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Summary

Parametric tests were conducted in a high-speed, high-temperature bearing tester with a 35-mm-bore, angular-contact ball bearing with a double-outer-land-guided cage. The bearing had a nominal unmounted 24° contact angle. Provisions were made for jet lubrication of the bearing and for outer-ring cooling. Results were compared with results previously obtained from tests of a similar bearing but having a single-outer-land-guided cage. Test conditions included a combined load of 667 Newtons (150 lb) thrust and 222 N (50 lb) radial, nominal shaft speeds of 48 000 to 72 000 rpm, and an oil-in temperature of 394 K (250° F). Tests were run at flow rates ranging from 303 to 1894 cm^3/min (0.08 to 0.50 gal/min) and a jet velocity of 20 m/sec (66 ft/sec). The lubricant was neopentylopolyol (tetra)ester meeting the MIL–L–23699 specifications.

Successful operation of the 35-mm-bore ball bearing, employing a double-outer-land-guided cage and jet lubrication, was accomplished up to 2.5 million DN.

A bearing with a double-outer-land-guided cage generated higher temperatures than one with a single-outer-land-guided cage. Other than cage design, the two bearings were dimensionally the same. Identical ranges of shaft speeds and lubricant flow rates were used. Outer-ring cooling of the double-outer-land-guided cage bearing to thermally balance the bearing temperature resulted in lower overall bearing operating temperatures at all speeds and lubricant flow rates employed.

Power loss increased with speed and lubricant flow rate for both the single- and double-outer-land-guided cage bearings. Power loss increased with speed at a faster rate as the lubricant flow rate increased. The power loss of the double-outer-land-guided cage bearing was always higher than that of the single-land-guided design at similar test conditions.

Percent cage slip was minimal for both bearing configurations for all speeds and flow rates tested. Pecent 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. Cooling the outer ring did not appreciably affect the cage slip or power loss for either bearing configuration.

Introduction

The next generation of turbojet engines in the small class 4.45 to 44.48 N/sec (1 to 10 lb/sec) total air flow requires bearings that operate at a DN range up to 2.5 million. (DN is defined as the shaft speed in rpm multiplied by the bearing bore in mm.) These bearings must be capable of performing satisfactorily at the high temperatures that are common to high-speed operation. The bearing designs and lubrication techniques used for these engines are two of several elements that must be refined and optimized for reliable performance and long life.

Large-bore ball and roller bearings have been successfully tested at speeds to 3.0 million DN (refs. 1 to 4). However, in these tests 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 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.

One of the prinicipal elements of a bearing that affects its satisfactory high-speed operation is the bearing cage. As indicated in reference 5, bearing wear and ultimate failure at a high DN value 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. In reference 5, using 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 where failure occurred. For satisfactory high-speed operation of a small bore jet-lubricated bearing, therefore, an outer-ring-land-guided cage is recommended because its outer-ring land surfaces are more efficiently lubricated by means of centrifugal oil flow effect and because they are generally cooler than the corresponding surfaces of an inner-ring controlled cage (ref. 6). This hypothesis was borne out for the 75-millimeter, jet-lubricated roller bearing tests described in reference 7, where the best overall performance was reported for bearings with outer-ring-land-guided cages.

Reference 8 reports results of a high-speed jet-lubricated, 35-mm ball bearing with a cage having a single outer land. Bearing data for a thrust only and a combined thrust and radial load are compared. The experimental work described herein is a continuation of the experiments performed in reference 8. The test bearing is identical in size and contact angle to that in reference 8; only the cage and outer ring have been altered to accommodate a double-outer-land.

The objectives of this study were to determine the parametric effects of cage design, shaft speed, lubricant flow rate, and outer ring cooling on the operation of a 35-mm-bore angular contact bearing. Test conditions included combined radial and thrust loads of 222 N (50 lb) and 667 N (150 lb), respectively, and nominal shaft speeds from 48 000 to 72 000 rpm.

