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EXPERIMENTAL PERFORMANCE OF 75-MILLIMETER-BORE ARCHED OUTER-RACE BALL BEARINGS TO 2.1 MILLION DN

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16. Abstract

An experimental investigation was performed to determine the operating characteristics of arched outer-race bearings and to compare the data with those for a similar, but conventional, deep-groove ball bearing. The bearings were tested over a range of shaft speeds up to 28 000 rpm at a thrust load of 2200 newtons (500 lb). Also, one bearing was operated at 26 000 rpm with a range of thrust loads. The amounts of arching were 0.13, 0.25, and 0.51 millimeter (0.005, 0.010, and 0.020 in.). All bearings operated satisfactorily. However, the arched bearing outer-race temperatures and torques were consistently higher than those for the conventional bearing.

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EXPERIMENTAL PERFORMANCE OF 75-MILLIMETER-BORE ARCHED OUTER-RACE BALL BEARINGS TO 2.1 MILLION DN

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SUMMARY

Theoretically, an arched outer-race ball bearing, operating at relatively high speeds and light loads, will have two contact points per ball at the outer race. Thus, the ball-race contact load will be reduced and the bearing fatigue life significantly improved. However, analyses also show that a considerable amount of spinning occurs at these outer-race contacts. Therefore, performance characteristics of arched bearings were determined experimentally and compared with those of a similar, but conventional, bearing.

The 75-millimeter bore bearings were tested at a constant 2200-newton (500-lb) thrust load while varying the shaft speed up to a maximum of 28 000 rpm or 2.1 million DN (bearing bore in millimeters times shaft speed in rpm). Then the oil flow rate was varied from 8×10^{-3} to 40×10^{-3} kilogram per second (1 to 5 lb/min) with the bearings operating at a constant 20 000 rpm. Finally, one bearing was operated at 26 000 rpm while the load was varied from 4440 to 440 newtons (1000 to 100 lb). The amounts of arching used were 0.13, 0.25, and 0.51 millimeter (0.005, 0.010, and 0.020 in.).

The results of these tests showed that the outer-race temperature and torque of the arched bearings were consistently higher than those of a similar conventional bearing operating under the same conditions. It was also observed that the outer-race temperature and cage slip decreased as the amount of arching increased.

INTRODUCTION

Bearings in current commercial aircraft turbine engines operate in a speed range up to 2 million DN (bearing bore in millimeters multiplied by shaft speed in rpm). However, trends in gas turbine design indicate that future engines may require bearings that can operate at DN values of 3 million or higher (ref. 1). But, when bearings operate at high DN values, analyses predict a significant reduction in fatigue life due to the high centrifugal forces developed by the balls at the outer race contact (ref. 2).

A possible solution to the high-speed bearing problem would be to reduce the mass of the ball and thereby reduce the centrifugal force. One method of reducing the mass is to make the ball hollow. Theory indicates that a significant improvement in bearing fatigue life can be obtained at the higher DN values if the mass of the balls is reduced 50 percent below that of a comparable solid ball (ref. 3). Both spherically hollow (refs. 4 and 5) and cylindrically hollow drilled (refs. 6 to 8) balls have been fabricated and tested. Both the hollow and the drilled balls experienced early flexure fatigue failures during testing. A further analysis showed that during these tests relatively high stresses existed at the inner surface of the hollow balls (ref. 9) and at the bore of the drilled balls (ref. 10).

Other approaches to the problem of high-speed bearings have also been investigated. A parallel-hybrid bearing (ref. 11) in which a ball bearing and a fluid film bearing share the system load can be used. A series hybrid bearing (refs. 12 and 13) in which a ball bearing and a fluid film bearing share the system rotational speed has been suggested. A large (150-mm bore) series hybrid thrust bearing has been designed and successfully tested at high speeds (refs. 14 and 15). A problem with the hybrid bearings, however, is the mechanical complexity of the system.

There is, though, at least one other approach that requires further evaluation. Initial tests with a concept called an arched outer-race bearing (ref. 16) showed this design operating with a lower torque than a conventional angular contact bearing. Theoretically, an arched bearing operates at low speeds like a conventional bearing with each ball having two ball-race contacts. However, above some higher transition speed, this arched bearing operates with three ball-race contact points per ball. Therefore, when the arched bearing has two contact points per ball at the outer race, the centrifugal loading can be shared, thereby reducing the contact load and increasing the bearing fatigue life. A first-order thrust load analysis of an arched bearing design (ref. 17) indicated the possibility of significant fatigue life improvement. A more complete analysis (ref. 18) also found that the arched outer-race bearing showed a significant improvement in fatigue life over that of a conventional bearing for high-speed, light load applications. However, the analysis of reference 18 also showed that a considerable amount of spinning occurs at the outer-race contacts for the arched bearing.

