NASA Technical Paper 1808

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FEBRUARY 1981

NASA

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Scientific and Technical
Information Branch

1981

Summary

Parametric tests were conducted in a high-speed bearing tester with a 35-millimeter-bore, angular-contact ball bearing having a single-outer-land guided cage. Lubrication was achieved by flowing oil through axial grooves and radial holes machined in the inner ring of the bearing. Test parameters were a thrust load of 667 newtons (150 lb), shaft speeds from 48 000 to 72 000 rpm, and an oil-inlet temperature of 394 K (250° F). Lubricant flow rates to the bearing ranged from 300 to 1900 cm^3/min (0.08 to 0.50 gal/min). An outer ring cooling oil flow rate maintained at 1700 cm^3/min (0.45 gal/min) at 394 K (250° F) oil-inlet temperature was used in some tests. Data from tests where the distribution of the total oil supplied to the inner ring was 50 percent for bearing lubrication and 50 percent for inner ring cooling were compared with data from tests where the distribution was 25 percent lubrication and 75 percent cooling. The results of these tests were also compared with reported results obtained with a jet-lubricated bearing with identical dimensions and cage design in a previous investigation.

Successful operation of the 35-millimeter-bore bearing with through-the-inner-ring lubrication was accomplished to 2.5 million DN at a thrust load of 667 newtons (150 lb) for both oil distribution patterns. Cooler bearing operation was experienced with a total oil distribution of 50–50 percent than with 25–75 percent. However, data from a previous study with jet lubrication showed lower outer ring and higher inner ring temperatures than did the data from the inner-ring-lubricated bearings of this investigation.

Outer ring cooling for both oil flow distribution patterns of this investigation and for the referenced jet-lubricated bearing resulted in a substantial decrease in outer ring temperature but had a minimal effect on inner ring temperature. A maximum power loss of 2.8 kilowatts (3.7 hp) occurred at 72 300 rpm with an inner ring oil flow distribution pattern of 25–75 percent at a total oil flow rate of 1900 cm^3/min (0.50 gal/min). The power loss increased a maximum of 1.3 kilowatts (1.7 hp) over the speed range 47 200 to 72 300 rpm at an oil flow rate of 1900 cm^3/min (0.50 gal/min)for all lubrication methods employed. A maximum cage slip of 7.0 percent occurred at 72 300 rpm at a total oil flow rate of 1900 cm^3/min (0.50 gal/min) with a total oil flow distribution pattern of 50–50 percent. The increase in percent cage slip with lubricant flow rate was minimal for each lubrication method used.

Introduction

Small advanced engines, 0.5 to 4.6 kg/sec (1 to 10 lb/sec) of total airflow, require bearings that can operate in the speed range 2.5 million DN (product of the bearing bore in millimeters and the shaft speed in rpm) at high temperatures to achieve the performance objectives set by the U.S. Army for programs such as STAGG (Small Turbine Advanced Gas Generator) and UTTAS (Utility Tactical Transport Aircraft System). The bearing designs and lubrication techniques used must be refined and optimized for reliable engine performance and long bearing life.

There is a limiting DN value above which a jet-lubricated bearing is no longer adequate. The centrifugal forces prevent the oil jet from properly lubricating and cooling the rolling elements and cage, and this results in bearing thermal instability or cage wear. This limiting DN value is about 2.5 million for small-bore bearings. With sufficient oil flow and proper cage design this limit can be increased slightly to 2.8 million as was achieved in reference 1 for a 30-millimeter-bore, deep-groove ball bearing with an outer-land-guided cage.

In an effort to overcome the detrimental centrifugal effects in high-speed applications, which cause some of the lubricant supplied by jets to be slung off the inner ring, oil can be supplied to the bearing through grooves in the bore and then through radial holes from the grooves to the rolling elements of the bearing. This method has been successfully applied to large-bore ball and roller bearings operating to 3.0 million DN (refs. 2 to 4). Even though customarily used, DN is a somewhat misleading severity parameter unless comparisons are limited to a narrow band of bore diameters. Centrifugal effects vary as DN^2 so that in small-bore bearings these effects can easily be eight times as severe as in large-bore bearings (ref. 5). Effective lubrication and cooling in a very severe centrifugal force field are the principal problems in achieving successful operation of small-bore bearings at ultra high speeds. Inner ring lubrication might be a solution although the limited space available for machining grooves and radial holes can be a problem with small bearings.

Reference 6 reports results of a high-speed, jet-lubricated ball bearing with a single-outer-land-guided-cage. The results of the experimental tests reported herein were obtained with a 35-millimeter-bore ball bearing dimensionally identical to that in reference 6. The only difference is a modification of the inner ring, where radial holes are machined in the axial grooves of the bearing bore to facilitate inner ring lubrication.

The primary objective of this study was to determine the operating characteristics of the bearing under varying coolant and lubricant flow rates for both inner and outer ring cooling and inner ring injected lubrication. In some tests the distribution of the total oil supplied to the inner ring was 50 percent for bearing lubrication and 50 percent for inner ring cooling. In other tests 25 percent of the oil was used for lubrication and 75 percent for cooling. A secondary objective was to compare the performance of a jet-lubricated bearing (ref. 6) with that of a dimensionally identical bearing having inner ring injected lubrication.

The bearing had a nominal, unmounted contact angle of 24° and a single-outer-land-guided cage. Provisions were made for inner ring lubrication of the bearing and for outer ring cooling. Test conditions were a thrust load of 667 newtons (150 lb), shaft speeds from 48 000 to 72 000 rpm, and an oil-inlet temperature of 394 K (250° F). The oil was injected to the inner ring at flow rates from 300 to 1900 cm^3/min (0.08 to 0.50 gal/min). Outer ring cooling oil flow rate was maintained at 1700 cm^3/min (0.45 gal/min) at 394 K (250° F) oil-inlet temperature. The lubricant was a neopentylpolyol (tetra) ester that meets the MIL-L-23699 specifications.