Lubricant was introduced to the bearing 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). Outer-ring oil flow cooling rates were 0 to 246 cm^3/min (0 to 0.065 gal/min) at a 394 K (250° F) oil-in temperature. The lubricant was neopentylpolyol (tetra)ester meeting 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, which incorporates the hardware for lubrication, oil removal, and thrust and radial load application and the instrumentation for cage speed measurement. Test bearing torque is measured with strain gages attached to the bearing housing. 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. Radial load is applied to the test bearing through knife-edge bearings, which effectively minimize friction. The test bearing was lubricated by two jets on the nonloaded side of the inner ring. The jet outlets, located approximately 3.0 mm (0.12 in.) from the face of the bearing 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 insured the most efficient lubrication of the test bearing; that velocity was used in all the tests reported. (Ref. 5 reports a similar efficiency with a 20-m/sec (66-ft/sec) jet velocity over other velocities investigated.) Cooling oil is 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 obtained with a magnetic probe. Cage speed was measured with a semiconductor strain gage mounted in a cavity of the housing. Two thermocouples, assembled in the shaft, measure inner-ring temperatures by means of 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 Bearing

The test bearing was an ABEC7 grade, 35-mm-bore angular-contact ball bearing with a double-outer-land-guided cage as shown in figure 3(a). The bearing contained 16 balls with a nominal 7.14-mm (0.281-in.) diameter. 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 (AMS6415) heat treated to a Rockwell C

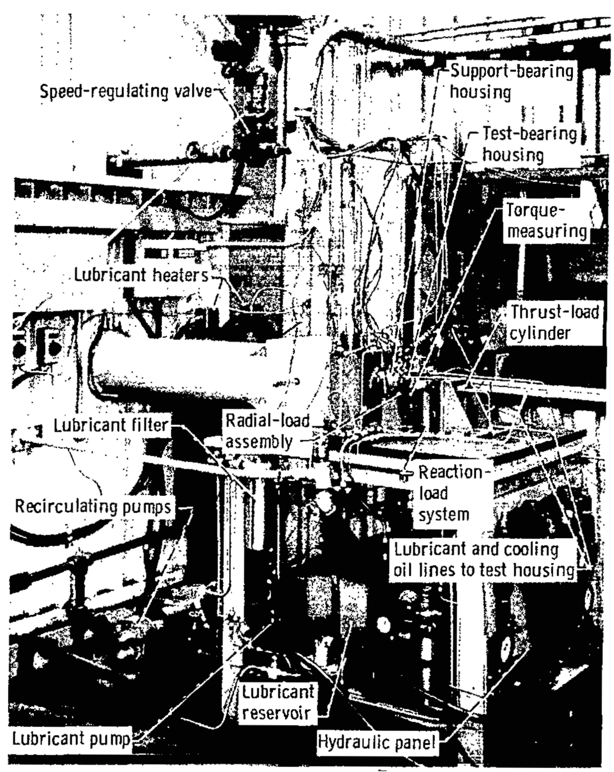


Figure 1. - High-speed, small-bore-bearing test machine.

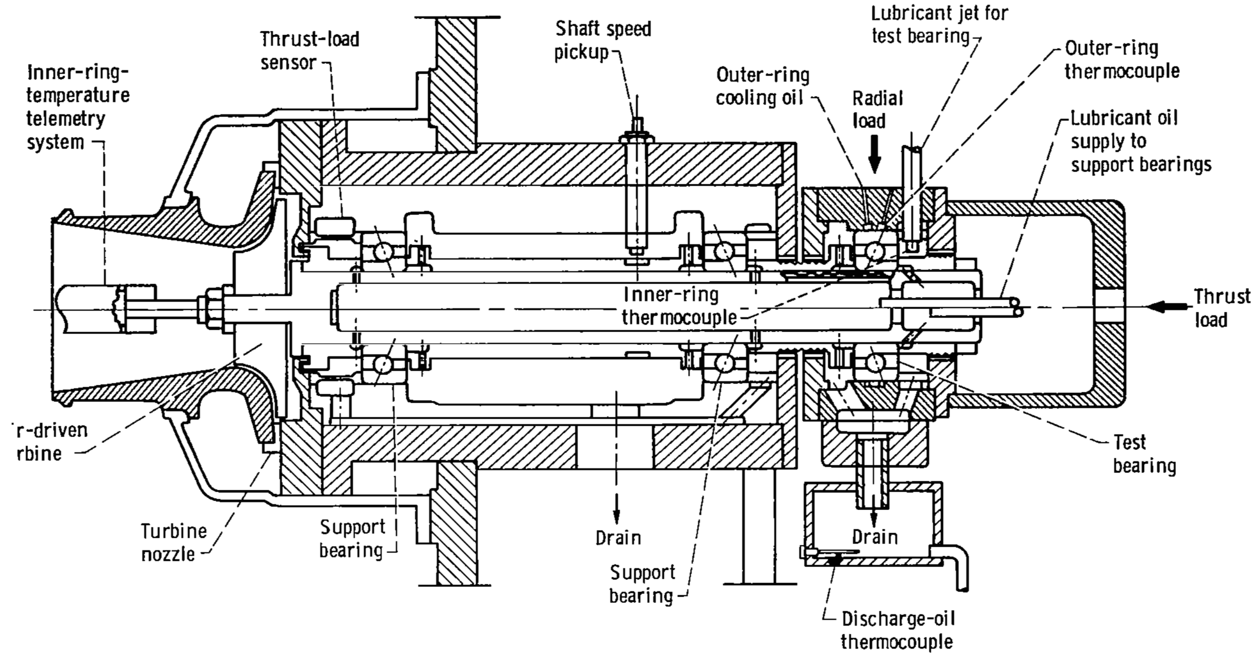
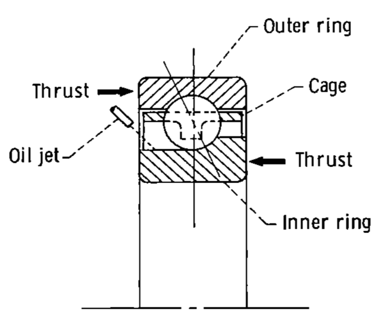
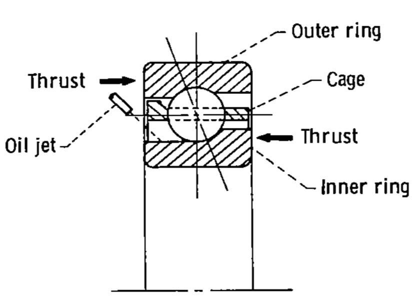


