Lubrication of 35-Millimeter-Bore Ball Bearings of Several Designs to 2.5 Million DN

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Parametric tests were conducted in a high-speed, high-temperature bearing tester with a 35-mm-bore, angular-contact ball bearing with either a single- or a double-outer-land-guided cage. The bearings were either lubricated by jets (using a jet velocity of 20 m/sec (66 ft/sec)) or lubrication was achieved by flowing oil through axial grooves and radial holes machined in the inner ring of the bearing.

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. For 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 the distribution was 25 percent lubrication and 75 percent cooling.

All tests were conducted at a radial load of 222 N (50 lb) and/or 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). Lubricant flow rates to the bearing ranged from 300 to 1900 cm^3/min (0.08 to 0.50 gal/min). The lubricant was neopentylpolyol (tetra)ester meeting the MIL–L–23699 specifications.

Successful operation of the bearings with jet lubrication was accomplished up to 2.5 million DN. The jet-lubricated bearing with a double-outer-land-guided cage always had a higher power loss and generated higher temperatures than the bearing with a single-outer-land-guided cage over the range of shaft speeds and lubricant flow rates used.

Percent cage slip for a double-outer-land-guided-cage-bearing ranged from 1.5 to 2.7 times that for a single-outer-land-guided-cage bearing.

Successful operation of the bearing with through-the-inner-ring lubrication was also accomplished to 2.5 million DN for both oil distribution patterns. Cooler bearing operation was experienced with a total oil distribution of 50–50 percent than with 25–75 percent.

Outer-ring cooling for both oil-flow distribution and for the 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 kW (3.7 hp) occurred at 72 300 rpm with an inner-ring oil-flow distribution of 25–75 percent at a total oil-flow rate of 1900 cm^3/min (0.50 gal/min).

For the inner-ring-lubricated bearings, 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 of 50-50 percent. The increase in percent cage slip with lubricant flow rate was minimal for each lubrication method used; however, cage slip increased significantly with speed.

Introduction

Small, advanced turbine engines with total airflows of 0.5 to 4.6 kg/sec (1 to 10 lb/sec) require bearings that can operate in the speed range 2.5 million DN (product of the bearing bore in mm 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.

Large-bore ball and roller bearings have been successfully tested at speeds to 3.0 million DN (refs. 1 to 4). In these tests, however, the lubricant was fed to the bearings through radial holes in the inner ring. Because of the dimensional limitations of the inner ring in smaller bore bearings, the

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fabrication of radial holes and axial grooves for lubricant passages through the inner ring can become complex and cost restrictive. In these circumstances jet lubrication is the more practical method of bearing lubrication.

However, there is a limiting DN value above which jet lubrication is no longer adequate for reliable bearing operation. Centrifugal forces prevent the oil jet from properly lubricating and cooling the rolling elements and cage. This results in bearing thermal instability and/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 5 for a 30-mm-bore, deep-groove ball bearing with an outer-land-guided cage.

One of the principal bearing elements that affects satisfactory high-speed operation is the bearing cage. As indicated in reference 5, bearing wear and ultimate failure at a high DN value has occurred on the land surface of the cage, and cage lubrication was the principal factor that brought about the failure. Consequently, the cage design is expected to greatly affect the limiting speed. For example, in reference 5, with a jet-lubricated bearing, an outer-ring-land-guided cage limited the DN value to 2.85×10^6, where failure occurred; whereas an inner-ring-land-guided cage limited the DN value to 1.65×10^6. For satisfactory high-speed operation of a small-bore, jet-lubricated bearing, therefore, an outer-ring-land-guided cage is usually recommended. The outer-ring-land surfaces are more efficiently lubricated by means of centrifugal oil flow effect, and they are generally cooler than the corresponding surfaces of an inner-ring controlled cage (ref. 6). Several cage designs for a 75-mm jet-lubricated roller bearing have been tested, and the best overall performance was reported in reference 7 for bearings with outer- ring- land-guided cages.

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 was supplied to the bearing through grooves in the bore and then through radial holes from the grooves to the rolling elements of the bearing. Effective lubrication and cooling in a very severe centrifugal force field are the principal problems in achieving successful operation of small-bore bearings at ultrahigh speeds. Inner-ring lubrication might be a solution, although the limited space available for machining grooves and radial holes can be a problem.

The objectives of this study were to determine the parametric effects of cage design, shaft speed, lubricant flow rate, and method of lubrication on the operation of a 35-mm-bore angular contact bearing. Test conditions included a radial load of 222 N (50 lb) and/or a thrust load of 667 N (150 lb). Nominal shaft speeds were from 48 000 to 72 000 rpm. Jet lubrication was accomplished by dual jets at flow rates from 303 to 1894 cm^3/min (0.08 to 0.50 gal/min) at a calculated jet velocity of 20 m/sec (66 ft/sec) with an oil-in temperature of 394 K (250° F). Inner-ring lubrication was accomplished by a modification of the inner ring, where radial holes are machined in the axial grooves of the bearing bore.

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.

The bearing had a nominal unmounted contact angle of 24° and either a single- or double-outer-land-guided cage. Provisions were also made for outer-ring cooling. 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 in all tests reported herein was a neopentylpolyol (tetra)ester that meets the MIL–L–23699 specifications.

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

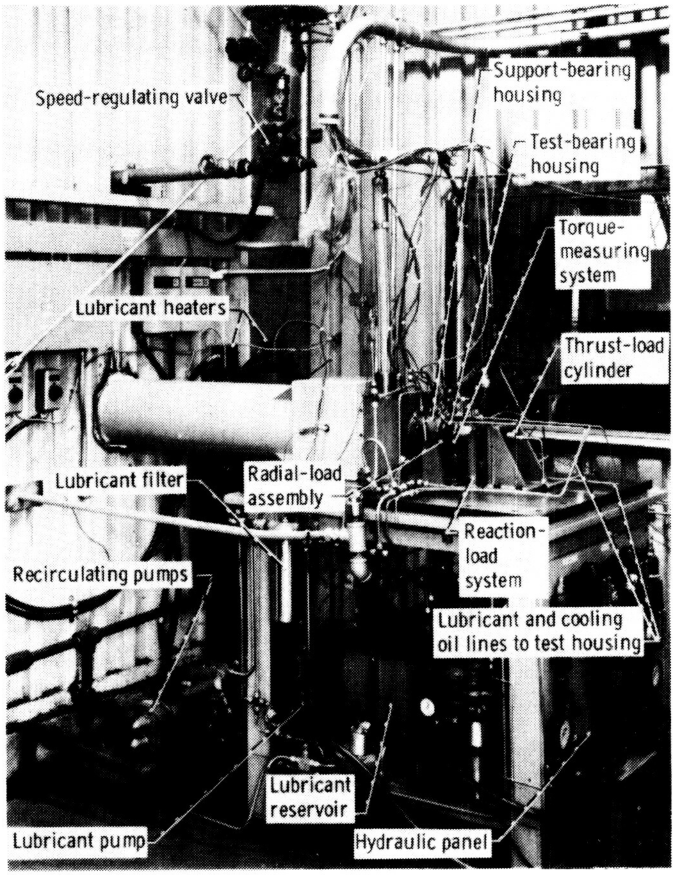


Figure 1. - High-speed, small-bore-bearing test machine.