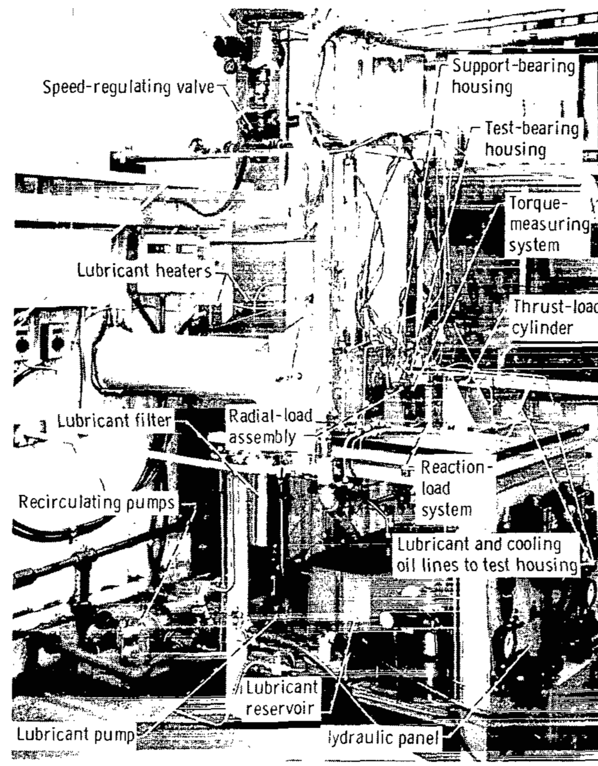


Figure 1. - High-speed, small-bore-bearing test machine.

Apparatus

High-Speed Bearing Tester

A general view of the air-turbine-driven test machine is shown in figure 1. A sectional drawing is shown in figure 2. The shaft is mounted horizontally and is supported by two preloaded, angular-contact ball bearings. The test bearing is assembled into a separate housing that incorporates the hardware for lubrication, oil removal, thrust and radial load application, and instrumentation for cage speed measurement. Test bearing torque is measured with strain gages located near the end of an arm that prevents the housing from rotating. Thrust force is applied through a combination of a thrust needle bearing and a small roller support bearing to minimize test-housing restraint during torque measurements. The test bearing was lubricated through the inner ring. Oil was pumped by centrifugal force from the center of the hollow shaft through axial grooves in the test-bearing bore and through a series of small radial holes, 0.762 millimeter (0.030 in.) in diameter, to the bearing inner race. Those axial grooves in the bearing bore that did not have radial holes allowed oil to flow under the ring for inner ring cooling. To vary the distribution of the total oil flow for lubrication and for inner ring cooling, certain radial holes were plugged before the test bearing was installed. Cooling oil was supplied to the outer ring by means of holes and grooves in the bearing housing, as shown in figure 2.

Shaft speed (inner ring speed) was measured with a magnetic probe. Ball-pass frequency (cage speed) was determined by analyzing signals from a semiconductor strain gage mounted on the inside diameter of the test-bearing housing. Two thermocouples were assembled in the shaft to measure inner ring temperatures through a rotating telemetry system. Outer ring temperatures were obtained by two thermocouples installed in the test-bearing housing. The high-speed bearing tester is described in detail in references 6 and 7.

Test Bearing

The test bearing was an ABEC-7 grade, 35-millimeter-bore, angular-contact ball bearing with a single-outer-land-guided cage, as shown in figure

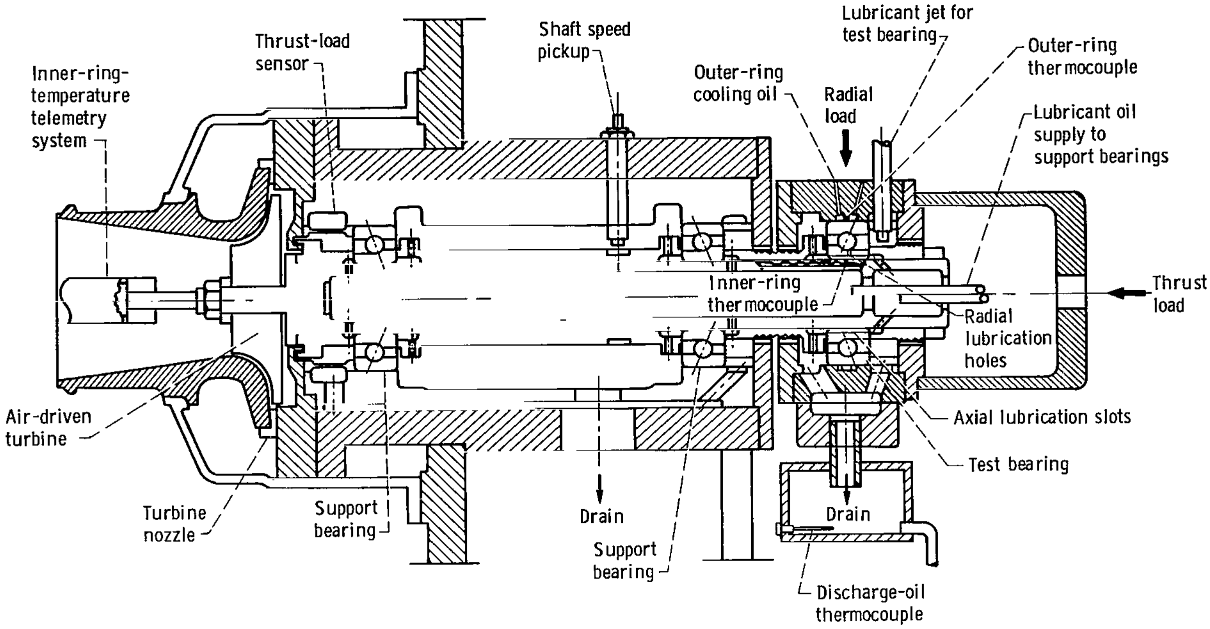
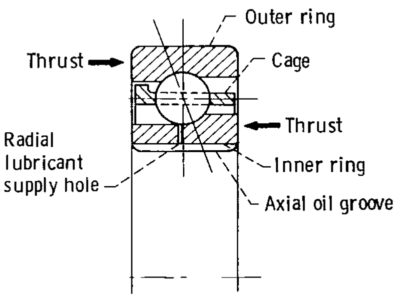
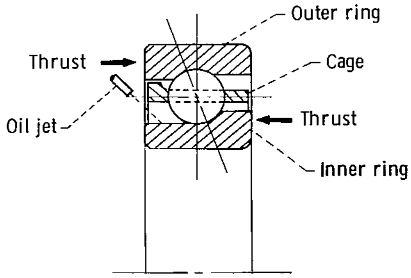


Figure 2. - Schematic of high-speed, small-bore-bearing test rig.



(a) Inner-ring lubricated.



(b) Jet lubricated.

Figure 3. - Angular-contact ball bearing.

3(a). The bearing contained 16 balls each with a nominal diameter of 7.14 millimeters (0.281 in.). The bearing design permitted lubrication through the inner ring by means of axial grooves machined in the bore. There were 16 axial grooves in the bearing bore. Eight 0.762-millimeter (0.030-in.) diameter holes (one in every other axial groove) radiating from the bearing bore formed a flowpath for bearing lubrication. Therefore it was assumed that 50 percent of the oil supplied to the inner ring lubricated the bearing and 50 percent flowed axially through those grooves that contained no radial holes. The latter flow cooled the inner ring. In some tests, four of the eight radial holes were plugged to allow 25 percent of the total flow to be used for bearing lubrication and 75 percent for inner ring cooling. The results of these tests were compared with those of a jet-lubricated bearing with identical dimensions and cage design described in reference 6 and shown in figure 3(b).

