NASA TECHNICAL MEMORANDUM

NASA TM X-73442

NASA TM X-73442

(NASA-TM-X-73442) PERFORMANCE OF 75 MILLIMETER-BORE ARCHED OUTER-RACE BALL BEARINGS (NASA) 23 p HC $3.50 CSCL 13I

N76-26513

Unclas

G3/37 44479

PERFORMANCE OF 75 MILLIMETER-BORE ARCHED OUTER-RACE BALL BEARINGS

by Harold H. Coe and Bernard J. Hamrock
Lewis Research Center
Cleveland, Ohio 44135

TECHNICAL PAPER proposed for presentation at Joint Lubrication Conference cosponsored by the American Society of Mechanical Engineers and the American Society of Lubrication Engineers Boston, Massachusetts, October 5-7, 1976

E-8504

PERFORMANCE OF 75 MILLIMETER-BORE ARCHED OUTER-RACE BALL BEARINGS

by Harold H. Coe and Bernard J. Hamrock

Lewis Research Center

ABSTRACT

An investigation was performed to determine the operating characteristics of 75-mm bore, arched outer-race bearings, and to compare the data with those for a similar, but conventional, deep groove ball bearing. Further, results of an analytical study, made using a computer program developed previously, were compared with the experimental data. Bearings were tested up to 28 000 rpm shaft speed with a load of 2200 N (500 lb). The amount of arching was 0.13, 0.25, and 0.51 mm (.005, .010, and .020 in.). All bearings operated satisfactorily. The outer-race temperatures and the torques, however, were consistently higher for the arched bearings than for the conventional bearing.

INTRODUCTION

Bearings in current commercial aircraft turbine engines operate in a speed range up to 2 million dN (bearing bore in mm 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 [1].[1] But, when bearings operate at high dN values, analyses show that a significant reduction in fatigue life can occur due to the high centrifugal forces developed by the balls at the outer race contact [2].

One proposed solution to the high-speed bearing problem was to reduce the mass of the ball by making it hollow, and thereby reduce the centrifugal force. Theory indicated that a significant improvement in bearing

[1] Numbers in brackets refer to references at the end of the paper.

fatigue life could be obtained at the higher dN values if the mass of the balls were reduced 50 percent below that of a comparable solid ball [3]. Therefore, both spherically hollow [4 and 5] and cylindrically hollow drilled [6 to 8] balls have been fabricated and tested. However, both the hollow and the drilled balls experienced early flexure fatigue failures during testing. Subsequent analyses showed that during these tests relatively high stresses existed at the inner surface of the hollow balls [9] at the bore of the drilled balls [10].

Other approaches to the problem of high-speed bearings have also been investigated. A parallel-hybrid bearing [11] in which a ball bearing and a fluid film bearing share the system load could be used. However, the effectiveness of the parallel hybrid bearing diminishes at high speeds because it does not attenuate the centrifugal force in the ball bearing. A series hybrid bearing [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 [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 [16] showed this design operating with a lower torque than a conventional angular contact bearing. Theoretically, an arched bearing operates like a conventional bearing at very low speeds, with each ball having two ball-race contacts. However, above some 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, and thereby reduce the load of each contact and increase the bearing fatigue life. A first-order thrust load analysis of an arched bearing design [17] indicated the possibility of significant fatigue life improvement. In particular, the results of [17] show that for a dN value of 3 million and an applied axial load of 4400 N (1000 lb) a 150-mm bore arched bearing shows an improvement in fatigue life of 306 percent over that of a similar conventional bearing. A more complete analysis [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 [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, (2) compare these arched bearing experimental results with data from a similar conventional bearing having the same diametral play, and (3) compare some of the experimental results with trends predicted by the theoretical analysis [17].

The tests were conducted with 215-series, 75-mm bore, deep groove, arched outer-race ball bearings. The amount of arching was up to 0.51 mm (0.020 in.). The bearings were operated at 2200 N (500-lb) thrust load at shaft speeds up to 28 000 rpm (2.1×10^6 dN) using oil-jet lubrication. A detailed account of the experimental portion of this paper can be found in [19].

APPARATUS AND INSTRUMENTATION

Bearing Test Rig

A cutaway view of the bearing test apparatus is shown in Fig. 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 Fig. 1. This torque was recorded continuously by a millivolt potentiometer. Estimated accuracy of the data recording system was ±0.006 N-m (±0.05 lb-in).