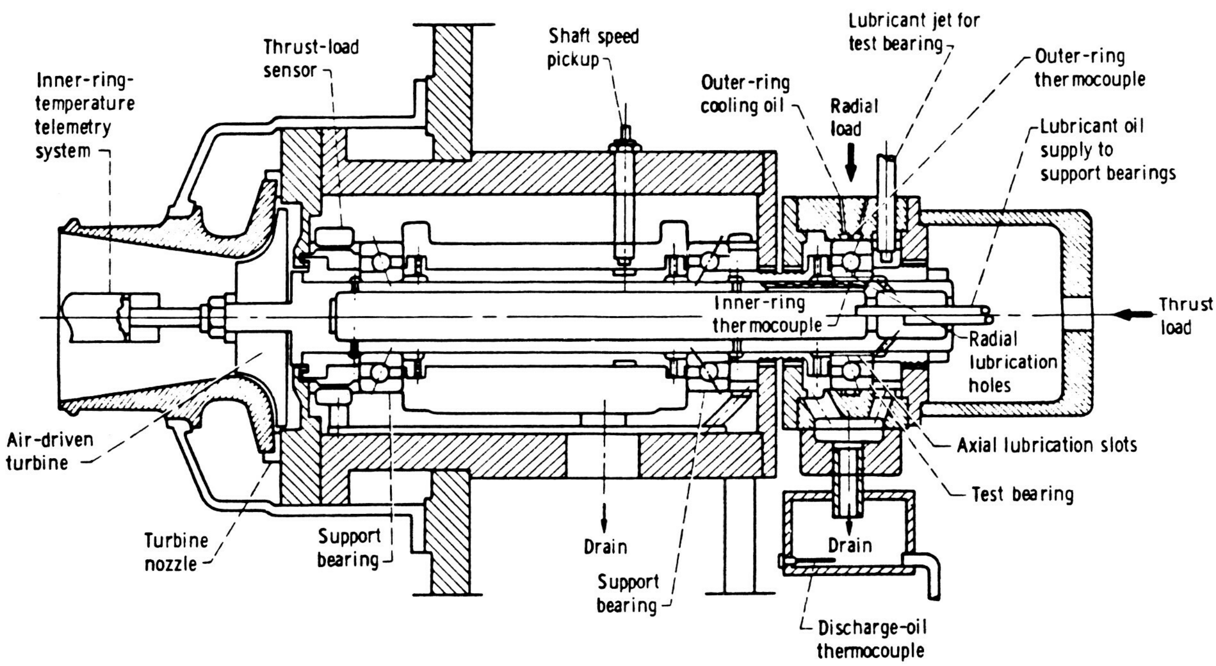


Figure 2. - Schematic of high-speed, small-bore-bearing test machine.

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 either by jets or through the inner ring. The jet outlets on the nonloaded side of the inner ring were located approximately 3.0 mm (0.12 in.) from the face of the bearing and were aimed at the inner raceway. In separate tests, not reported herein, it was determined that a 20-m/sec (66-ft/sec) jet velocity provided the most efficient lubrication of the test bearing; that velocity was used in all the tests reported. (Ref. 5 reports a similar result with a 20-m/sec (66-ft/sec) jet velocity compared with other velocities investigated.) When inner-ring-injected lubrication was used, 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 mm (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 8 and 9.

Test Bearings

The test bearings were ABEC7 grade, 35-mm-bore, angular-contact ball bearings with a double- or single-outer-land-guided cage (fig. 3). The outside diameter of both cages was a nominal 52.68 mm (2.074 in.). The effective land area of the double-outer-land-guided cage bearing was approximately three times that of the single-outer-land-guided cage bearing. The double-outer-land-guided cage weighed 16 percent more than the single-outer-land-guided cage. The inner rings of both bearings were geometrically and dimensionally the same. The land diameter on the oil-inlet side of the outer ring of the single-land-guided-cage bearing was 13.21 mm (0.52 in.) smaller than that on the outlet side. The land diameters of both sides of the double land-guided-cage bearing outer ring were of equal dimension.

The bearings contained 16 balls, each with a nominal diameter of 7.14 mm (0.281 in.). One bearing design (fig. 3(c)) 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-mm (0.030-in.) diameter holes (one in every other axial groove), radiating from the bearing bore, formed a flow path 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 and 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 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 C 62 at room temperature. The cage was made from AISI 4340 steel (AMS 6415) heat treated to a hardness of Rockwell C 28 to 36 and completely plated with 0.0203- to 0.0381-mm (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.

Lubricant

The oil used for the parametric studies was a neopentylpolyol (tetra)ester. This type II oil is qualified to the MIL-L-23699 specifications. 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 test load of 222 N (50 lb) radial and/or a test load of 667 N (150 lb) thrust load 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