The inner and outer rings and the balls were manufactured from consumable-electrode-vacuum-melted AISI M-50 steel. Nominal hardness of the balls and rings was Rockwell C62 at room temperature. The cage was made from AISI 4340 steel (AMS 6415) heat treated to Rockwell C28 to C36 hardness and completely plated with 0.0203- to 0.0381-millimeter (0.0008- to 0.0015-in.) thick silver (AMS 2412). The cage balance was within 0.05 g-cm (7×10^{-4} oz-in.). More complete specifications are shown in table I.

TABLE I.—TEST-BEARING SPECIFICATIONS

Bearing:	
Bore, mm (in.)	35 (1.3780)
Outside diameter, mm (in.)	62 (2.4409)
Width, mm (in.)	14 (0.5512)
Cage:	
Diametral land clearance, mm (in.)	0.406 (0.016)
Diametral ball pocket clearance, mm (in.)	0.660 (0.026)
Material	4340 per AMS 6415 (silver plated)
Hardness	Rockwell C28-36
Balls:	
Number	16
Size (diameter), mm (in.)	7.14 (0.28)
Grade	10
Material	CEVM M-50 per AMS 6490
Hardness	Rockwell C60 (min.)
Race:	
Inner conformity, percent	54
Outer conformity, percent	52
Assembly:	
Internal radial clearance, mm (in.)	0.074 (0.0029)
Contact angle, deg	24

Lubricant

The oil used for the parametric studies was a neopentypolyol (tetra) ester. This type II oil is qualified to the MIL-L-23699 specifications as well as to the internal oil specifications of most major aircraft engine producers. The major properties of the oil are presented in table II.

Test Procedure

After warming the test machine by recirculating heated oil and calibrating the torque-measuring system, a thrust load of 667 newtons (150 lb) was applied and the lubricant flow rate was set at 1900 cm^3/min (0.50 gal/min). Outer ring cooling was not employed at this time. The shaft speed was then slowly brought up to a nominal 28 000 rpm. When bearing and test machine temperatures stabilized (after 20 to 25 min) the oil-inlet temperature, lubricant flow rate, and speed were set to the desired values. A test series was run by starting at the lowest speed, a nominal 48 000 rpm, and progressing through 65 000 and 72 000 rpm before changing the lubricant flow rate. At each speed and flow condition a separate test was run during which outer ring cooling oil flow was employed. Four lubricant flow rates to the bearing inner ring of 300 to 1900 cm^3/min (0.08 to 0.50 gal/min) were used.

After these test runs the resulting data were compared with those from a similar bearing run with jet lubrication, as described in reference 6. If it became apparent during the course of testing that a test condition would result in predictable distress of the test bearing or test rig, or generate a bearing temperature above 491 K (425° F), that test point was aborted or omitted.

Results and Discussion

Parametric tests were conducted in a high-speed bearing tester with a 35-millimeter-bore ball bearing having a single-outer-land-guided cage. Test results from this bearing with lubricant supplied through the inner ring are compared with those results obtained with a jet-lubricated bearing of identical dimensions and cage design as reported in reference 6.

Effect of Oil Flow Distribution Through Inner Ring on Bearing Temperature

The effect that the distribution of lubricant through the inner ring has on bearing temperature is shown in figures 4 and 5. The total flow was apportioned in two ways. In one way 50 percent of the oil flowed through the bearing inner ring to lubricate the bearing and 50 percent flowed axially only, for inner ring cooling. In the other way the distribution of oil flow was 25 percent for lubrication and 75 percent for cooling. Figures 4 and 5 also show the effect of outer ring cooling on bearing temperature, to be discussed later. Bearing temperature decreased with increased lubricant flow rate to the bearing for all conditions investigated. The flow distribution allowing the majority of the oil

TABLE II.—PROPERTIES OF TETRAESTER LUBRICANTS

Additives	Corrosion and oxidation inhibitors; and antiwear and antifoam additives
Kinematic viscosity, cS, at —	
311 K (100° F)	28.5
372 K (210° F)	5.22
477 K (400° F)	1.31
Flashpoint, K (°F)	533 (500)
Autogenous ignition temperature, K (°F)	694 (800)
Pour point, K (°F)	214 (−75)
Volatility (6.5 hr at 477 K (400° F)), wt%	3.2
Specific heat at 372 K (210° F), J/kg K (Btu/lb °F)	2140 (0.493)
Thermal conductivity at 477 K (400° F), J/m sec K (Btu/hr ft °F)	0.13 (0.075)
Specific gravity at 372 K (210° F)	0.931