Therefore, it was the object of this investigation to (1) determine experimentally the operating temperature and torque of arched outer-race bearings with different amounts of arching and (2) compare the arched bearing results with data from a similar conventional bearing having the same diametral play.

The tests were conducted with 215-series, 75-millimeter bore, deep groove, arched outer-race ball bearings. The amount of arching was up to 0.51 millimeter (0.020 in.).

The bearings were operated at a 2200-newton (500-lb) thrust load at shaft speeds up to 28 000 rpm (2.1×10^6 DN) using oil-jet lubrication.

APPARATUS AND INSTRUMENTATION

Bearing Test Rig

A cutaway view of the bearing test apparatus is shown in figure 1. A variable-speed, direct-current motor drives the test bearing shaft through a gear speed increaser. The ratio of the test shaft speed to the motor shaft speed was 14. The limiting speed of the test shaft was 28 000 rpm.

The test shaft was supported by two oil-jet-lubricated ball bearings and was cantilevered at the driven end. The test bearing was thrust loaded by a pneumatic cylinder through an externally pressurized gas thrust bearing. A gas bearing was used so that test bearing torque could be measured.

Bearing torque was measured with an unbonded strain-gage force transducer connected to the periphery of the test bearing housing as shown in figure 1. This torque was recorded continuously by a millivolt potentiometer. Estimated accuracy of the data recording system was ±0.006 newton-meter (±0.05 lb-in.).

Bearing outer-race temperature was measured with two iron-constantan thermocouples positioned as shown in figure 1. The estimated accuracy of the temperature measuring system was about ±1 K ($\pm2^\circ$ F).

The bearing cage speed was measured utilizing a semiconductor strain gage attached to the outer race. This technique is the same as that noted in reference 7.

The lubricant used for this investigation was superrefined naphthenic mineral oil with a viscosity of 75×10^{-6} square meter per second at 311 K (75 cS at 100° F).

Test Bearings

Test bearing specifications are listed in table I. The bearings were 75-millimeter bore, deep-groove arched outer-race ball bearings with 17.5-millimeter- (0.6875-in.-) diameter balls. The inner races and balls were made from AISI M-2 CVM steel. The outer races were SAE 52100. The two-piece machined cages were outer-race riding and were made from annealed AISI M-2 steel. One shoulder of the inner race was removed to make two of the ball bearings separable. A photograph of a separable bearing is shown in figure 2.

The geometry of the arched outer race is shown in figure 3 along with a sketch of a conventional outer race. Here r_o is the groove radius and g the distance between the two outer-race groove radius centers. Note that g is equal to the portion of the conventional outer race that is removed in forming an arched outer race. For the bearings used in this investigation, the arched profile was form ground and the outer race was made in one piece. The diametral play of these bearings was nominally 0.051 millimeter (0.0020 in.). The diametral play is defined as the total amount of radial movement allowed in the bearing. This should be distinguished from the diametral clearance, which is the diametral play plus twice the distance from the bottom of the ball to the tip of the arch when the bearing is in a radial contact position. The relation of the diametral play S_d with the diametral clearance P_d, ball diameter D, and raceway diameters d_i and d_o is shown in figure 4. Further definitions can be found in reference 17.

Measured values of diametral play along with the amounts of arching g are identified for each test bearing in table II. Also noted in table II is the theoretical transition speed range for each bearing.

PROCEDURE

Each bearing was started under a 2200-newton (500-lb) thrust load with an oil flow rate of 8×10^{-3} kilogram per second (1 lb/min). After 5 minutes at idle (700 rpm) the shaft speed was increased to 7000 rpm. After an additional 15 minutes the oil flow rate was increased 15×10^{-3} kilogram per second (2 lb/min) and the speed increased to a minimum of 16 000 rpm. Each bearing was operated at this initial test condition until temperature equilibrium was achieved. Equilibrium was assumed to have been achieved for each data point when the bearing outer-race temperature had not changed more than 1 K (2^{o} F) in 10 minutes. The oil inlet temperature was maintained at 316 K (110^{o} F).

