbox) for various loads and lubrication conditions as shown in table 2. The load was held constant during these tests, which is a more severe load condition than in the gearbox where the load is proportional to speed squared. The oil supply pressure to the bearing for the start-stop tests was 0.65 bars (9.4 psig). In test 1 the oil supply was continuous. In tests 2 and 3 the oil supply was off for 10 seconds prior to and after shaft rotation was started and it was shut off 10 seconds prior to stopping the shaft rotation. Testing with an intermittent oil supply was done to simulate an expected delay in getting oil to the bearing in the gearbox application. Test 2 had a fast start (5 seconds) and test 3 had a slow start (35 seconds) to bound the possible start sequences for the gearbox. In tests 2 and 3 the load was reduced to 67 N (15 lb) to better match the gearbox condition.

Table 2.—Start and stop tests.

Test No.	Lubrication	Number of Cycles	Load, N (lbf)	Time at speed, min		Acceleration Time, seconds
				3200	0 rpm	
1	Continuous	125	111 (25)	2	2	5
2	Intermittent	100	67 (15)	1	1	5
3	Intermittent	200	67 (15)	2	2	35

In test 1, the bearing was inspected after 25, 75, and 125 cycles. The bearing surfaces were found to be without any significant change. In test 2 the bearing surfaces were again inspected after 50 and 100 cycles and were found without any significant changes.

At this time the bearing was run to 16 000 rpm and a 1700 lb radial load was applied. The bearing ran very well without any change in its performance.

After 200 cycles were completed in test 3 the bearing parts were visually inspected. Again, the bearing surfaces were found without any significant change. Thus it is likely that the bearing can perform an "unlimited number" of such cycles without any significant wear. Moreover, review of all visual inspections of the bearing sleeve that were made throughout the test period shows that the polish marks, and hence the minimum fluid film thickness, was always in the same position; ~40–50 degrees from the load direction as the theory predicted. This means that the wear on the sleeve surface was always located in a "valley" of the wave profile and did not affect the inside peaks of the waves. Because the peaks of the waves are responsible for the bearing performance, this means that the wave journal bearing performance will not degrade over the bearing life.

After the start-stop tests, an attempt was made to verify the performance of the bearing assembly at maximum speed and load. In these tests, four accidental stops occurred while at high speeds (16 000 to 17 000 rpm) and large loads (1980 to 6370 N (445 to 1431 lb)) which are 30 to 100 times the loads that would be seen in the gearbox application during starts and stops. These accidental stops were mainly due to the changes to the turbine controller when the automatic cycle was added to the system. Since the shaft was found to rotate freely after each of these unexpected shutdowns, attempts were made to increase the turbine power by all means including increasing the running speed by small amounts. However, after 4 unexpected shutdowns with high radial loads the test was suspended and a visual inspection was performed. The journal surface and the bearing sleeve surface had damage to the coating near the turbine end of the bearing over part of the polished areas seen earlier, as shown in figures 9(a) and 9(b). Since repairs to the turbine controller couldn't be done immediately, testing was ended and the bearing sleeve