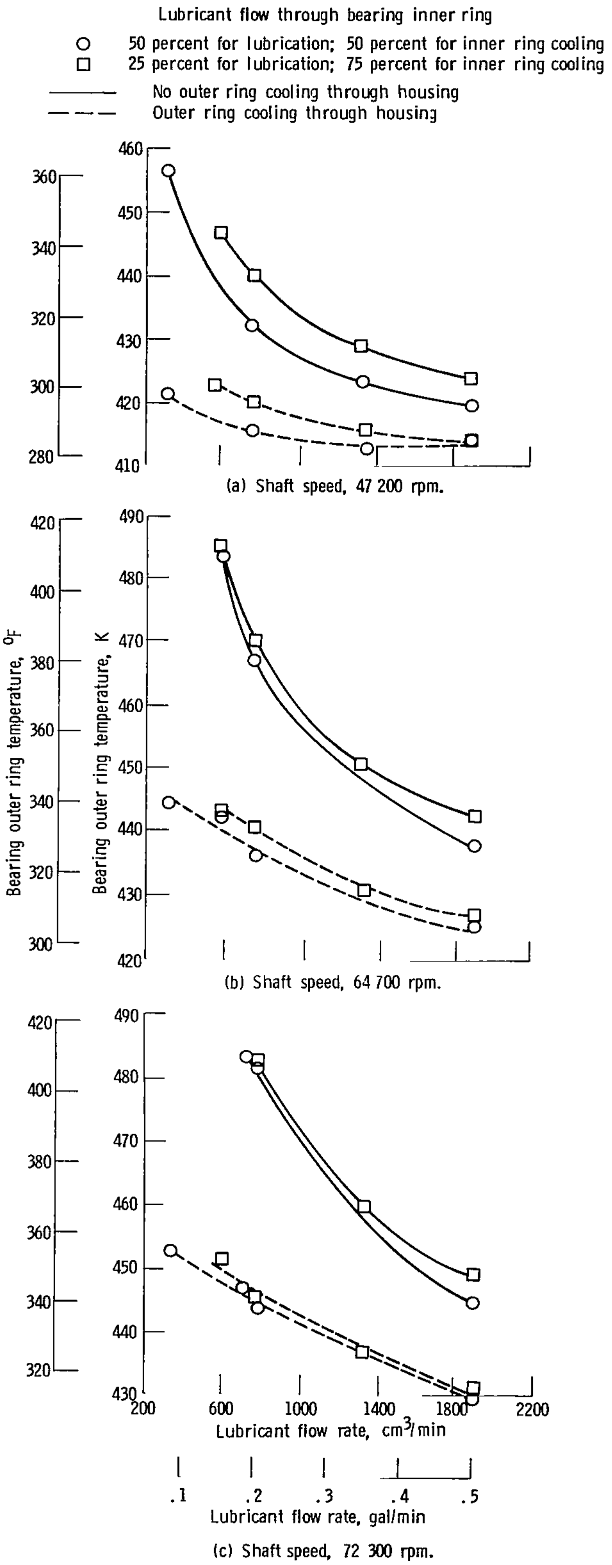


Figure 4. - Effect of lubricant flow rate on bearing outer ring temperature.

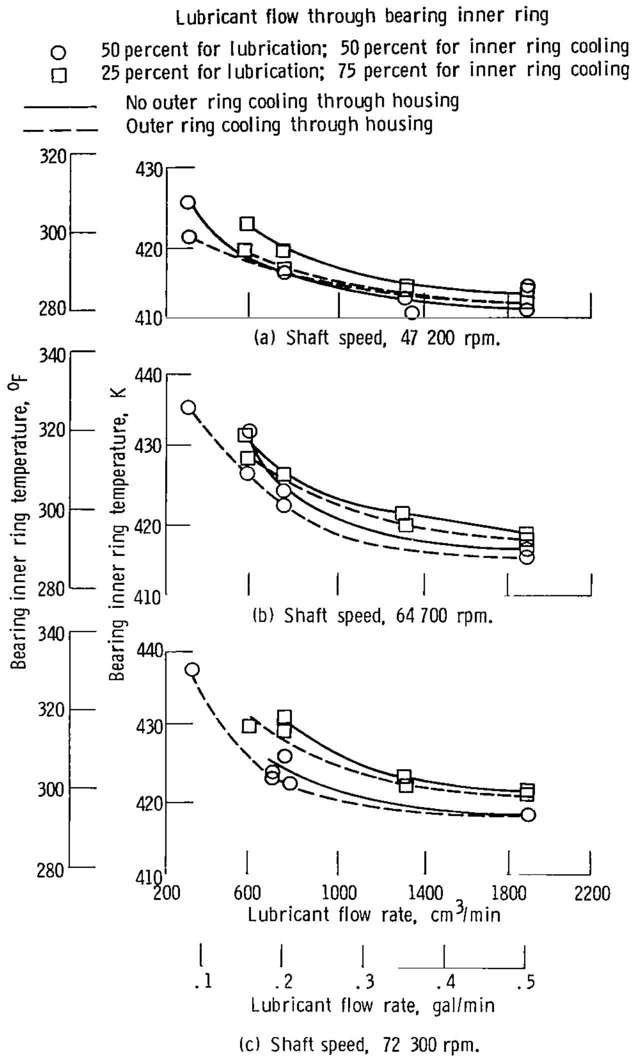


Figure 5. - Effect of lubricant flow rate on bearing inner ring temperature.

to cool the inner ring, namely the 25-75 percent, resulted in a higher outer ring temperature than the 50-50 percent distribution. This was true for all three shaft speeds (47 200, 64 700, and 72 300 rpm, fig. 4) although the effect was diminished at the two higher speeds. This higher outer ring temperature was the result of the decreased amount of oil flowing radially through the bearing. The added cooling flow through the axial grooves in the inner ring afforded by a flow distribution of 25-75 percent did not aid in cooling the inner ring. This is shown in figure 5, where the inner ring temperature is actually higher for a flow distribution of 25-75 percent than for a 50-50 percent distribution at speeds from 47 200 to 72 300 rpm. The cooling effect of the oil is less when it is channeled through axial grooves at the inner ring than when it is permitted to radially enter the bearing. The amount of cooling that can be accomplished with the axial oil flowpath in the inner ring grooves is greatly limited by the small surface area that this oil contacts.

The results of tests shown in figures 4 and 5 indicate that an oil flow distribution of 50–50 for lubrication and inner ring cooling is the more desirable of the two methods used here. Both inner and outer ring temperatures were lower at all speeds and lubricant flow rates when the bearing was tested under the conditions of 50–50 percent flow distribution.

Outer ring cooling in which the oil was maintained at a flow rate of 1700 cm^3/min (0.45 gal/min) at 394 K (250° F) oil-inlet temperature was employed in some tests. The results are also shown in figures 4 and 5. At the lowest shaft speed of 47 200 rpm (fig. 4(a)) outer ring cooling reduced the outer ring temperature by about 36 to 8 kelvins (65 to 14 deg F) as the total oil flow to the inner ring was increased from 300 to 1900 cm^3/min (0.08 to 0.50 gal/min). At the higher speeds, 64 700 and 72 300 rpm (figs. 4(b) and (c)), the outer ring temperature reduction was about 42 to 16 kelvins (75 to 28 deg F) as the total oil flow was increased from 580 to 1900 cm^3/min (0.15 to 0.50 gal/min). The magnitude of the reduction in outer ring temperature with outer ring cooling was approximately equal for the 50–50 percent and 25–75 percent total oil flow distributions.

Outer ring cooling had very little effect on the inner ring temperature (fig. 5) for either oil flow distribution pattern. The reduction of inner ring temperature varied from 0 to 6 kelvins (0 to 11 deg F) over the entire range of total oil flows, shaft speeds, and oil distribution patterns employed in these tests.

Comparison of Inner Ring and Jet Lubrication

The effect of speed on test bearing temperature for 50–50 and 25–75 percent total oil flow distribution patterns through the inner ring is shown in figure 6 and compared with that of the jet-lubricated bearing reported in reference 6. Both bearings were dimensionally identical and also had a single-outer-land-guided cage. Bearing temperature varied directly with speed and inversely with total oil flow. Outer ring temperatures were higher than inner ring temperatures for both lubrication methods used. Outer ring temperatures were lower and inner ring temperatures higher with a jet-lubricated bearing over the range of speeds and total oil flow rates investigated. The oil impinging on the inner ring from the jets was apparently slung off the ring so rapidly that only a minimal amount of inner ring cooling occurred, and thus inner ring temperatures were high. However, since this oil had not gained much heat from the inner ring, it impinged on the outer ring at a cooler temperature than did the oil in the inner-ring-lubricated bearings. Also, most of the total oil flow through the jets contacted the outer race to provide cooling, whereas a maximum of 50 or 25 percent of the total oil flowing through the inner ring grooves contacted the outer race when inner ring lubrication was used. For these reasons the outer ring of the jet-lubricated bearing was at a lower temperature than the outer ring of the inner-ring-lubricated bearing.