Figure 2. - Schematic of high-speed, small-bore-bearing test machine.



(a) Double-outer-land-guided cage.



(b) Single-outer-land-guided cage.

Figure 3. - Angular contact ball bearing.

hardness of 28 to 36 and having 0.0203- to 0.0381-mm (0.0008- to 0.0015-in.) thick silver plating (AMS2412) all over. The cage balance was within 4.9×10^{-6} N m (7×10^{-4} oz-in.). Additional specifications are shown in table I. The outside diameter of both the double- and single-outer-land-guided cages, shown in figures 3(a) and (b), 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 used in the bearing in reference 8. The inner ring of both bearings was geometrically and dimensionally the same.

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 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

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/lb °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

system, a combined test load of 222 N (50 lb) radial and 667 N (150 lb) thrust and a total lubricant flow rate of 1894 cm^3/min (0.500 gal/min) were applied. 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 (20 to 25 min), the oil-in temperature and lubricant flow rate was set and the speed was increased to the desired value. A series of tests 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 oil flow rate. At each speed and flow condition a separate test was run during which the outer-ring cooling flow was adjusted to achieve equal inner- and outer-ring temperature. Three lubricant oil flow rates of 303 to 1894 cm^3/min (0.08 to 0.50 gal/min) with a nominal jet velocity of 20 m/sec (66 ft/sec) were used.

After the test runs described above, the resulting data were compared with those of a similar bearing with a single-outer-land-guided cage, reported in reference 8. If it became apparent during the course of testing that the conditions would result in distress of the bearing or rig or that the conditions would generate a bearing temperature above 491 K (425° F), the test point was aborted or omitted.

Results and Discussion

Parametric tests were conducted in a high speed bearing tester with a 35-mm-bore ball bearing having a double-outer-land-guided cage. Tests results obtained are compared with those of a previously run single-outer-land-guided cage bearing. Other than cage design and the inside diameter of the outer ring (fig. 3), 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-outer-land-guided cage (ref. 8) and the double-outer-land-guided cage decrease with an increase in lubricant flow rate for each test speed, 47 600, 64 900, and 72 250 rpm. 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. At 47 600 rpm (fig. 4(a)) and a lubricant flow rate of 1894 cm^3/min (0.50 gal/min), the outer ring temperature of the double-outer-land-guided cage bearing without outer ring cooling is approximately 13 K (23° F) higher than the single-outer-land-guided cage bearing. At 64 900 rpm (fig. 4(b)) this differential increases to 21 K (38° F), and at 72 250 rpm (fig. 4(c)) to 25 K (45° F) at a maximum 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