After the initial data point was taken, the shaft speed was increased in increments of 2000 rpm while the load was held constant. The maximum Hertz stress of the conventional ball bearing at 28 000 rpm was approximately 1.7×10^{9} pascal (250 000 psi) at the outer-race-ball contact.

Two types of bearing tests were conducted. In the first, the previously described procedure was used with the oil flow rate held constant while the shaft speed was varied; in the second, the same procedure was used until the shaft speed reached 20 000 rpm, at which point the shaft speed was held constant and the oil flow rate was varied. Oil flow rate was first increased to about 4×10^{-2} kilogram per second (5 lb/min) and then decreased to about 8×10^{-3} kilogram per second (1 lb/min) in about eight increments. Data at equilibrium conditions were taken at each flow rate. As a final check point, data were then taken again at a flow rate of 15×10^{-3} kilogram per second (2 lb/min) to make certain the bearing operating characteristics had not changed.

Additionally, one bearing was tested holding the shaft speed constant at 26 000 rpm while the thrust load was varied. This was essentially a skidding test. The load was decreased from 4400 newtons (1000 lb) to 44 newtons (100 lb) in about eight increments. Since only cage speed data were taken, conditions were not maintained until equilibrium was achieved, because earlier testing had shown that the cage speed changed very little after the initial setting of the test conditions.

RESULTS AND DISCUSSION

Variable Speed Tests

The results of the variable speed tests are shown in figure 5. The outer-race temperature for each arched bearing tested (fig. 5(a)) was higher than that of the conventional ($g = 0$) bearing (8-S) over the speed range tested. With the exception of bearing 1-ARCH, the trend seems to be that as the arching is increased the outer-race temperature decreases at a given shaft speed. Unfortunately, the variation in diametral play among the bearings makes the analysis of the data more difficult. However, it is probable that the slightly higher temperature of bearing 1-ARCH was due to the very low value of diametral play (see table II). For all bearings, the outer race temperature increases quickly as the shaft speed is increased.

The measured torque of all the arched bearings was 15 to 25 percent higher in every case than that of the conventional bearing (fig. 5(b)). No definite trend with the amount of arching is apparent, although the two arched bearings with the most arching ($g =$ 0.51 mm (0.020 in.)) had slightly less torque than the other three arched bearings. The torque changed very little for any of the bearings over the speed range shown.

The bearing cage- to shaft-speed ratio (fig. 5(c)) generally decreased with increasing shaft speed. Cage speed data were not available for the conventional bearing. Again no definite trend with arching is apparent, although once more the two bearings with the most arching have slightly lower cage speed ratios. The change in speed ratios was very slight over the speed range for all bearings.

The analysis of reference 18 indicates that the cage- to shaft-speed ratio should decrease slightly with increasing speed and/or with increased arching. However, the amount of diametral play in the bearing will also affect the cage speed ratio. Therefore, in an attempt to eliminate the effect of the differences in diametral play, and to make the observed values of cage- to shaft-speed ratio more meaningful, theoretical values were computed using the computer program developed in reference 17. These calculated values were then used to determine the amount of cage slip in the bearings. Cage slip (in percent) is defined here as

$$\text{Percent cage slip} = \left(\frac{N_{calc} - N_{meas}}{N_{calc}}\right) 100$$

where

N_{calc} theoretical cage rotational speed, rpm

N_{meas} measured value of cage rotational speed, rpm

The diametral play used in the calculations for the theoretical cage speed was the measured unmounted value noted in table II.

The results are shown in figure 6 where the cage slip is plotted as a function of shaft speed for each arched bearing. The slip is relatively small for all bearings and increases with increasing shaft speed. The slip also appears to diminish as the amount of arching increases. Since the analysis (ref. 17) does not correct for change of diametral play due to bearing operating conditions, the values of cage slip should be considered approximate. Nevertheless, the values seem reasonable.

Variable Oil Flow Tests

The results of the variable oil flow tests are shown in figure 7. The outer race temperatures for the arched bearings were generally higher than those for the conventional bearing over the flow range (fig. 7(a)). Due to a test rig vibrational problem, bearings 2-ARCH and 4-ARCH could not be operated safely at 20 000 rpm; therefore, variable oil-flow type tests were not run on these bearings.