Bearing Power Loss

Two approaches were used to determine bearing power loss. In the first, outer ring torque was measured. In the second, heat rejected to the lubricant was determined.

Bearing power loss is dissipated in the form of heat rejected to the lubricant and surrounding environment by conduction, convection, and radiation. To obtain a measure of this heat rejection and thus power loss within the bearing, oil-inlet and -outlet temperatures were obtained for all conditions of lubricant flow. Total heat absorbed by the lubricant was obtained from the standard heat transfer equation

$$Q_T = MC_p(t_{out} - t_{in}) \tag{1}$$

where

Q_T total heat transfer rate to the lubricant, J/min (Btu/min)

M mass flow rate, kg/min (lb/min)

C_p specific heat, J/kg K (Btu/lb °F)

t_{out} oil-outlet temperature, K (°F)

t_{in} oil-inlet temperature, K (°F)

The power loss in the bearing obtained from torque measurements taken with a strain gage attached to the bearing housing is shown in figure 7. The measured torque for the jet-lubricated and inner-ring-lubricated bearings was 0.011 to 0.033 newton meter (1.00 to 2.90 lb in.) over the lubricant flow range and speeds tested. Power loss increased with speed at each flow rate investigated. The greatest increase in power loss was about 1.27 kilowatts (1.7 hp) at a total oil flow rate of 1900 cm^3/min (0.50 gal/min) over a speed range of about 47 000 to 72 000 rpm (fig. 7(d)). There was very little difference in power loss for the two methods of lubrication

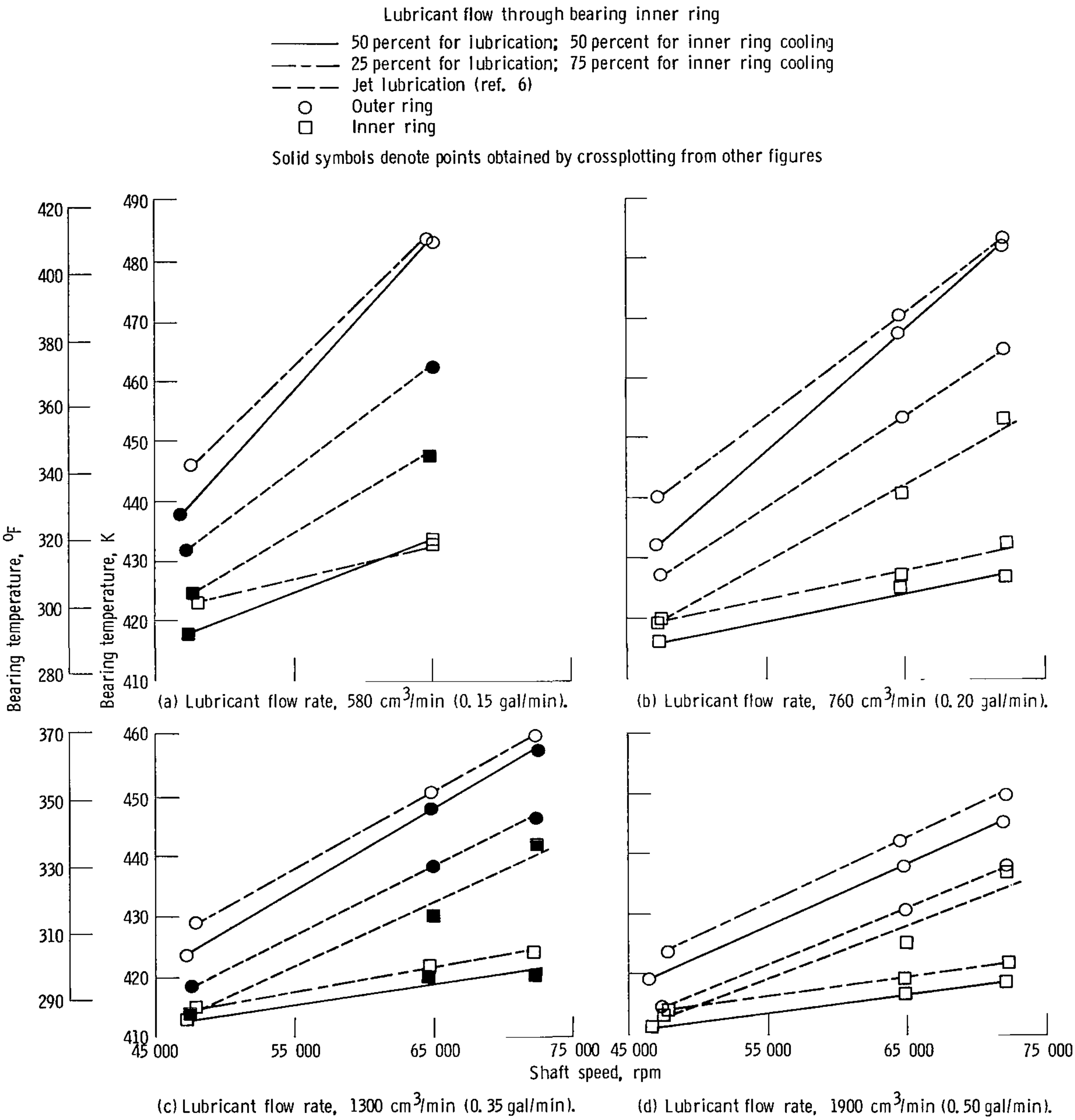


Figure 6. - Effect of shaft speed on test-bearing temperature for two different lubricant supply systems. No outer ring cooling through housing.

employed over the range of oil flows and shaft speeds tested except for the jet-lubricated bearings at the two highest flow rates (figs. 7(c) and (d)). The jet-lubricated bearing at oil flow rates of 1300 and 1900 cm^3/min (0.35 and 0.50 gal/min) showed slightly less power loss than the inner-ring-lubricated bearing.