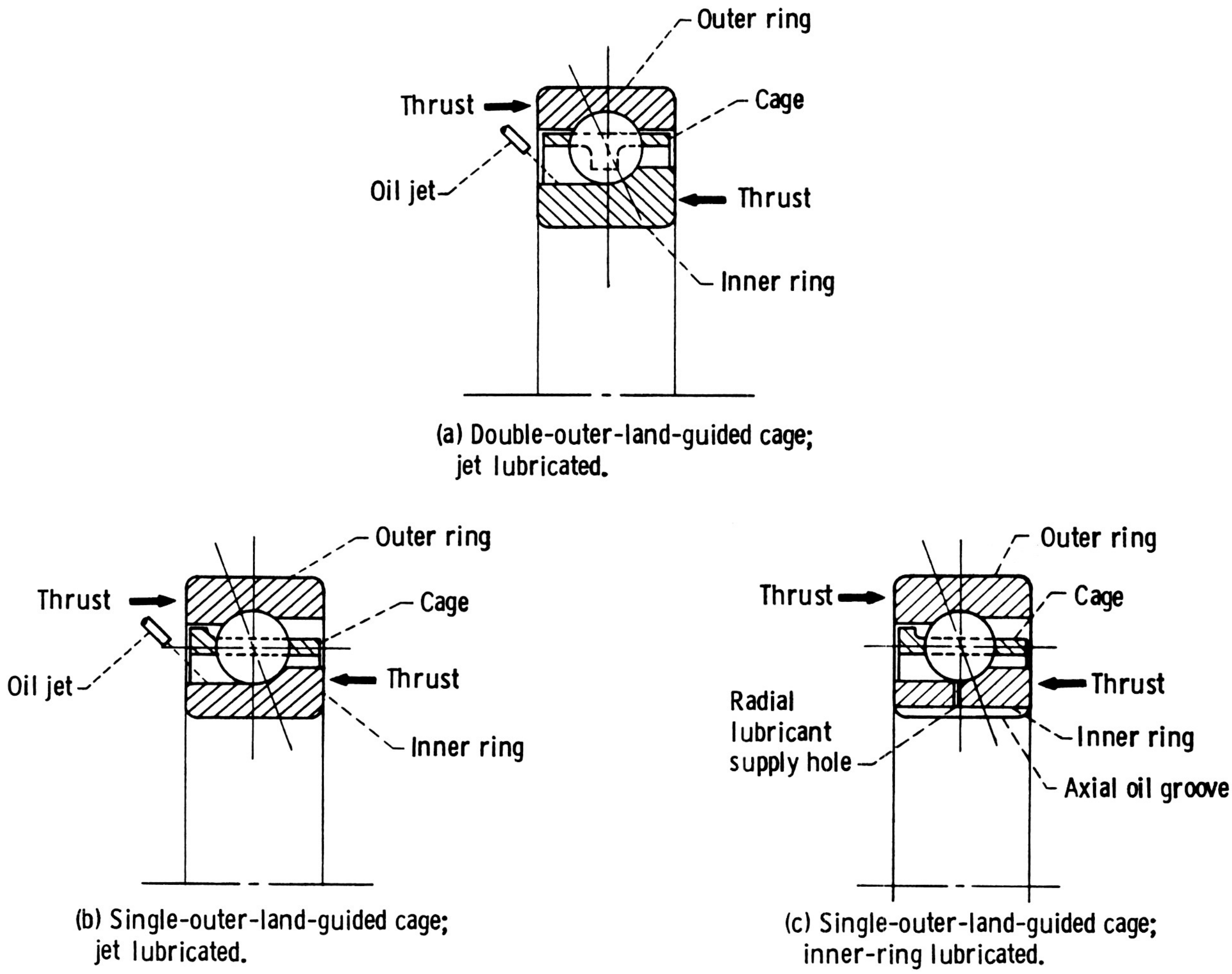


Figure 3. - Angular-contact ball bearing.

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. 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. In some tests a separate run was made, during which outer-ring cooling oil flow was employed.

If it became apparent during the course of testing that a test condition would result in distress of the test bearing or test rig or generate a bearing temperature above 491 K (425° F), that test was terminated.

Results and Discussion

Comparison of Single- and Double-Outer-Land Riding Cage Bearings

In the cage design portion of this study parametric tests were conducted in the high-speed bearing tester with jet-lubricated 35-mm-bore ball bearings having either a single- or a double-outer-land-guided cage. Other than cage design and the inside diameter of the outer ring (figs. 3(a) and (b)), the two bearings were identical.

Effect of cage design on bearing temperature. – The effect of lubricant flow rate on bearing temperature at three different speeds is shown in figure 4. Bearing temperatures for both the single-

TABLE I. – TEST BEARING SPECIFICATIONS

Bearing dimensions, mm (in.):	
Bore	35 (1.3780)
Outside diameter	62 (2.4409)
Width	14 (0.5512)
Cage specifications:	
Diametral land clearance, mm (in.)	0.406 (0.016)
Diametral ball-pocket clearance, mm (in.)	0.660 (0.026)
Material	[a]AISI 4340, silver plated
Rockwell C hardness	28–36
Bearing ball specifications:	
Number	16
Diameter, mm (in.)	7.14 (0.28)
Grade	10
Material	[b]CEVM M-50
Rockwell C hardness (minimum)	60
Race conformity, percent:	
Inner	54
Outer	52
Assembly:	
Internal radial clearance, mm (in.)	0.074 (0.0029)
Contact angle, deg	24

[a]AMS 6415.

[b]AMS 6490.

TABLE II. – PROPERTIES OF TETRAESTER LUBRICANTS

Additives	Antiwear, corrosion and oxidation inhibitors, and antifoam
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)
Pourpoint, 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/hr ft °F)	2140 (0.493)
Thermal conductivity at 372 K (210° F), J/m sec K (Btu/hr ft °F)	0.15 (0.088)
Specific gravity at 372 K (210° F)	0.931

and the double-outer-land-guided cage decrease with an increase in lubricant flow rate for each test speed.

The double-outer-land-guided cage bearing generated the higher bearing temperature at each speed tested. Figure 4 shows that the temperature differential between the two bearings of different cage design increases with speed and reaches 25 K (45° F) at 72 250 rpm at a flow rate of 1894 cm^3/min (0.50 gal/min). This indicates that the double-outer-land-guided cage bearing becomes less desirable at high speeds.

The elevated temperatures of the bearing with the double-outer-land-guided cage are partially due to the heat generated by the shearing of oil over an area about three times that of the single-outer-land-guided cage bearing. Another disadvantage of the double-outer-land-guided cage is that the oil entering and lubricating the rotating members of the bearing has a tendency to become trapped by the extra land, adding heat due to excessive churning within the bearing. The single land, in contrast,

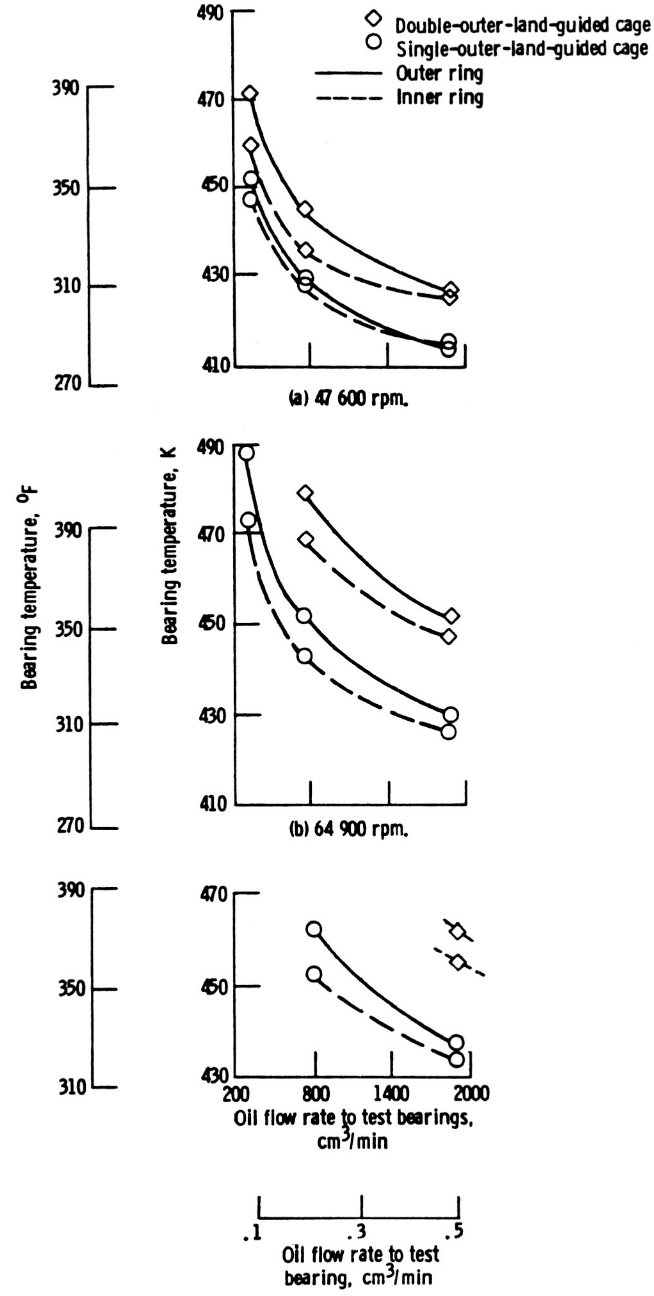


Figure 4. - Effect of oil flow on test bearing temperature for two jet-lubricated bearing configurations; no outer-ring cooling. Combined load: thrust, 667 N (150 lb); radial, 222 N (50 lb).

allows the free flow of lubricant into the bearing and a less restrictive exit of oil, reducing churning and subsequent internal heat generation in the bearing.