The measured bearing torques (fig. 7(b)) increased with increasing oil flow rate for all bearings tested. The arched bearings all showed a higher torque, for any given flow rate, than that of the conventional bearing. Also, the torque seems to be lower as the arching increases. The torque values for all bearings increased 75 to 100 percent with a fivefold increase in oil flow rate.

The bearing cage- to shaft-speed ratio (fig. 7(c)) changed only slightly (less than 1 percent) over the flow range for the three bearings tested. As before, the bearing with the most arching (1-ARCH) has the lowest cage speed ratio. However, this bearing also has the smallest diametral play, which probably is a contributing factor. Calculations made with the computer program based on the analysis of reference 17 indicate that for an arched bearing the cage speed ratio decreases as the diametral play decreases, with a constant amount of arching.

Variable Load Tests

The additional tests with varying thrust load, mentioned in the PROCEDURE section, were performed using bearing 4-ARCH, and the results are presented in figure 8. Shown are the cage-to-shaft speed ratio (fig. 8(a)) and percent cage slip (fig. 8(b)) plotted as functions of bearing thrust load. The slip was determined the same way as for figure 6. The change in speed ratio, while very small, was nonetheless very pronounced. Noise could be heard from the rig as the load was lowered from 1300 to 890 newtons (300 to 200 lb). The noise changed as the load was changed to 560 newtons (125 lb) and was loudest at 440 newtons (100 lb). This was about the smallest load that could be practically applied, and at this point the cage speed became somewhat unsteady. It is interesting to note that while complete thermal equilibrium was not attained in these tests the outer-race temperature did tend to decrease with decreasing load, even though the slip increased. This is consistent with the results of reference 19.

It may be concluded then from the foregoing that the arched outer race bearing will operate over a range of shaft speeds and thrust loads and that, in general, the arched bearing will exhibit a higher torque and a higher outer-race temperature than a conventional ball bearing with the same diametral play operating under the same conditions.

CONCLUDING REMARKS

Theoretically, these arched bearings operate as a two-point contact bearing at low speeds. Then, at some higher transition speed, three-point contact operation is achieved. The transition speed depends on the amount of arching, the internal clearance, and the applied load for a given size bearing. Analysis indicates that the cage- to shaft-speed ratio drops significantly at this transition. Unfortunately, it was not possible to observe this ratio change, because the transition speeds were in the range where the rig vibrational problem, mentioned in the RESULTS AND DISCUSSION section, was prevalent. Data could not be taken in this range as the shaft speed had to be increased rather quickly to achieve quiet, stable rig operation.

The results of this investigation differ somewhat from those noted in reference 16 in that the arched bearings in the present work indicated higher power losses than the conventional bearing. In reference 16 the three-point contact bearing indicated less power loss than did the two-point contact. However, it would appear that in reference 16 the diametral play of the two-point contact bearing was considerably greater than that of the three-point contact bearing. In the present work, the average diametral play of the three-point contact bearings tested was approximately the same as that of the two-point contact bearing. It is interesting to note that in the closure to the discussions in reference 16 results are presented for a three-point contact bearing operating at slightly higher temperatures than a conventional bearing during starvation tests. Also, the author of reference 16, in a discussion to reference 17, noted there were certain conditions where the three-point contact bearing did not show an advantage in power loss. Perhaps the bearings in the present work, operating at a high speed with a light load, meet those certain conditions.

Finally, it seems logical that the outer-race temperature for the arched bearings should be higher than for conventional bearings with the same play since, as mentioned in the INTRODUCTION, a considerable amount of spinning occurs at the outer-race contacts for the arched bearing. Nevertheless, the arched bearings did operate and thus could be useful to increase fatigue life at very high speeds and relatively light thrust loads.

SUMMARY OF RESULTS

An experimental investigation was conducted to determine the operating characteristics of full-scale, arched outer-race bearings and to compare the results with those of a similar conventional deep-groove bearing. The 75-millimeter bore bearings were operated up to 28 000 rpm with a 2200-newton (500-lb) thrust load.

The following results were obtained:

1. The arched outer-race bearing operated successfully over a range of shaft speeds. The bearing outer-race temperature and torque were consistently higher for the arched bearing than for a similar conventional bearing.

2. As the shaft speed was increased, the outer-race temperature and cage slip also increased, the cage- to shaft-speed ratio decreased, and the bearing torque changed very little.