The results of heat transfer calculations for a single-outer-land-guided-cage bearing using two different methods of lubrication are compared in figure 8. For convenience, values for heat transfer were converted from joules per minute to kilowatts. These data are almost identical to those in figure 7 for power loss from torque measurements. The values of power rejected to the lubricant for both methods of lubrication are similar except at the two higher lubricant flow rates (figs. 8(c) and (d)). At

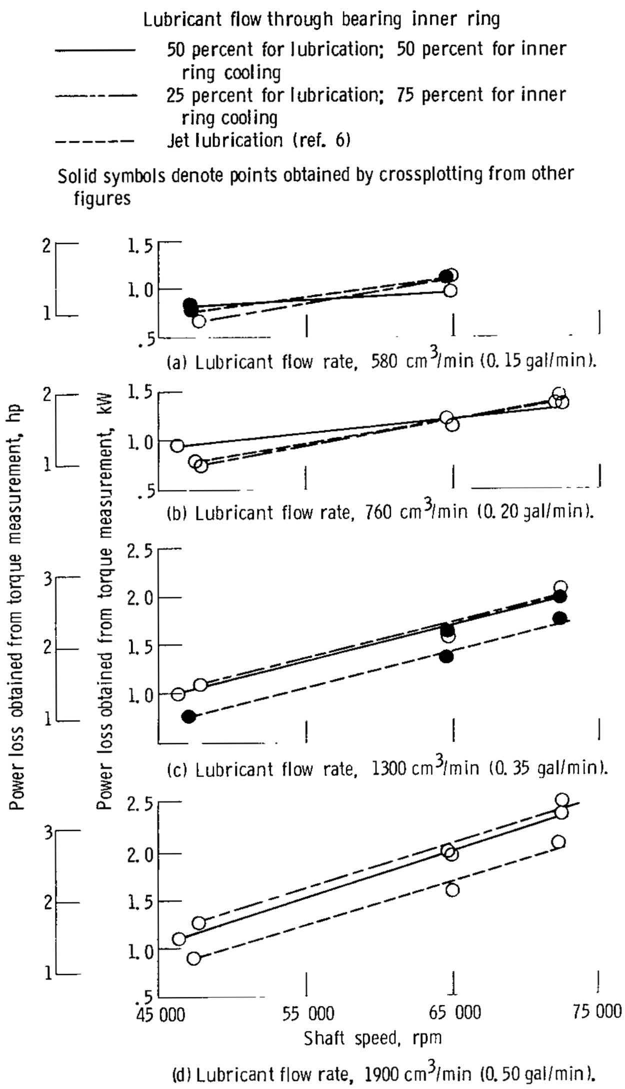


Figure 7. - Effect of shaft speed on power loss obtained from torque measurements for two different lubricant supply systems. No outer ring cooling through housing.

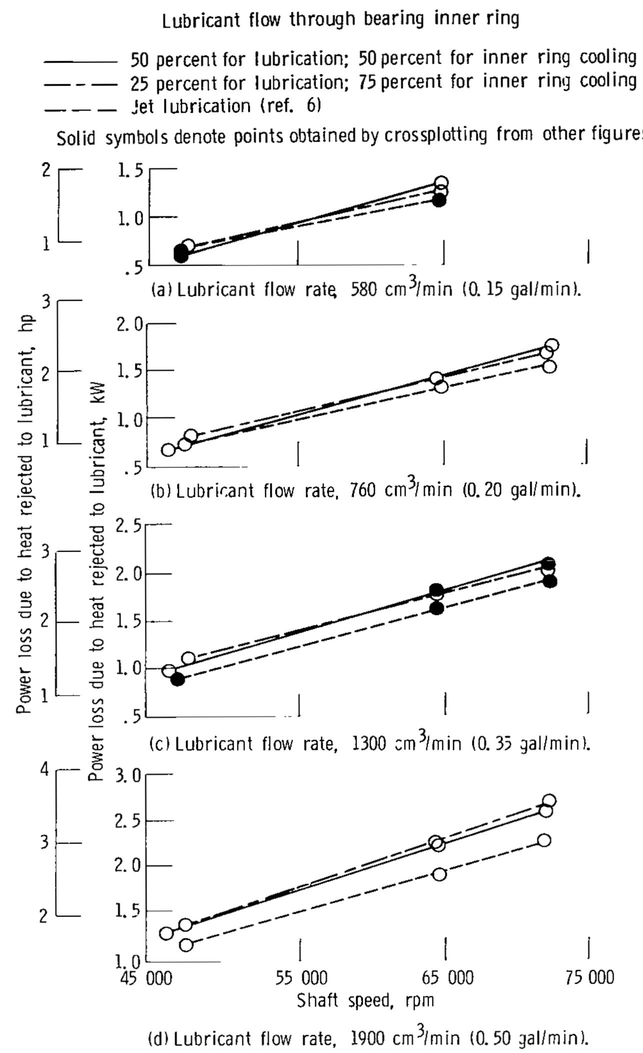


Figure 8. - Effect of shaft speed on power loss due to heat rejected to lubricant for two different lubricant supply systems. No outer ring cooling through housing.

these flow rates the jet-lubricated bearing had the lowest value of power rejected to the lubricant. Maximum power loss was 2.8 kilowatts (3.7 hp) at 72 300 rpm (fig. 8(d)) with a total oil flow distribution pattern of 25-75 percent through the bearing.

Effect of Lubricant Flow on Cage Slip

To determine percent cage slip, the epicyclic cage speed C_{epi} at the various test shaft speeds was obtained from a computer program called SHABERTH (ref. 8), which considers centrifugal force effects on contact angle. Elastic-contact forces are considered in a race-control type of solution. Thermal and lubricant effects are not considered in this computer solution of epicyclic cage speed. A fit analysis was not included in the SHABERTH computations for this report or for references 6 and 9. However, subsequent calculations at the highest speed (72 300 rpm) and a thrust load of 667 N (150 lb), considering all the centrifugal growth effects on the inner ring, showed the epicyclic speed to change only from 33 000 rpm to 32 720 rpm. This resulted in the calculated slip changing from 7.0 to 6.2 percent. The calculated epicyclic cage speeds were combined with the measured experimental cage speed C_{exp} to