Effect of cage design on bearing power loss. – Two approaches were used to determine bearing power loss. In the first outer-ring torque was measured. In the second the heat rejected to the lubricant was determined.

Bearing power loss is dissipated in the form of heat rejected to the lubricant and transferred by conduction, convection, and radiation to the surrounding environment. 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.

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

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 of the double- and single-outer-land-guided cage jet-lubricated bearings is shown in figure 5. Power loss obtained from torque readings and power loss determined from heat rejected to the oil are plotted.

Figure 5 shows that power loss increases with speed and lubricant flow rate for both the single- and double-outer-land-guided cage bearing configurations. However, comparison of power losses shows that the double-outer-land-guided cage bearing has a decided disadvantage compared with the single-outer-land-guided-cage design. The higher power loss of the double-outer-land-guided cage bearing is due, in part, to the excessive oil churning in this design bearing and, in part, to the drag resulting from a total land area three times that of the single-land design. Figure 5 shows that a very high lubricant flow rate (1894 cm^3/min (0.50 gal/min)) can be undesirable because it increases the power loss in the bearing (see fig. 5(a)).

Figure 5 also compares power losses obtained from torque readings with those from heat rejected to the lubricant. No outer-ring cooling was used in these particular tests. Power losses determined from heat rejection to the lubricant were lower than those obtained from torque readings over the speed range tested. Two reasons for this are (1) that the heat dissipated by conduction, convection, and radiation to the surrounding environment is not accounted for and (2) that locating thermocouples in ideal positions for a most accurate reading of oil-in and oil-out temperatures is difficult.

The results from both methods of obtaining bearing power loss (fig. 5) are in agreement except for magnitude.

Effect of lubricant flow rate 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. 10), which considers centrifugal force effects on contact angle. Elastic-contact forces are considered in a race-control type of solution. Thermal and lubricant effects were not considered in this computer solution of epicyclic cage speed. Also, a fit analysis was not included in the Shaberth computations for this report.

However, a subsequent calculation at the highest speed (72 300 rpm), at a thrust load of 667 N (150 lb), and for all the centrifugal growth effects on the inner ring showed the epicyclic speed to change only from 33 000 to 32 720 rpm. This resulted in the calculated slip changing from 7.0 to 6.2 percent for that point. Therefore, all the values of C_{epi} were not recalculated. The calculated epicyclic cage speeds were combined with the measured experimental cage speed C_{exp} to obtain percent cage slip as follows:

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

Figure 6 shows that percent cage slip increases with speed at about the same rate for each of the three lubricant flow rates tested. The double-outer-land-guided cage bearing showed a higher percent cage slip than the single-outer-land-guided cage at all speeds and lubricant flow rates. It also showed a higher change of cage slip with increased flow.

For all speeds and flow rates tested, the percent cage slip ranged from 1.0 to 10.2 percent for both jet-lubricated bearing configurations. The small increase in slip with flow rate is primarily due to additional drag on the balls and cage. Percent cage slip for a double-outer-land-guided cage bearing ranged from 1.5 to 2.7 times that for a single-outer-land-guided cage bearing over the range

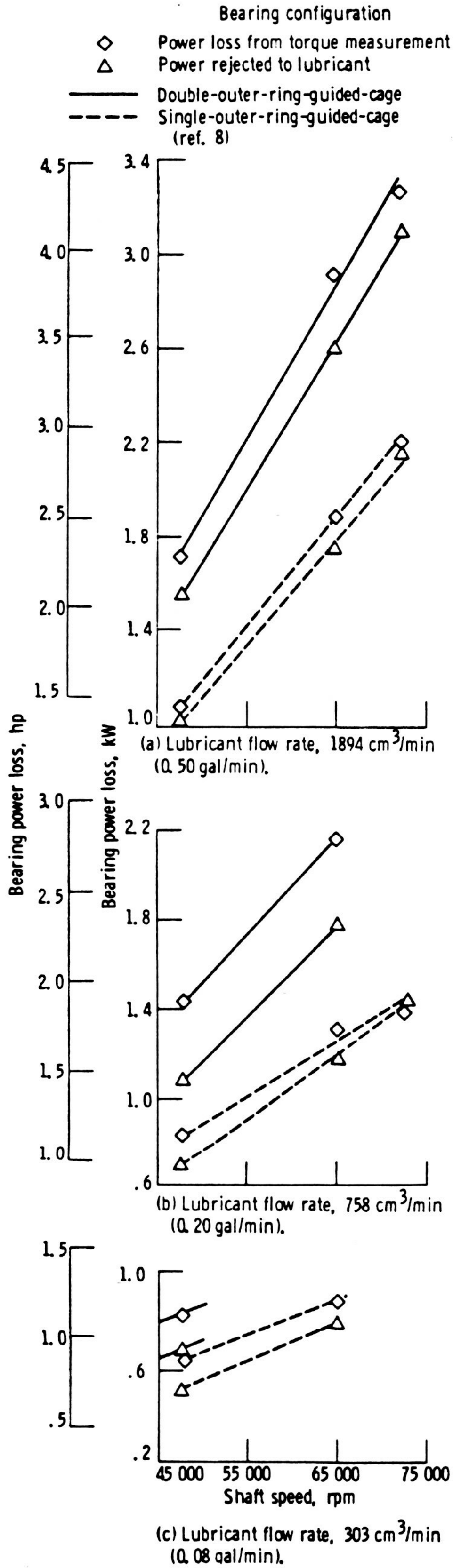


Figure 5. - Power loss as function of shaft speed for two jet lubricated bearing configurations; no outer-ring cooling. Combined load: thrust, 667 N (150 lb); radial, 222 N (50 lb).