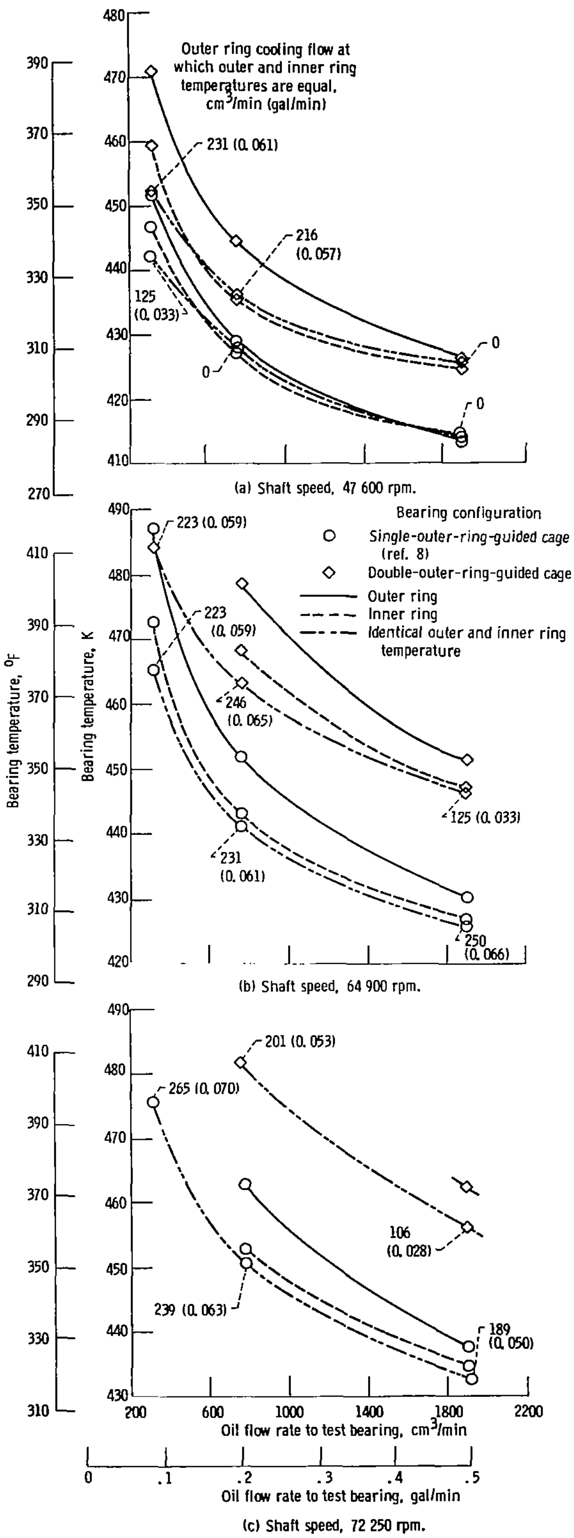


Figure 4. - Effect of oil flow on test bearing temperature for two bearing configurations with and without outer-ring cooling. Combined load: thrust, 667 N (150 lb); radial, 222 N (50 lb).

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, 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.

Because many applications require (for best performance and optimal operating clearance) a minimal temperature gradient between the bearing inner and outer rings, tests were conducted to find the outer-ring cooling-oil flow rate that produced a thermally balanced bearing. These results are also shown in figure 4 with the required outer ring cooling flow labeled at each data point. Adding outer-ring cooling flow in an amount to thermally balance the inner and outer ring affected the outer ring much more than the inner ring, as expected. Generally, outer-ring cooling resulted in a decrease in the overall bearing operating temperature regardless of speed, lubricant flow rate, or type of cage design.

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 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 (from oil company specifications), J/kg K, (Btu/lb °F)
t_{out}	oil outlet temperature, K (°F)
t_{in}	oil inlet temperature, K (°F)

The horsepower loss of a double, compared with that of a single-outer-land-guided cage bearing is shown in figures 5 and 6. Figure 5 shows that power loss obtained from torque readings and figure 6, the power loss determined from heat rejected to the oil, including the heat (power) absorbed by the outer-ring cooling oil. For convenience, values of power loss were converted from J/min to kW.

Figures 5 and 6 show that power loss increases with speed and lubricant flow rate for both the single- and double-outer-land-guided cage bearing configurations. Outer-ring cooling does not appreciably affect power loss for either bearing configuration. A comparison of power losses shows that the double-outer-land-guided cage bearing has a decided disadvantage compared with the single-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 the drag resulting from a total land area three times that of the single land design. Figures 5(a) and 6(a) show that a very high lubricant flow rate (1894 cm^3/min (0.50 gal/min)) can be undesirable because of the increased power loss in the bearing over that for a low flow rate (such as 303 cm^3/min (0.08 gal/min); (figs. 5(c) and 6(c)).