3. As the flow rate was increased, the outer-race temperature decreased and the torque increased. The torque increased 75 to 100 percent for a fivefold increase in oil flow.

4. It was observed that as the amount of arching increased, the outer-race temperature and percent slip decreased. No other trends with arching were established.

5. The test results showed good agreement with the theoretical analysis.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, October 24, 1975,
505-04.

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TABLE I. - BEARING[a] SPECIFICATIONS

Bearing outside diameter, mm	130
Bearing inside diameter, mm	75
Bearing width, mm	25
Bearing internal radial clearance, mm (in.)	0.051 (0.0020)
Outer-race curvature	0.53
Inner-race curvature	0.53
Number of balls	11
Ball diameter, mm (in.)	17.5 (0.6875)
Retainer design	Two-piece machined, riveted
Retainer material	Annealed AISI M-2
Inner-race and ball material	AISI M-2[b]
Arched outer-race material	SAE 52100
Amount of arching, mm (in.)	0.13, 0.25, 0.51 (0.005, 0.010, 0.020)

[a]Tolerance grade, ABEC-5.
[b]Consumable-electrode vacuum-melted.

TABLE II. - TEST BEARING IDENTIFICATION

Bearing	Amount of arching, g, mm (in.)	Measured unmounted diametral play, mm (in.)	Theoretical transition speed, N_T, rpm
1-ARCH	0.51 (0.020)	0.028 (0.0011)	<4000
2-ARCH	.51 (.020)	.056 (.0022)	<4000
3-ARCH	.13 (.005)	.043 (.0017)	$16\,000 < N_T < 20\,000$
4-ARCH	.25 (.010)	.069 (.0027)	$12\,000 < N_T < 16\,000$
5-ARCH	.25 (.010)	.043 (.0017)	$12\,000 < N_T < 16\,000$
8-S	.00 (.000)	.051 (.0020)	-------------------

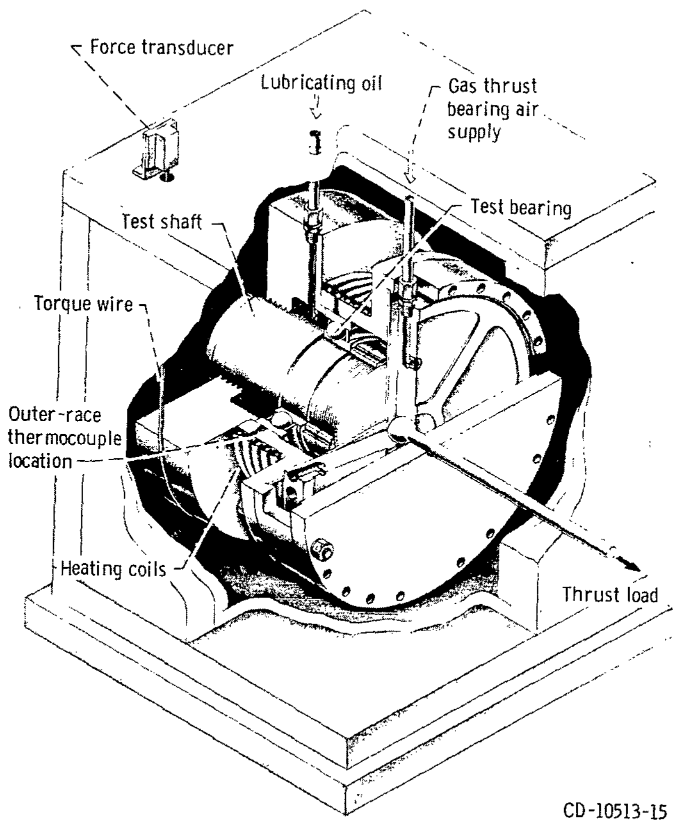


Figure 1. - Bearing test apparatus.