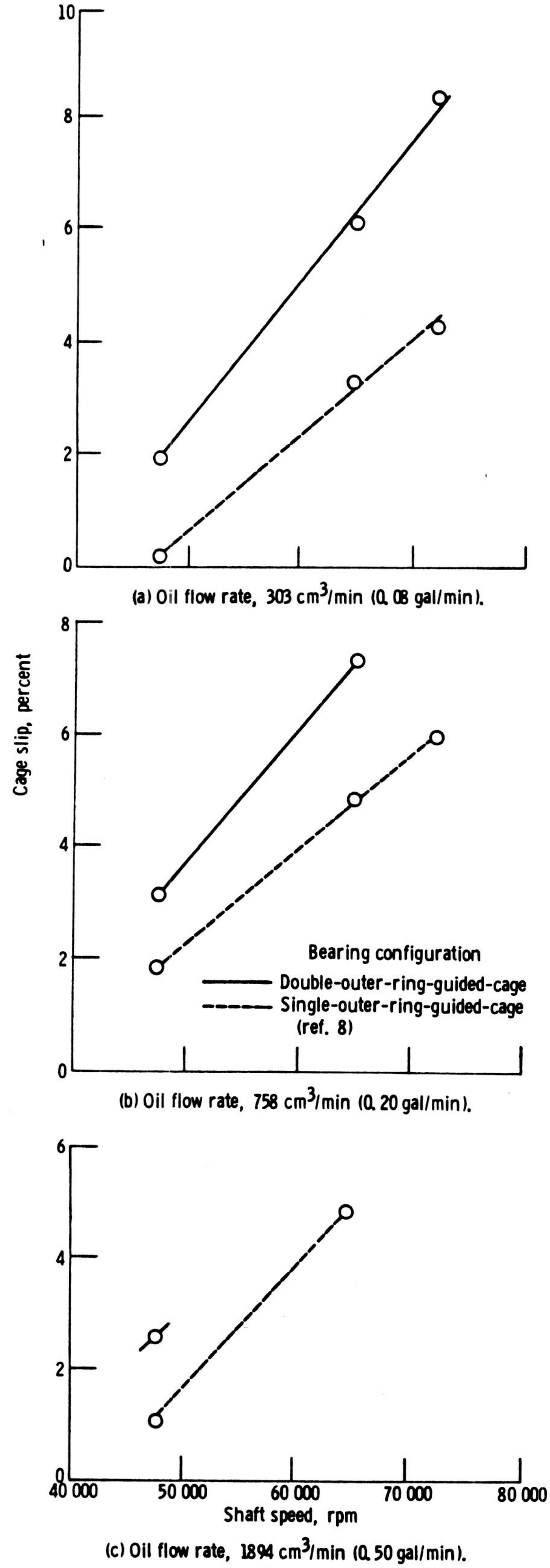


Figure 6. - Effect of shaft speed on cage slip for two jet lubricated bearing configurations; no outer-ring cooling. Combined load: thrust, 667 N (150 lb); radial, 222 N (50 lb).

of lubricant flow rates and speeds shown. The double-outer-land-guided cage, because of its greater surface area, has more drag as it rotates against the outer ring. This increased drag, and the oil churning, reduces cage speed (increases slip) to a greater extent than with the (less surface area) single-outer-land-guided cage. The observed increase in percent cage slip with increased shaft speed for both configurations could be expected because of centrifugal forces decreasing the ball load, and thus traction, at the inner raceway contact. Increased shaft speed also increases the drag at the land area, especially with the double-land design, increasing cage slip.

Through-the-Inner-Ring Lubricated Bearing

Parametric tests were also conducted in the high-speed bearing tester with bearings having a single-outer-land-guided cage with lubricant supplied through the inner ring (fig. 3(c)).

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 inner- and outer-ring temperature is shown in figures 7 and 8. 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 7 and 8 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 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. 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 8, 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 flow path in the inner-ring grooves is greatly limited by the small surface area in contact with the oil.

The results shown in figures 7 and 8 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.

Effect of outer-ring cooling. – 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. These results are also shown in figures 7 and 8. At the lowest shaft speed of 47 200 rpm (fig. 7(a)) outer-ring cooling reduced the outer-ring temperature by about 36 to 8 K (65° to 14° 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. 7(b) and (c)), the outer-ring temperature reduction was about 42 to 16 K (75° to 28° 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. 8) for either oil-flow distribution. The reduction of inner-ring temperature varied from 0 to 6 K (0° to 11° F) over the entire range of total oil flows, shaft speeds, and oil-flow distributions used in these tests.

Comparison of Inner-Ring and Jet Lubricated Bearings

Temperatures. – The effect of speed on test bearing temperature for 50–50 and 25–75 percent total oil-flow distributions through the inner ring is shown in figure 9 and is compared with that of the jet-lubricated bearing (fig. 3(b)). Both bearings were dimensionally identical and 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 for the jet-lubricated bearing over the range of speeds and total oil-flow rates investigated. The oil impinging on the inner

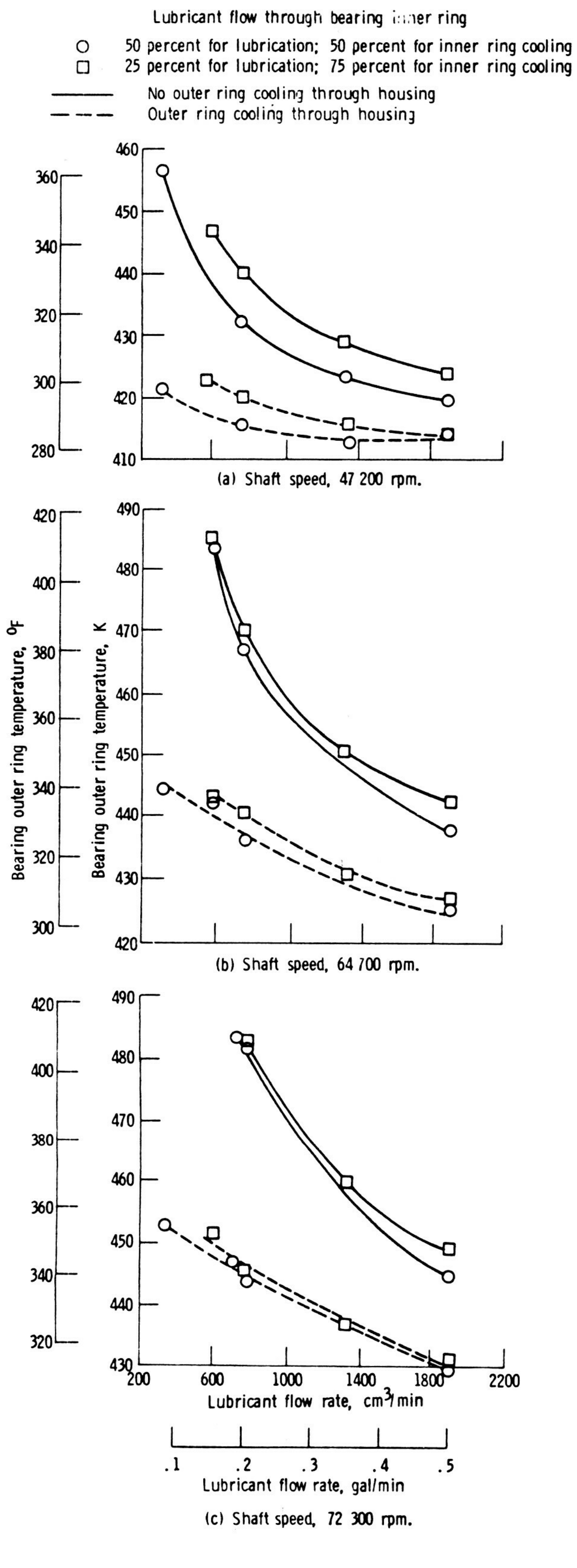


Figure 7. - Effect of lubricant flow rate on bearing outer-ring temperature for two oil distribution paths with and without outer-ring cooling.