Figure 7 compares power losses obtained from torque readings with those from heat rejected to the lubricant for double- and a single-land-guided-cage bearings. No outer-ring cooling was employed 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) the difficulty of accurately accounting for all the heat dissipated by conduction, convection, and radiation to the surrounding environment and (2) the problem of locating thermocouples in ideal positions for a most accurate reading of oil-in and oil-out temperatures.

The results from both methods of obtaining bearing power loss (fig. 7) are in agreement except for magnitude.

Effect of Lubricant Flow Rate on Cage Slip

In order to determine percent cage slip, the epicyclic cage speed, C_{epi}, at the various test speeds was obtained from a computer program which took into consideration centrifugal force effects on contact angle. Elastic contact forces are considered in a raceway control type solution. Thermal and lubricant effects were not considered in this computer solution of epicyclic cage speed. The epicyclic cage speed was combined with the measured

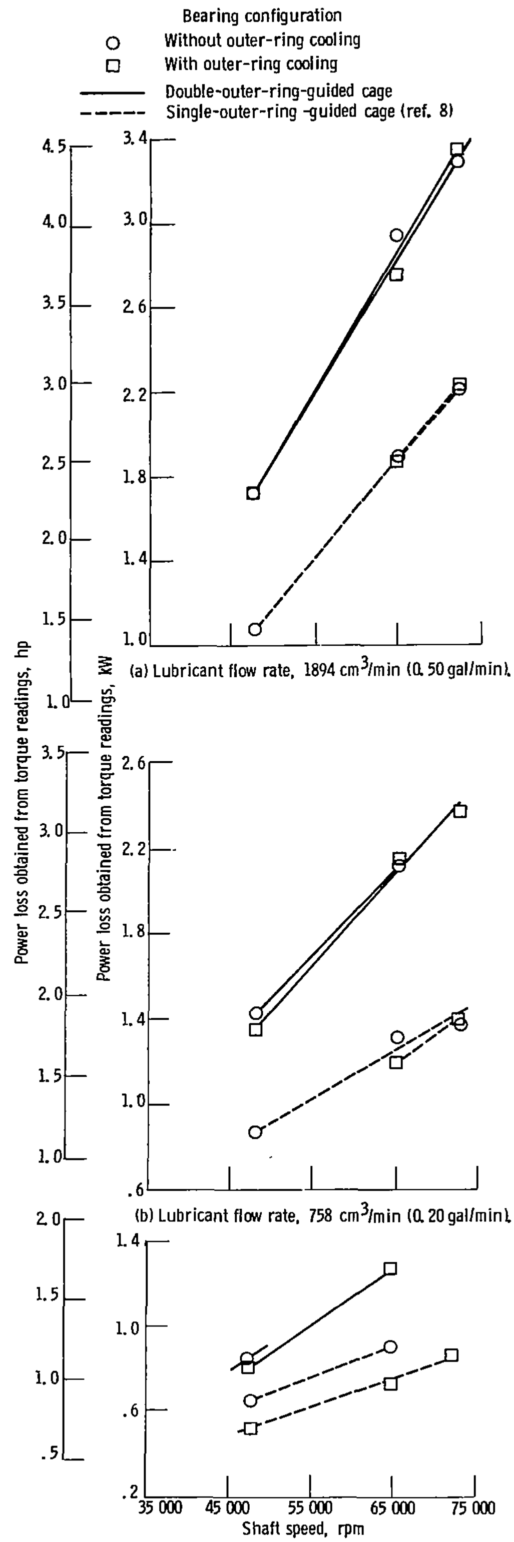


Figure 5. - Power loss obtained from torque readings as a function of shaft speed for two bearing configurations with and without outer-ring cooling. Combined load: thrust, 667 N (150 lb); radial, 222 N (50 lb).