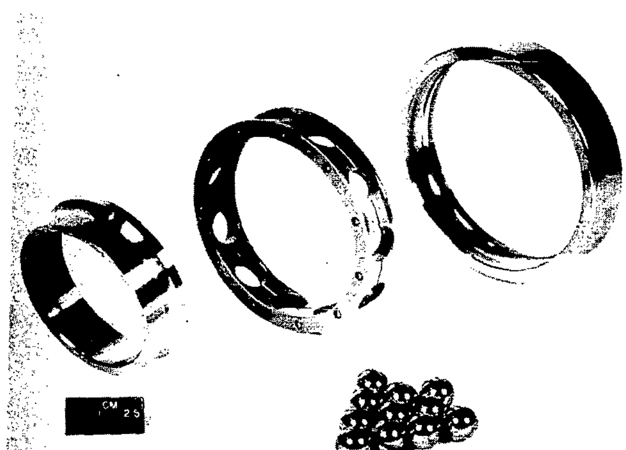
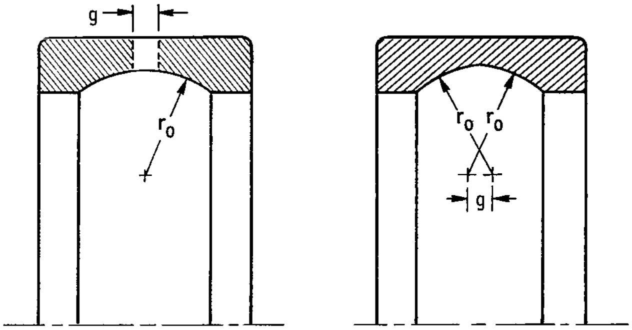


Figure 2. - Deep-groove test bearing with inner race shoulder removed; two-piece machined cage construction.



(a) Conventional. (b) Arched.

Figure 3. - Bearing outer-race geometries.

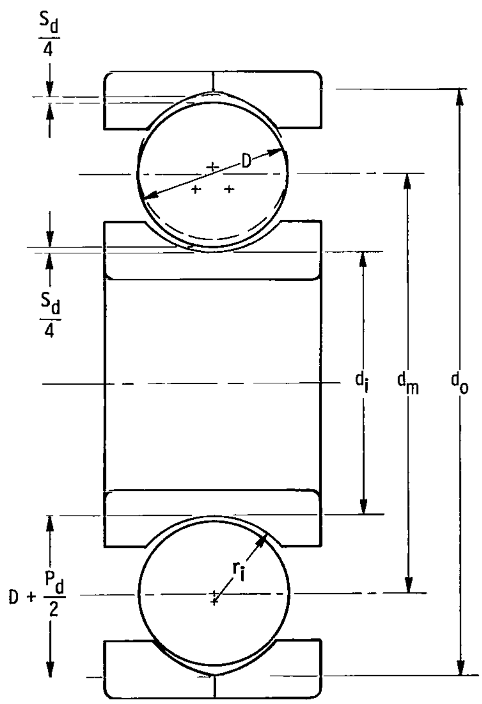
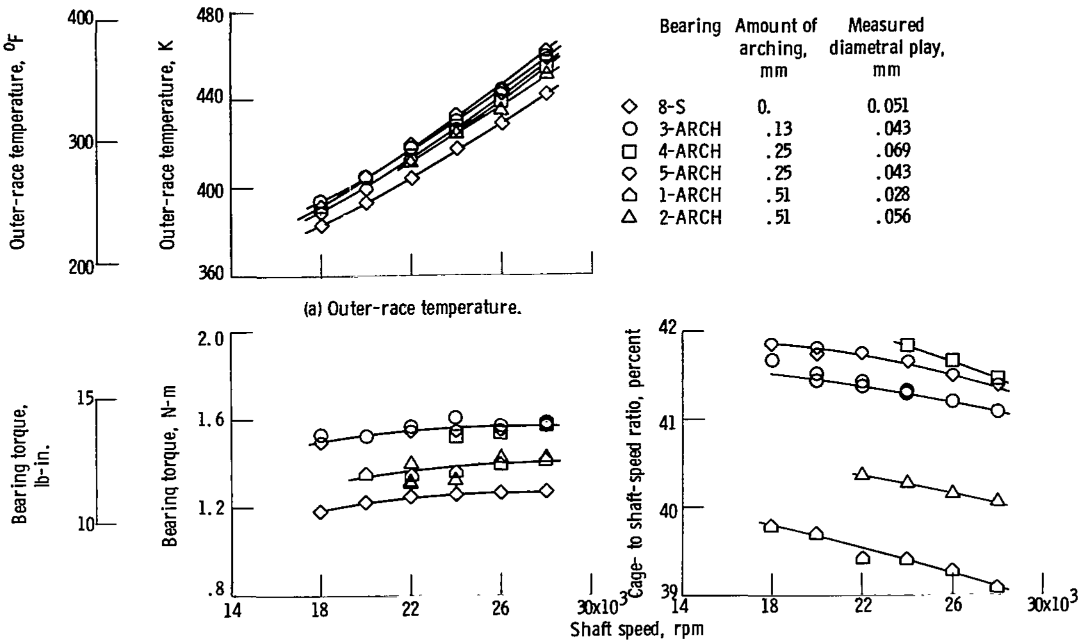


Figure 4. - Arched ball bearing in noncontacting position.