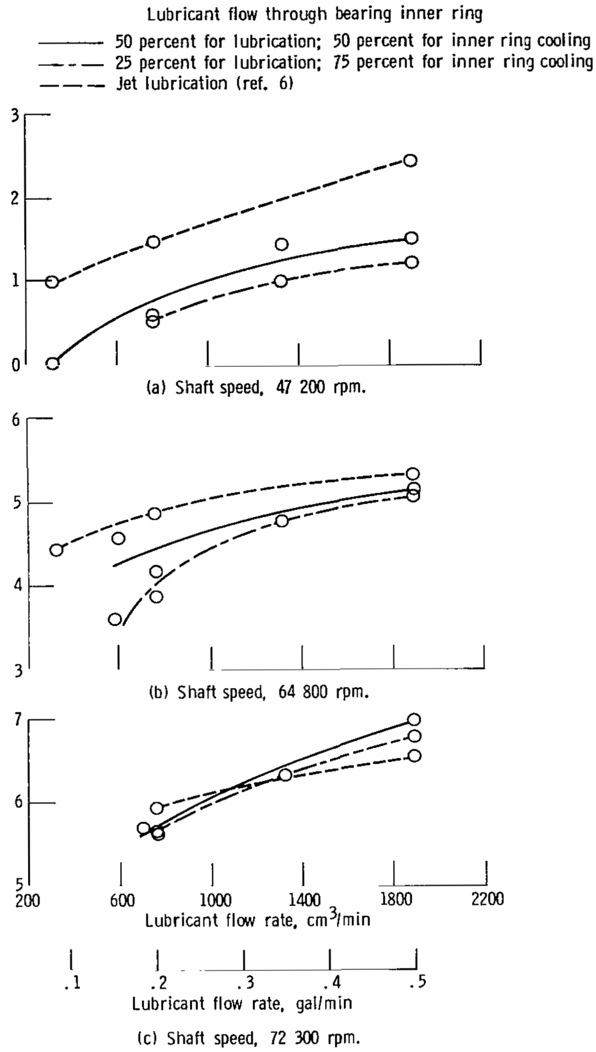


Figure 9. - Effect of lubricant flow rate on cage slip for two different lubricant supply systems. No outer ring cooling through housing.

obtain percent cage slip as follows:

$$\text{Percent cage slip} = (1 - C_{exp}/C_{epi})(100) \qquad (2)$$

The effect of lubricant flow rate on percent cage slip for a single-outer-land-guided-cage bearing using two different methods of lubrication is shown in figure 9. For the three speeds (47 200, 64 800, and 72 300 rpm) and the flow rates tested the percent cage slip was minimal for each method of lubrication. The small increase in slip with increasing flow rate is primarily due to additional drag on the balls. The jet-lubricated bearing showed a higher percent slip than the two inner-ring-lubricated bearings at speeds of 47 200 and 64 800 rpm (figs. 9(a) and (b)). However, at the maximum speed of 72 300 rpm (fig. 9(c)) the percent cage slip for all three bearings was essentially equal, with the jet-lubricated bearing showing a slightly lower percent cage slip at lubricant flow rates above approximately 1100 cm^3/min (0.30 gal/min). As the speed was increased, the centrifugal force generated caused more oil from the jets to be slung off the bearing, thus allowing less to enter as a lubricant for the rolling elements. With less oil present, less plowing of the balls and cage occurred, and this resulted in a lower rate of increase in cage slip than for the inner-ring-lubricated bearings. The maximum percent cage slip of 7.0 occurred at 72 300

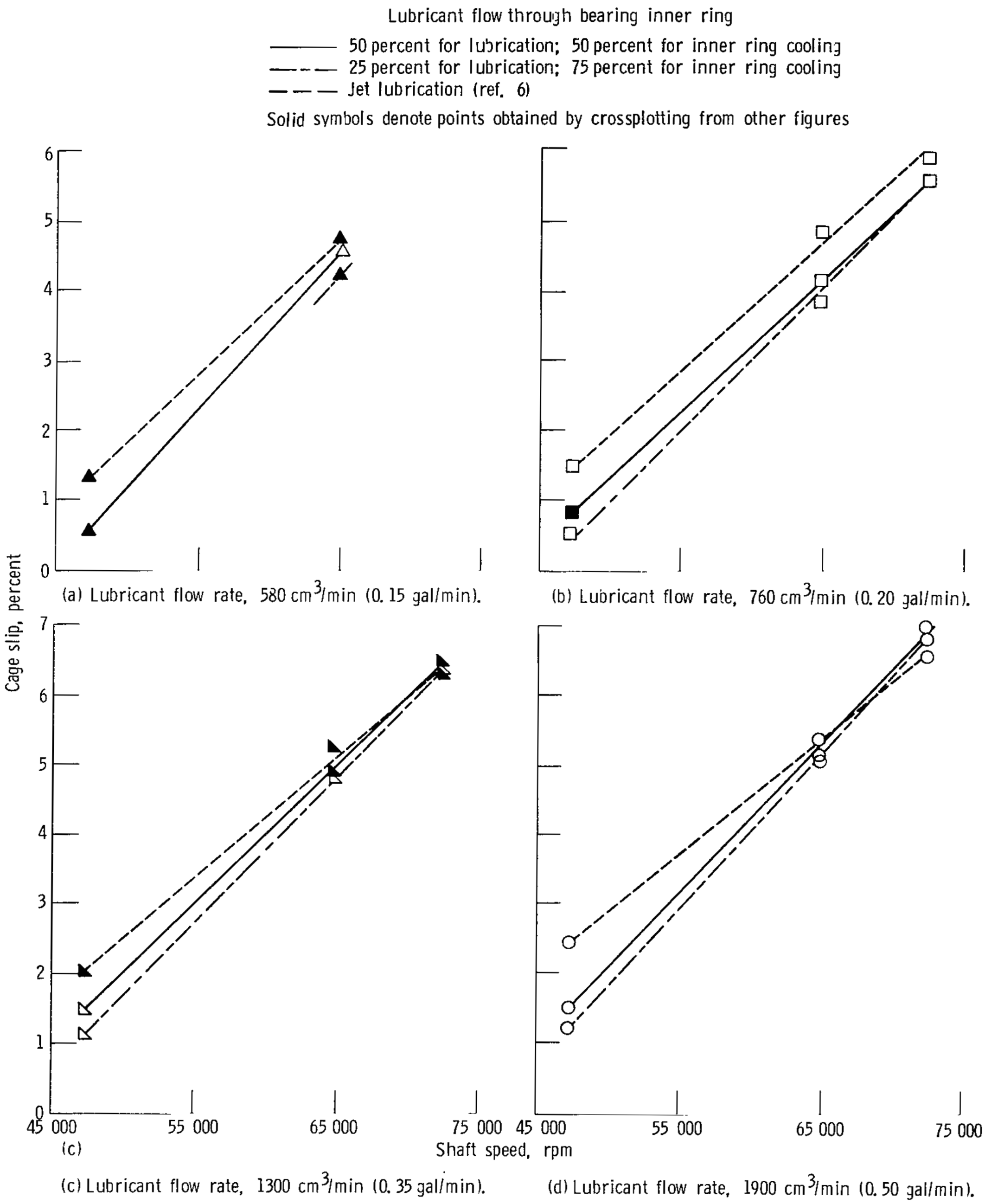


Figure 10. - Effect of shaft speed on cage slip for two different lubricant supply systems. No outer ring cooling through housing.

rpm at a total oil flow rate of 1900 cm^3/min (0.50 gal/min) (fig. 9(c)) with a total oil flow distribution pattern of 50–50 percent through the inner ring.