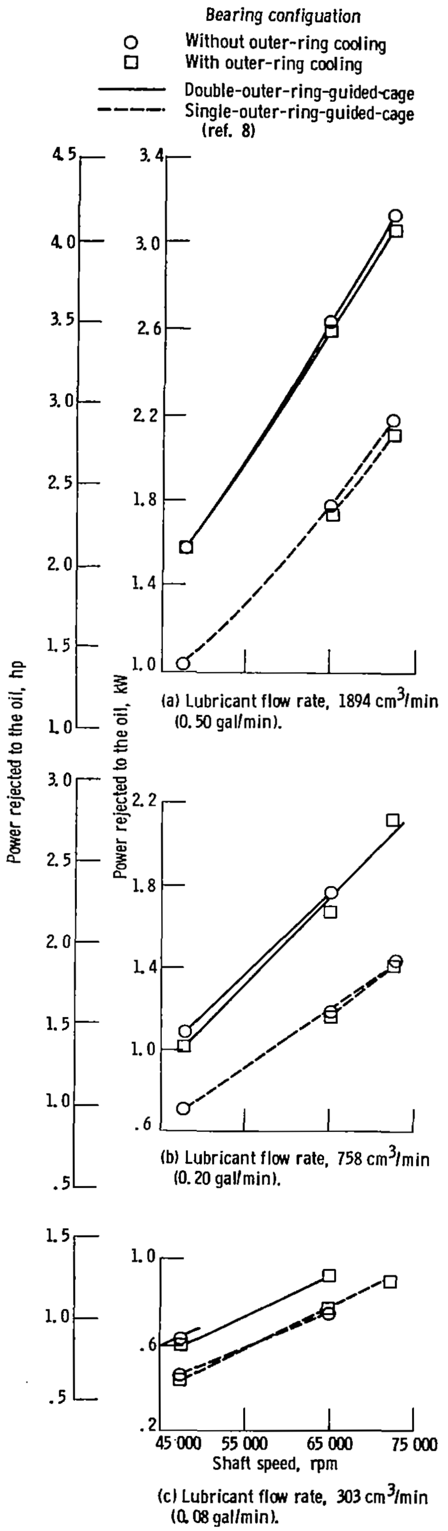


Figure 6. - Power rejected to the oil as function of shaft speed for two bearing configurations with and without outer-ring cooling. Combined load: thrust, 667 N (150 lb); radial, 222 N (50 lb).

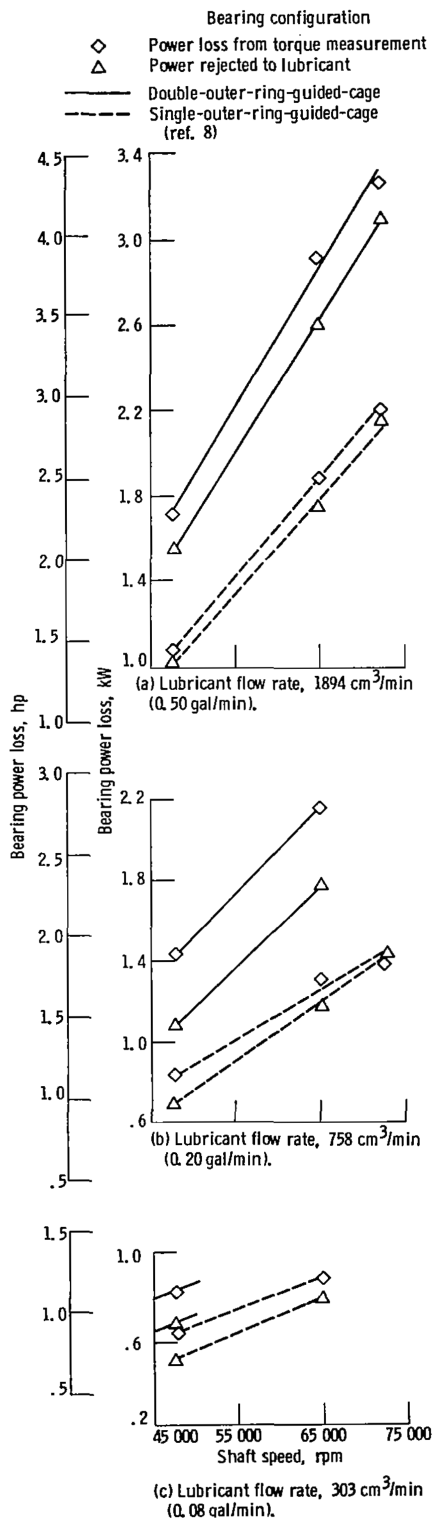


Figure 7. - Power loss as function of shaft speed for two bearing configurations. No outer ring cooling; combined load: thrust, 667 N (150 lb); radial, 222 N (50 lb).

experimental cage speed, C_{exp}, to obtain percent cage slip as follows:

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

The effect of lubricant flow rate on percent cage slip for a single and double-outer-land-guided cage bearing configuration is shown in figure 8. The effect of outer ring cooling on percent cage slip is also shown.