(a) Outer-race temperature.

(b) Bearing torque.

(c) Cage-to-shaft speed ratio.

Figure 5. - Bearing performance as function of shaft speed. Thrust load, 2200 newtons (500 lb); oil flow rate, 15×10^{-3} kilogram per second (2 lb/min); oil inlet temperature, 316 K (110° F).

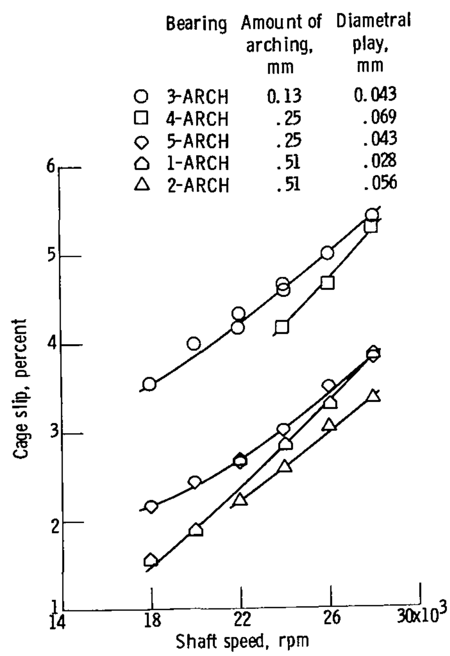


Figure 6. - Cage slip as function of shaft speed. Theoretical cage speed values computed from reference 17. Oil flow rate, 15×10^{-3} kilogram per second (2 lb/min); oil inlet temperature, 316 K (110° F); thrust load, 2200 newtons (500 lb).