Figure 10 shows that percent cage slip increases with speed at about the same rate for each method of lubrication tested and independently of the lubricant flow rate, especially for the inner-ring-lubricated bearings. Increased percent cage slip with increased shaft speed may be partially due to centrifugal forces decreasing the ball load and thus the traction at the inner race contact. The jet-lubricated bearing showed a higher percent cage slip than the inner-ring-lubricated bearings at the lower flow rates of 580 and 760 cm^3/min (0.15 and 0.20 gal/min) (figs. 10(a) and (b)) over the entire speed range tested. However, as the flow rate was increased to 1300 and 1900 cm^3/min (0.35 and 0.50 gal/min) (figs. 10(c) and (d)), the rate of increase in cage slip of the jet-lubricated bearing was less than that of the inner-ring-lubricated bearings at the higher speeds (above about 65 000 rpm). This resulted in a cage slip value equal to or less than the percent cage slip of the inner-ring-lubricated bearings. Although the flow rate was high, the jet-lubricated bearing had a large percentage of its oil slung off the bearing at the higher speeds, and the remaining amount of oil flowing through the bearing approximated the oil flowing through the inner-ring-lubricated bearings at these test conditions. Since there was less oil flowing through the jet-lubricated bearing under these conditions, the percent cage slip decreased to a point approximately equal to that of the inner-ring-lubricated bearings as shown in figures 10(c) and (d). The maximum cage slip of 7.0 percent occurred at 72 300 rpm at a total flow rate of 1900 cm^3/min (0.50 gal/min) with an oil distribution pattern of 50-50 percent (fig. 10(d)).

Summary of Results

Parametric tests were conducted in a high-speed bearing tester on a 35-millimeter-bore ball bearing with a single-outer-land-guided cage. Axial grooves and radial holes machined in the inner ring of the bearing permitted lubrication through the bearing inner ring. Test parameters were a thrust load of 667 newtons (150 lb), shaft speeds of a nominal 48 000 to 72 000 rpm, and an oil-inlet temperature of 394 K (250° F). Total oil flow to the bearing was 300 to 1900 cm^3/min (0.08 to 0.50 gal/min). An outer ring cooling oil flow rate maintained at 1700 cm^3/min (0.45 gal/min) at 394 K (250° F) oil-inlet temperature was used in some tests. The distribution of the total oil supplied to the inner ring was 50 percent for lubrication of the rolling elements and 50 percent for inner ring cooling in some tests. In other tests the distribution was 25 percent lubrication and 75 percent cooling. Test results for a bearing with oil supplied through the inner ring were compared with those for a jet-lubricated bearing with identical dimensions and cage design as reported in a previous investigation. The following major results were obtained:

1. A 35-millimeter-bore, angular-contact ball bearing with inner ring lubrication was successfully operated to 2.5 million DN for both the 50-50 percent and 25-75 percent oil flow distributions.

2. Cooler bearing operation was experienced with a total oil flow distribution pattern of 50 percent lubrication and 50 percent inner ring cooling than with a 25-75 percent distribution.

3. Jet lubrication data from a previous study showed lower outer ring temperatures and higher inner ring temperatures than the results for inner-ring-lubricated bearings from this investigation.

4. Outer ring cooling of both jet-lubricated and inner-ring-lubricated bearings resulted in a substantial decrease in outer ring temperature but had a minimal effect on inner ring temperature.

5. Maximum power loss of 2.8 kilowatts (3.7 hp) occurred at 72 300 rpm with a total oil flow distribution pattern of 25-75 percent at a total oil flow rate of 1900 cm^3/min (0.50 gal/min).

6. The greatest increase in power loss was about 1.27 kilowatts (1.7 hp) at a total oil flow rate of 1900 cm^3/min (0.50 gal/min) over a speed range of about 47 200 to 72 300 rpm and was common to both methods of inner-ring lubrication employed.

7. Maximum percent cage slip of 7.0 occurred at 72 300 rpm at a total oil flow rate of 1900 cm^3/min (0.50 gal/min) with a total oil flow distribution pattern of 50-50 percent. The increase in percent cage slip with lubricant flow rate was minimal for each flow distribution pattern used; however, cage slip increased significantly with speed.

Lewis Research Center
National Aeronautics and Space Administration
Cleveland, Ohio, September 17, 1980

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1. Report No. NASA TP-1808	2. Government Accession No.	3. Recipient's Catalog No.
4. Title and Subtitle PERFORMANCE OF JET- AND INNER-RING-LUBRICATED 35-MILLIMETER-BORE BALL BEARINGS OPERATING TO 2.5 MILLION DN		5. Report Date February 1981
		6. Performing Organization Code 505-32-42
7. Author(s) Fredrick T. Schuller and Hans R. Signer		8. Performing Organization Report No. E-515
		10. Work Unit No.
9. Performing Organization Name and Address National Aeronautics and Space Administration Lewis Research Center Cleveland, Ohio 44135		11. Contract or Grant No.
		13. Type of Report and Period Covered Technical Paper
12. Sponsoring Agency Name and Address National Aeronautics and Space Administration Washington, D.C. 20546		14. Sponsoring Agency Code

15. Supplementary Notes

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16. Abstract

Parametric tests were conducted with a 35-millimeter-bore, angular-contact ball bearing having a single-outer-land-guided cage. Lubrication was achieved by flowing oil through axial grooves and radial holes machined in the inner ring of the bearing. Test conditions were a thrust load of 667 N (150 lb), shaft speeds from 48 000 to 72 000 rpm, and an oil-inlet temperature of 394 K (250° F). Outer ring cooling was used in some tests. Data from tests where the distribution of the total oil supplied to the inner ring was 50 percent for bearing lubrication and 50 percent for bearing inner ring cooling were compared with those where the distribution pattern was 25 percent lubrication and 75 percent cooling. Results were also compared with those of a previously run jet-lubricated bearing with identical dimensions and cage design. Successful operation was experienced with both the 50-50 and 25-75 percent flow distribution patterns to 2.5 million DN. The 50-50 percent flow pattern provided the cooler bearing operation of the two inner-ring-lubricated bearings. The jet-lubricated bearing had lower outer ring and higher inner ring temperatures than the inner-ring-lubricated bearings. Maximum power loss of 2.8 kW (3.7 hp) was experienced with the 25-75 percent flow distribution, and maximum percent cage slip of 7.0 occurred at 72 300 rpm with the 50-50 percent flow distribution.

17. Key Words (Suggested by Author(s))	18. Distribution Statement
High-speed bearings; Rolling-element bearings; Bearings; Ball bearings; Bearing lubrication; Inner-ring-fed ball bearings	Unclassified - unlimited STAR Category 37

19. Security Classif. (of this report)	20. Security Classif. (of this page)	21. No. of Pages	22. Price*
Unclassified	Unclassified	14	A02

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