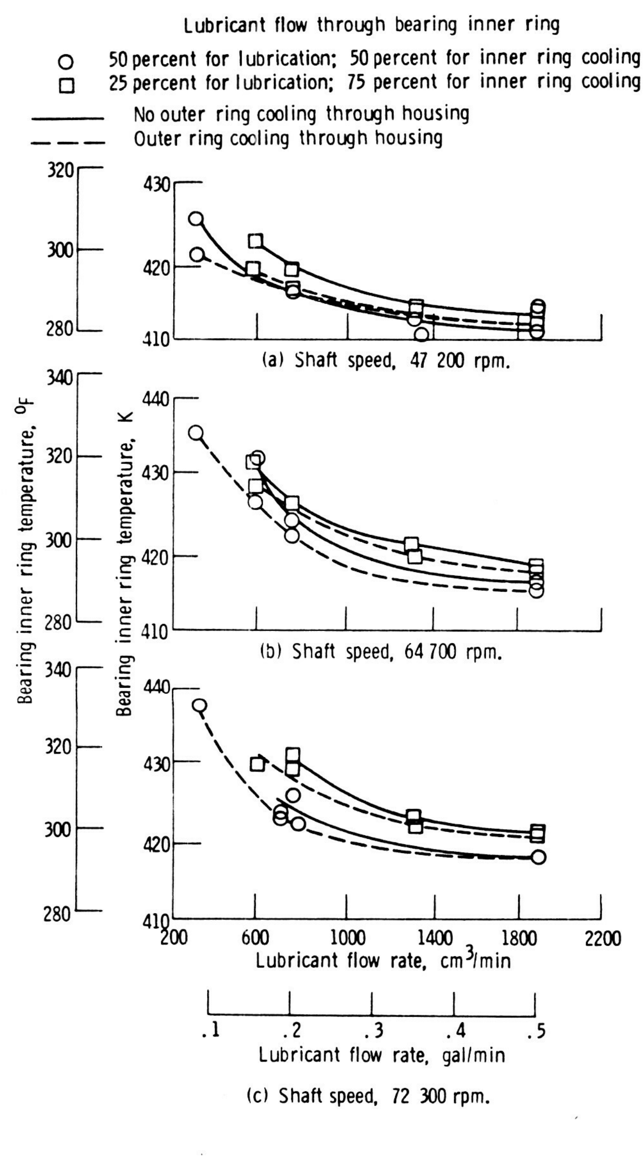


Figure 8. - Effect of lubricant flow rate on bearing inner-ring temperature for two oil distribution paths with and without outer-ring cooling.

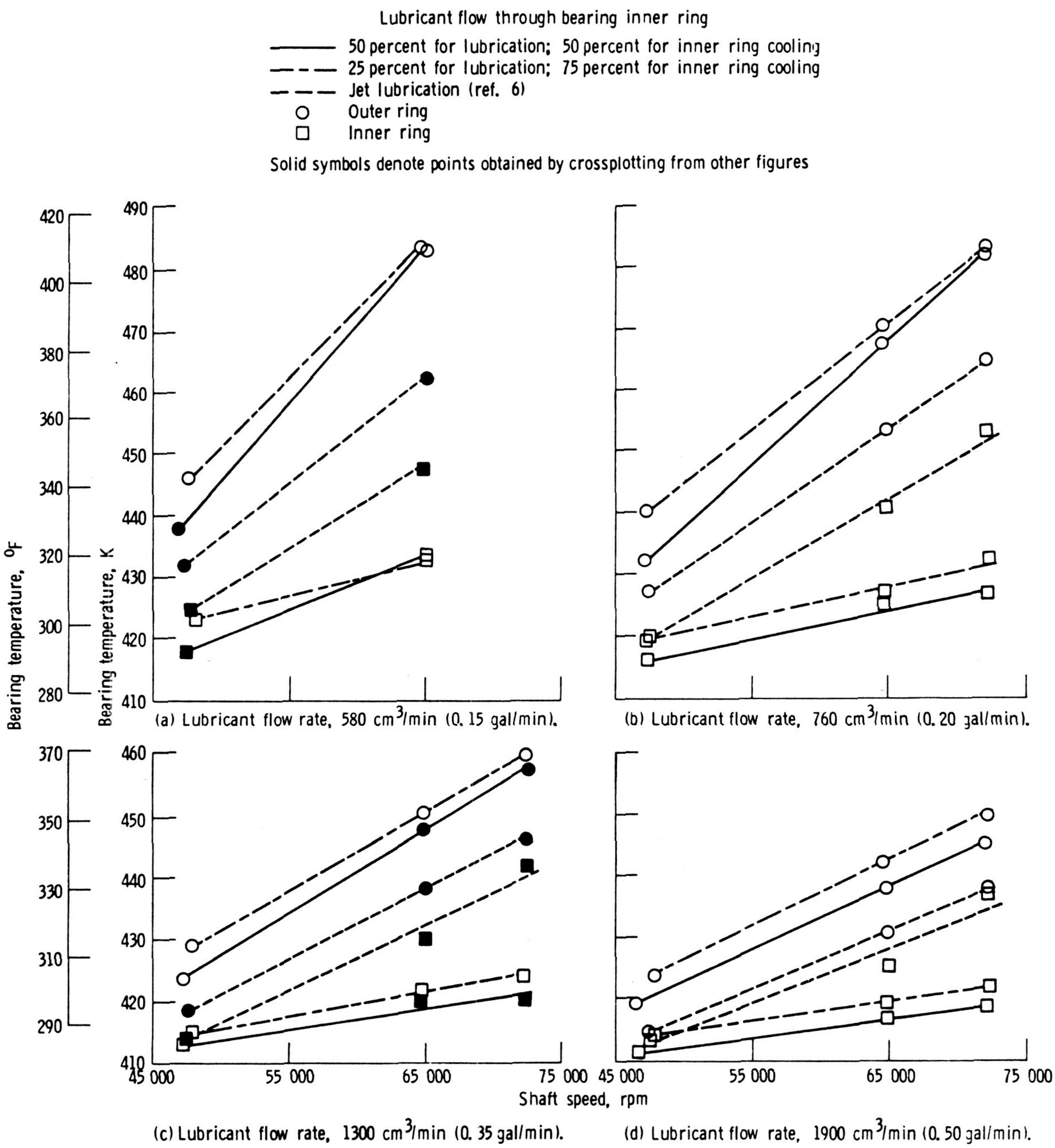


Figure 9. - Comparison of test bearing temperatures as function of shaft speed for two lubricant supply systems. No outer-ring cooling.

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.

Power loss. – The power loss in the bearing, obtained from torque measurements taken with a strain gage attached to the bearing housing, is shown in figure 10. The measured torque for the jet-lubricated and inner-ring-lubricated bearings was 0.011 to 0.033 N·m (1.00 to 2.90 lb·in.) over the range of lubricant flow rates and speeds tested. Power loss increased with speed at each flow rate investigated. The greatest increase in power loss was about 1.27 kW (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. 10(d)). There was very little difference in power loss for the two methods of lubrication except at the two highest flow rates (figs. 10(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 power loss as determined from heat-transfer calculations for a single-outer-land-guided-cage bearing using two different methods of lubrication are compared in figure 11. These data are almost identical to those in figure 10 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. 11(c) and (d)). At these flow rates the jet-lubricated bearing had the lowest value of heat rejected to the lubricant. Maximum power loss was 2.8 kW (3.7 hp) at 72 300 rpm (fig. 11(d)) with a total oil-flow distribution pattern of 25–75 percent through the bearing.