For all speeds and flow rates tested figure 8 the percent cage slip was minimal for both bearing configurations. The small increase in slip with flow rate is primarily due to additional drag on the balls. 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 of lubricant flow rates and speeds tested. The double-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 discussed earlier, reduces cage speed (increases slip) to a greater extent than with the (less surface area) single-outer-land-guided cage.

Figure 9 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-ring-guided cage bearing showed a higher percent cage slip than the single-outer-ring-guided cage at all speeds and lubricant flow rates tested. It also showed a higher rate of cage slip with increased flow. The observed increase in percent cage slip with increased shaft speed for both configurations could be expected due to 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.

Cooling the outer ring to produce a thermally balanced bearing did not appreciably affect the percent cage slip data for either of the two bearing configurations.

The results of this investigation indicate that a bearing can operate at reduced temperatures with outer-ring cooling (fig. 4) without sacrificing power loss (figs. 5 to 7) or increasing cage slip (figs. 8 and 9).

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.

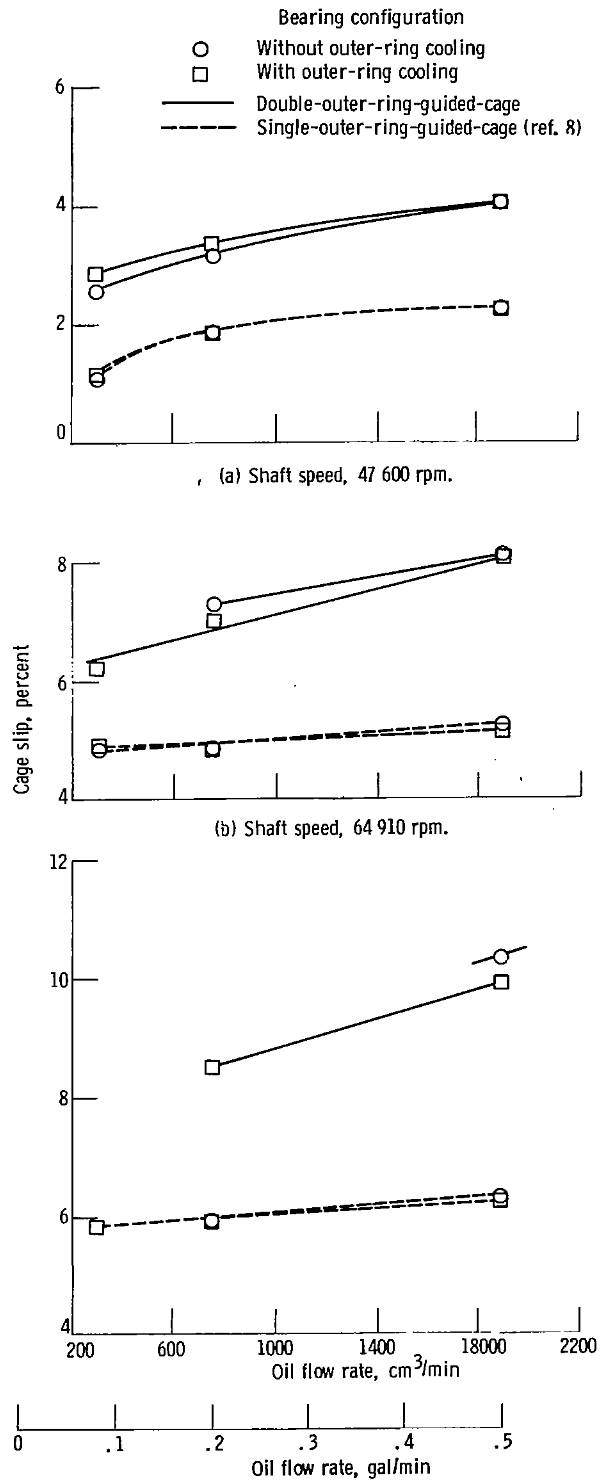


Figure 8. - Effect of oil flow rate on cage slip for two bearing configurations, with and without outer-ring cooling. Combined load: thrust, 667 N (150 lb); radial, 222 N (50 lb).