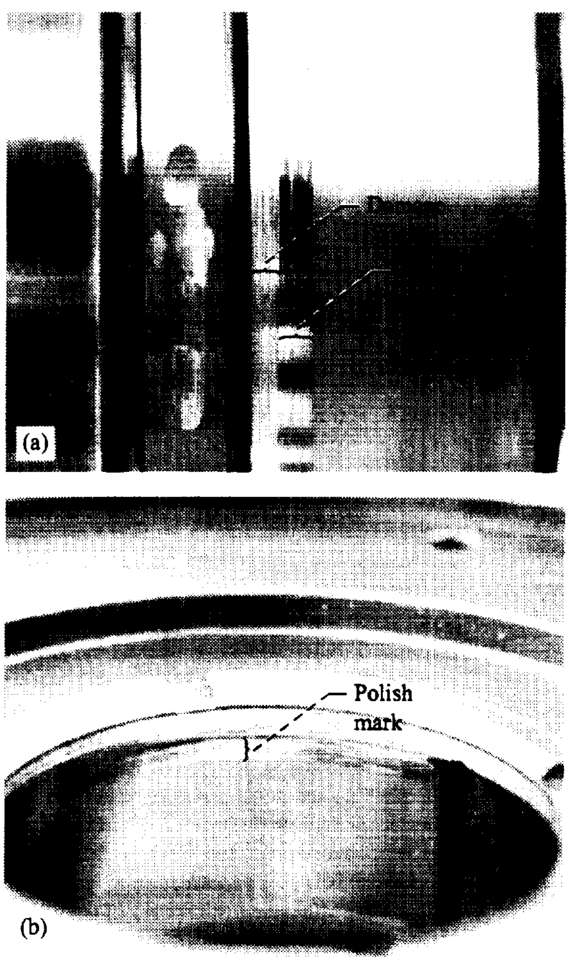


Figure 9.—Journal bearing surfaces at the end of all tests. Four accidental stops occurred from speeds over 16 000 rpm to 0 rpm and radial loads from 1980 N (445 lbf) to 6370 N (1431 lbf) that damaged the coating on both the journal and sleeve. (a) Journal surface. (b) Sleeve surface.

profile was measured. The original wave profile was maintained over more than 90 percent of the bearing length. The wave profile of the damaged area is shown in figure 10. It is believed that this damage resulted from the shaft bending in a manner that applied a heavier load on one end of the bearing. Although slightly damaged, the coating prevented seizure and catastrophic failure of the bearing.

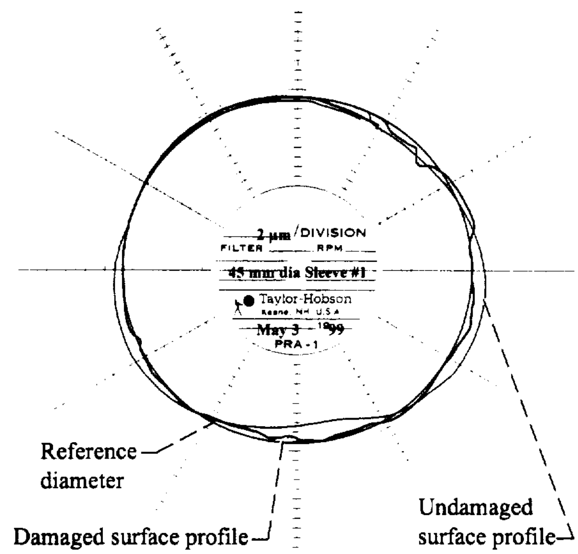


Figure 10.—Measured profile at both edges of the bearing sleeve inner diameter after tests were completed.

CONCLUDING REMARKS

An oil lubricated wave journal-thrust bearing was successfully demonstrated in rig tests simulating gearbox loads, speeds, and lubrication supply. The bearing performed very well under radial loads up to 8900 N (2000 lb) at 16 000 rpm demonstrating an average bearing pressure over 94 bars (1366 psi). Both bearing dynamic and thermal stability were excellent. Good agreement was found between measured and predicted temperature rise, flow rate, and friction loss.

Axial load capability of the thrust bearing was demonstrated up to 667 N (150 lb) at 12 600 rpm. Some damage was observed on the uncoated thrust bearing surface after rotation stopped while under heavy load. Coating both bearing surfaces should improve performance.

The journal bearing showed excellent behavior in the start and stop tests, withstanding several hundred starts and stops with minimal bearing distress. Shutting off the oil supply during starts and stops did not affect performance.

The tested tungsten-carbide/carbon coating worked very well for conditions expected in the gearbox application and, based on its performance during unexpected stops at high speeds and large loads, can be expected to prevent the bearing from seizing in a catastrophic manner.

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