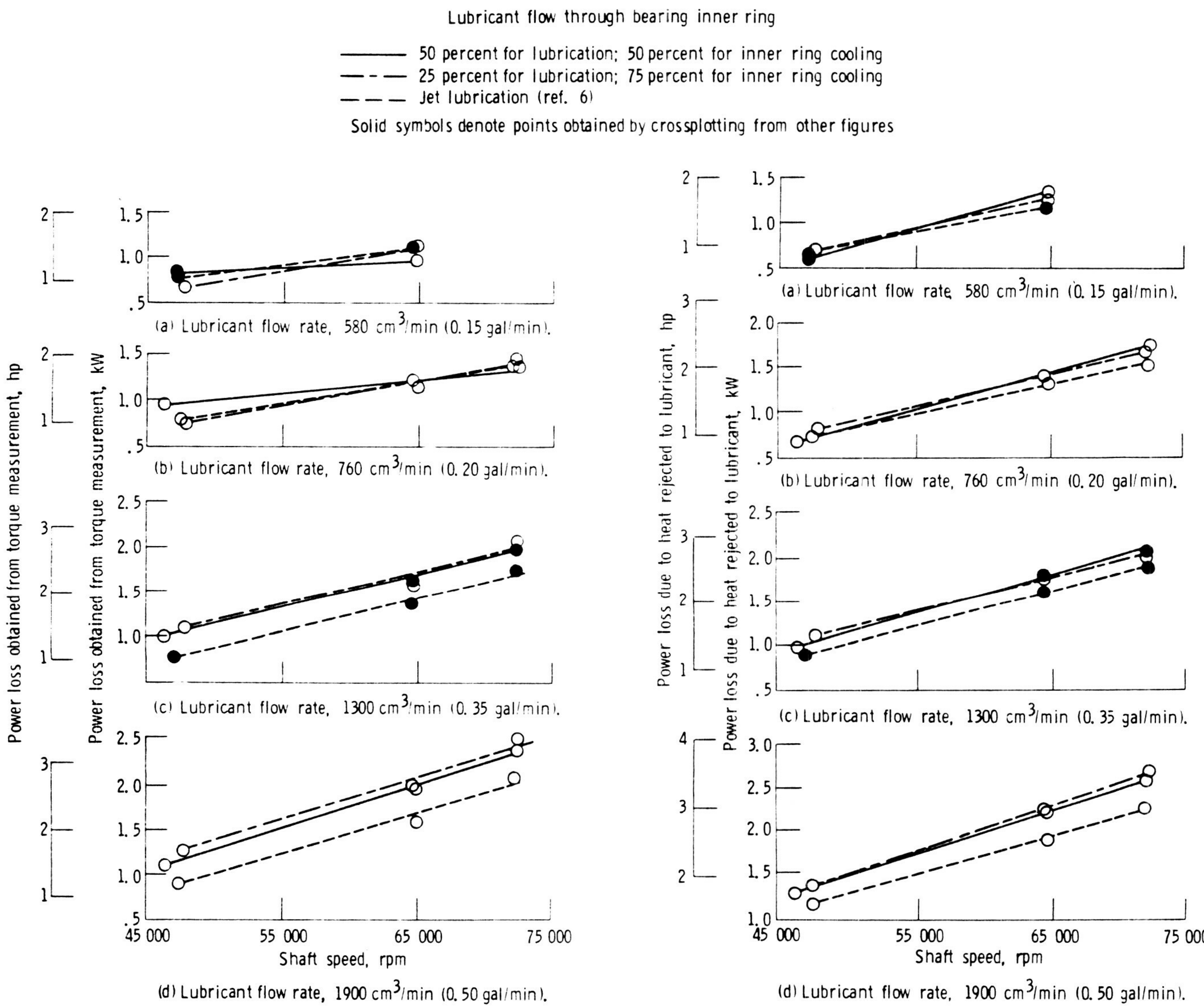


Figure 10. - Power loss obtained from torque measurement as function of shaft speed for inner-ring and jet lubrication. No outer-ring cooling.

Figure 11. - Power loss due to heat rejected to lubricant as function of shaft speed for inner-ring and jet lubrication. No outer-ring cooling.

Cage slip. – The effect of lubricant flow rate on percent cage slip for a single-outer-land-guided-cage bearing is shown in figure 12. 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. 12(a) and (b)). However, at the maximum speed of 72 300 rpm (fig. 12(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 rpm, at a total oil-flow rate of 1900 cm^3/min (0.50 gal/min) (fig. 12(c)) with a total oil-flow distribution pattern of 50–50 percent through the inner ring.