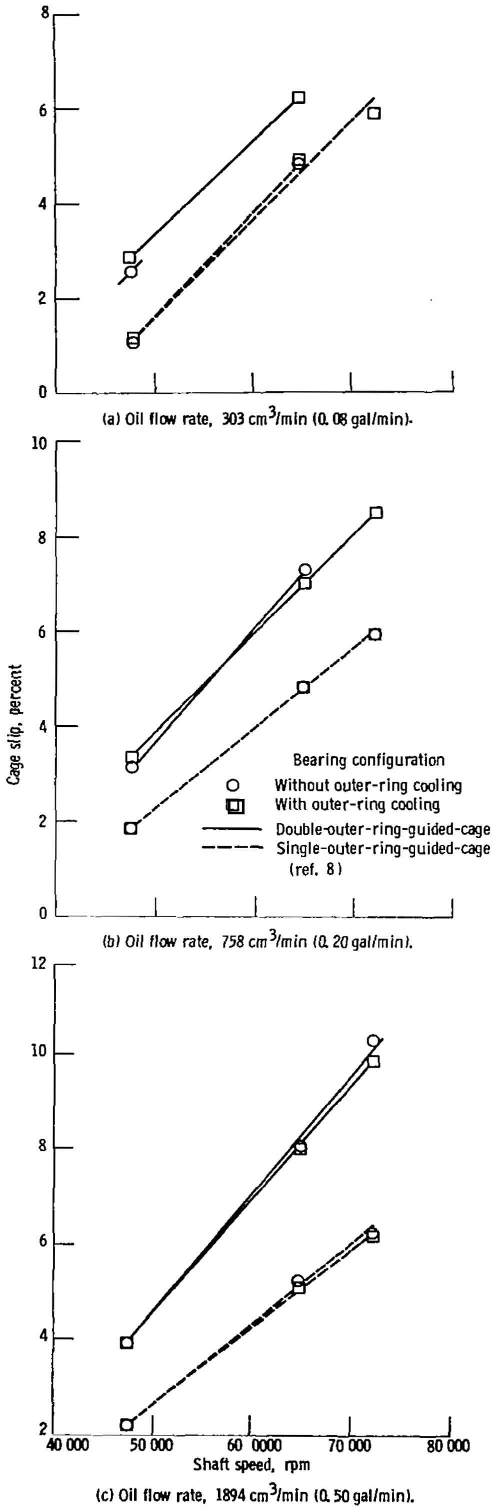


Figure 9. - Effect of shaft speed on cage slip for two bearing configurations with and without outer-ring cooling. Combined load: thrust, 667 N (150 lb); radial, 222 N (50 lb).

Summary of Results

Parametric tests were conducted in a high-speed, high-temperature bearing tester with a 35-mm-bore-angular-contact ball bearing with a double-outer-land-guided cage. The bearing had a nominal contact angle of 24°. Results were compared with those obtained in a previous investigation with a single-outer-land-guided cage bearing. The bearing was jet-lubricated and the outer ring was cooled. Test conditions included a combined load of 667 N (150 lb) thrust and 222 N (50 lb) radial, nominal shaft speeds of 48 000 to 72 000 rpm, and an oil-in temperature of 394 K (250° F). The bearing was jet-lubricated at flow rates from 303 to 1894 cm^3/min (0.08 to 0.50 gal/min) with a 20 m/sec (66 ft/sec) jet velocity. The lubricant was neopentylpolyol (tetra) ester, meeting the MIL–L–23699 specifications. The following 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. Cooling the outer ring of a double-outer-land-guided cage bearing decreased the overall bearing operating temperature regardless of shaft speed and lubricant flow rate employed.
4. Power loss increased with speed and lubricant flow rate for both bearing configurations. Power loss increased with speed at a faster rate as the lubricant flow rate increased. The power loss for the double-outer-land-guided cage bearing was always higher than that of the single at similar test conditions.
5. For all speeds and flow rates tested the percent cage slip was minimal for both bearing configurations. Cooling the outer ring did not appreciably affect the cage slip for either of the two bearing configurations.
6. 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.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, March 24, 1980,
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15. Supplementary Notes

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16. Abstract

Parametric tests were conducted with a 35-mm-bore angular contact ball bearing with a double-outer-land-guided cage. Provisions were made for jet lubrication and outer-ring cooling of the bearing. Test conditions included a combined thrust and radial load at nominal shaft speeds of 48 000 to 72 000 rpm, and an oil-in temperature of 394 K (250° F). Successful operation of the test bearing was accomplished up to 2.5 million DN. Test results were compared with those obtained with similar bearing having a single-outer-land-guided cage. Higher temperatures were generated with the double-outer-land-guided cage bearing, and bearing power loss and cage slip were greater. Cooling the outer ring resulted in a decrease in overall bearing operating temperature.

17. Key Words (Suggested by Author(s))	18. Distribution Statement
High speed bearings; Rolling element bearings; Ball bearings; Jet-lubricated high DN bearing cage design	Unclassified - unlimited STAR Category 37

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