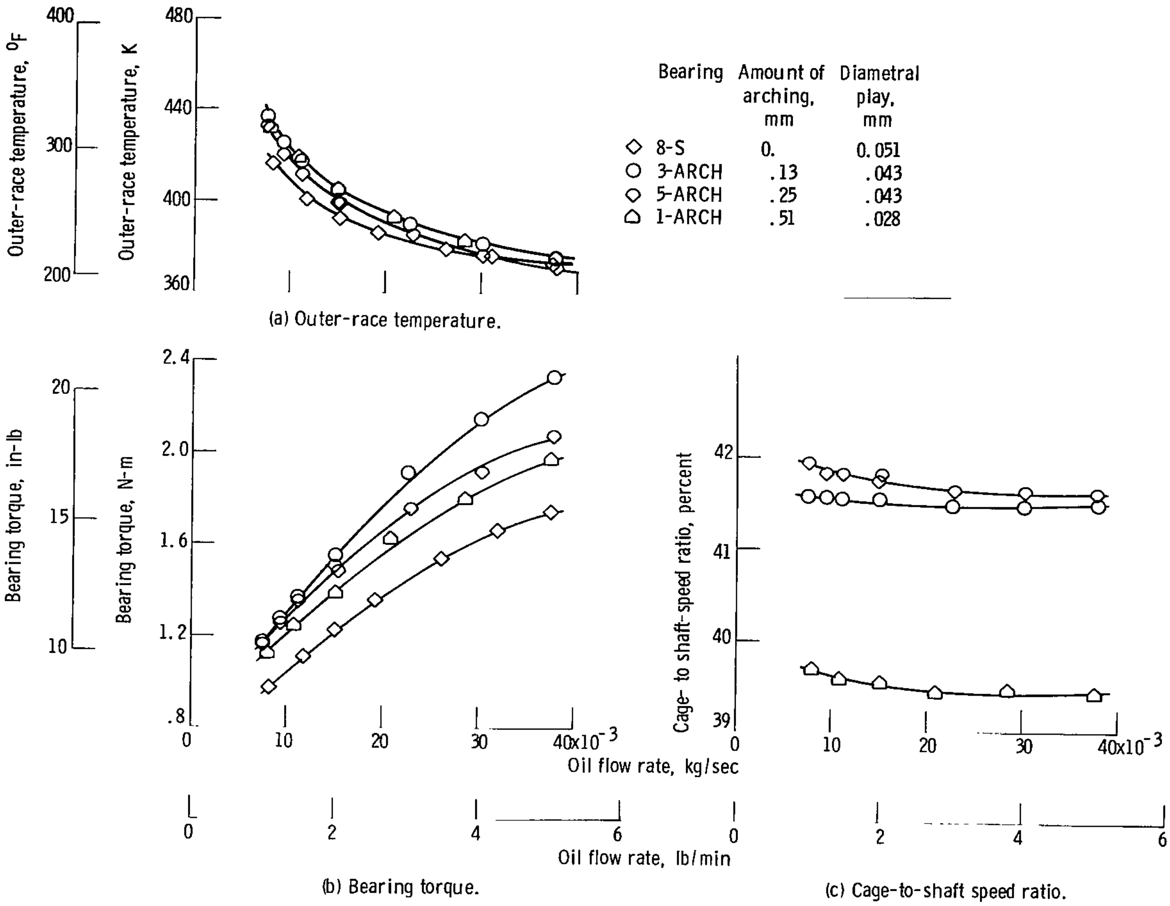
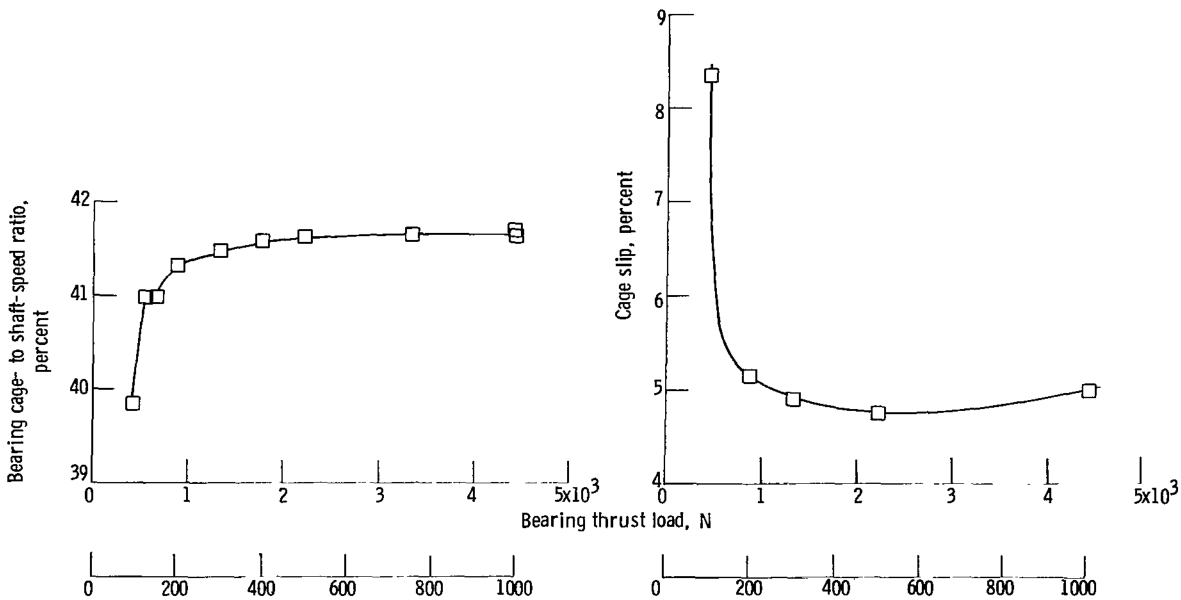


Figure 7. - Bearing performance as function of oil flow rate. Thrust load, 2200 newtons (500 lb); inlet oil temperature, 316 K (110^0 F); shaft speed, 20 000 rpm.



(a) Bearing cage- to shaft-speed ratio.

(b) Cage slip.

Figure 8. - Bearing performance as function of thrust load. Shaft speed, 26 000 rpm; oil flow rate, 15×10^{-3} kilogram per second (2 lb/min); oil inlet temperature, 316 K (110^0 F); bearing 4-ARCH; arching, 0.25 millimeter (0.010 in.).

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