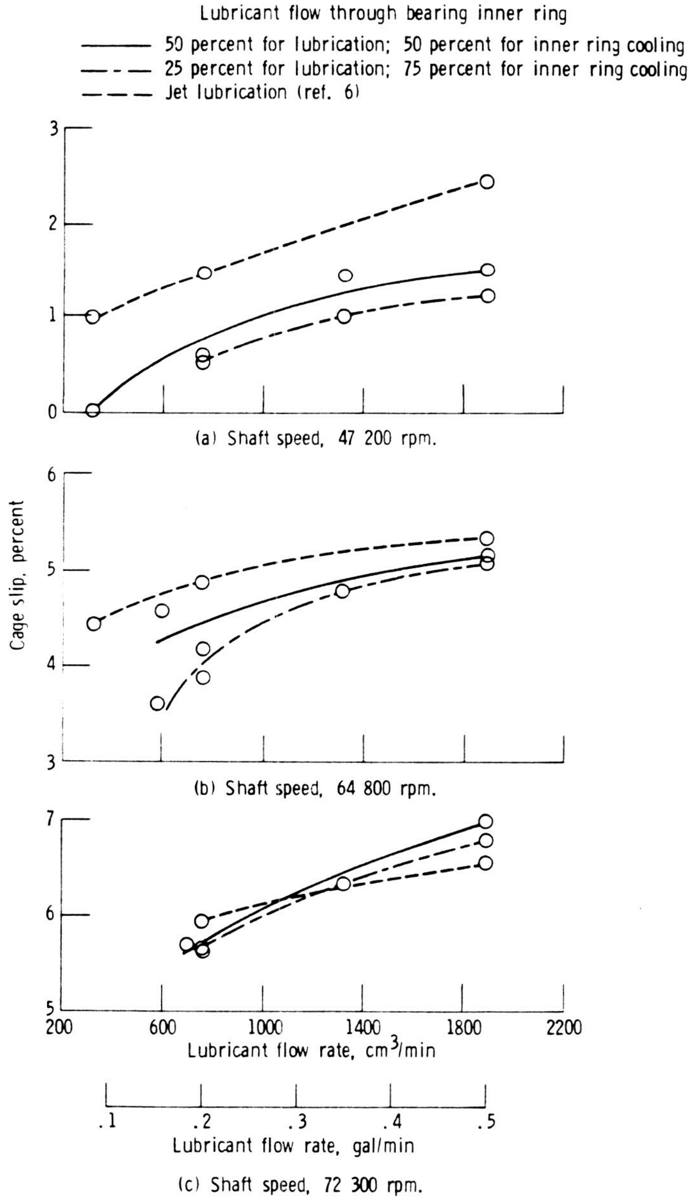
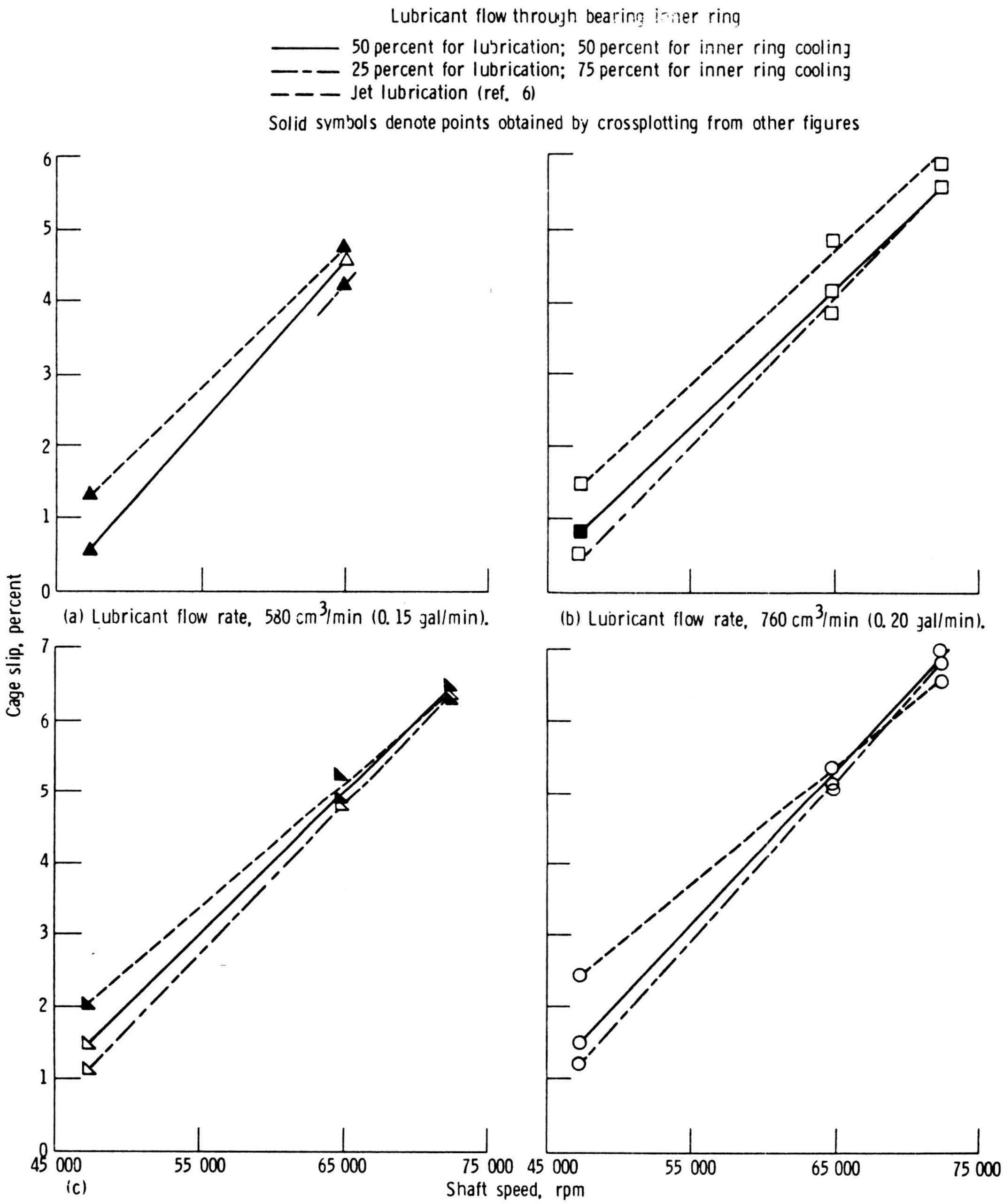


Figure 12. - Effect of lubricant flow rate on cage slip for inner-ring and jet lubrication. No outer-ring cooling.

Figure 13 shows that percent cage slip increases with speed at about the same rate for each method of lubrication 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) over the entire speed range tested (figs. 13(a) and (b)). However, as the flow rate was increased to 1300 and 1900 cm^3/min (0.35 and 0.50 gal/min), the rate of increase in cage slip of the jet-lubricated bearing was less than that of the inner-ring-lubricated bearings (figs. 13(c) and (d)). This resulted in a cage slip value equal to or less than the percent cage slip of the inner-ring-lubricated bearing at the higher speeds. Although the flow



(a) Lubricant flow rate, 580 cm^3/min (0.15 gal/min).

(b) Lubricant flow rate, 760 cm^3/min (0.20 gal/min).

(c) Lubricant flow rate, 1300 cm^3/min (0.35 gal/min).

(d) Lubricant flow rate, 1900 cm^3/min (0.50 gal/min).

Figure 13. - Effect of shaft speed on cage slip for inner-ring and jet lubrication. No outer-ring cooling.

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. Thus the percent cage slips became about equal, as shown in figures 13(c) and (d).

Visual examination of the bearing after running showed no damage to the raceways or the balls, indicating that the cage slip was not of sufficient magnitude to affect the satisfactory operation of the bearing. There was no sign of significant wear on the cage surfaces, and the silver plate had not worn through.

Summary of Results

Parametric tests were conducted in a high-speed, high-temperature bearing tester using 35-mm-bore, angular-contact ball bearings with either a single- or double-outer-land-guided cage. The bearing had a nominal contact angle of 24°. The bearing was jet-lubricated (with a 20 m/sec (66 ft/sec) jet velocity) or had axial grooves and radial holes which permitted lubrication through the bearing inner ring. 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.

Test conditions for all tests included a radial load 222 N (50 lb) and/or a thrust load of 667 N (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).

The oil used for the parametric studies was a neopentylpolyol (tetra)ester. This type II oil is qualified to the MIL–L–23699 specifications. The following major results were obtained:

1. Successful operation of a 35-mm-bore ball bearing, employing a double-outer-land-guided cage and jet lubrication, was accomplished up to 72 600 rpm (2.5 million DN) at a combined load of 667 N (150 lb) thrust and 222 N (50 lb) radial.
2. The double-outer-land-guided cage bearing generated substantially higher temperatures than the single-outer-land-guided cage bearing over the entire range of shaft speeds and lubricant flow rates tested.
3. The power loss for the double-outer-land-guided cage bearing was always higher than that for the single at similar test conditions.
4. Percent cage slip for a double-outer-land-guided cage bearing ranged from 1.5 to 2.7 times that for a single-outer-land-guided cage bearing of similar dimensions over the entire range of speed and lubricant flow rates tested.
5. A 35-mm-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.
6. Cooler bearing operation was experienced with a total oil-flow distribution of 50 percent lubrication and 50 percent inner-ring cooling than with a 25–75 percent distribution for the inner-ring-lubricated bearings.
7. Jet-lubrication data showed lower outer-ring temperatures and higher inner-ring temperatures than the results for inner-ring-lubricated bearings.
8. 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.
9. Maximum power loss of 2.8 kW (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).
10. For inner-ring-lubricated bearings a 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.

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