Bearing outer-race temperature was measured with two iron-constantan thermocouples positioned as shown in Fig. 1. The estimated accuracy of the temperature measuring system was about ±1 K ($\pm 2^{\circ}$ F).

The bearing cage speed was measured utilizing a semi-conductor strain gage attached to the outer race. This technique is the same as that noted in [7].

The lubricant used for this investigation was a superrefined naphthenic mineral oil with a viscosity of 75×10^{-6} m^2/sec at 311 K (75 cs at 100° F).

Test Bearings

Test bearing specifications are listed in table I. The bearings were

75-mm bore, deep groove arched-outer-race ball bearings, with 17.5-mm (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 bearings separable. A photograph of a separable bearing is shown in Fig. 2.

The geometry of the arched outer race is shown in Fig. 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 mm (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 relationship 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 Fig. 4. Further definitions can be found in [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 for each bearing.

PROCEDURE

Each bearing was started under a 2200-N (500-lb) thrust load with an oil flow rate of 8×10^{-3} kg/sec (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 to 15×10^{-3} kg/sec (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° F) in 10 minutes. The oil inlet temperature was maintained at 316 K (110° 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} Pa (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} kg/sec (5 lb/min) and then decreased to about 8×10^{-3} kg/sec (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} kg/sec (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 N (1000 lb) to 44 N (100 lb) in about eight increments. Since only cage speed data were taken, conditions were not maintained until full thermal 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 Fig. 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.

Theoretical values of cage-to-shaft speed ratios were calculated, using the computer program developed in [17], and the results are shown in table III. This analysis indicates that the cage speed ratio should increase slightly with increasing speed and/or with decreased arching. However, the cage speed ratio is also shown to increase with increased diametral play. 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 calculated, using the measured unmounted values of diametral play noted in table II. 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 results are shown in Fig. 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 [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 Fig. 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. This bearing also has the smallest diametral play, which is a contributing factor. These results are consistent with the trends of cage speed ratio for arched bearings shown in table III.

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 Fig. 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 Fig. 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 N (300 to 200 lb). The noise changed as the load was changed to 560 N (125 lb) and was loudest at 440 N (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 [20].

Bearing Transition Speed

Theoretically, these arched bearings operate as a two-point contact bearing at low speeds. At some higher speed a three point contact operation is achieved. The speed at which the three point contact first occurs is called the transition speed. The transition speed depends on the amount of arching, the internal clearance, and the applied load for a given size bearing. Values of transition speed for the bearings as specified in table I were determined, again using the computer program developed in [17]. This transition speed is shown in Fig. 9 as a function of the amount of arching, for three values of diametral play, at a 2200 N (500 lb) thrust load. The transition to three-point contact occurs at lower speeds as the amount of arching is increased, or the diametral play is decreased, for the constant load. To show the effect of bearing load, the transition speed is plotted in Fig. 10 as a function of thrust loads, for three values of arching g, with a diametral play S_d of .076 mm (.0030 in.). The transition speed increases as the thrust load is increased, for a given diametral play. When the arching was 0.51 mm (.020 in.) the transition speed was less than 1000 rpm for all three loads, and for all three values of diametral play. Unfortunately, experimental data could not be taken at speeds close to the transition point, because most of these speeds were in the range where the rig vibrational problem, mentioned previously, was prevalent, and quiet, stable rig operation was not possible.

It may be concluded however, 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

The results of this investigation differ somewhat from those noted in Ref. 16 in that the arched bearings in the present work indicated higher power losses than the conventional bearing. In [16] the three-point contact bearing indicated less power loss than did the two-point contact. However, it would appear that in [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 [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 [16], in a discussion to [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.

However, 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.

It should be noted here that the present results are for a 75-mm bore bearing whereas the fatigue life comparisons in [17] were for 150-mm bore bearings. The use of different bearing sizes for comparison can yield somewhat different results, but the comparisons should still be qualitatively similar. The relative importance of centrifugal effects in bearings of different sizes can be determined by comparing the ratio of D^3N^2 (D is ball diameter and N is shaft speed) to the dynamic capacity. The factor D^3N^2 is proportional to centrifugal force, and the dynamic capacity is a measure of the load capacity of the bearing. This ratio is shown in table IV for extra-light series angular contact ball bearings operating at 3 million dN. The ratio shows that the centrifugal effects are relatively more severe in small bearings when dN is kept constant. Thus the life improvement for the arched bearing will be greater for the smaller diameter bearings than for the larger ones.

SUMMARY OF RESULTS

An experimental and a theoretical 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-mm bore bearings were operated up to 28 000 rpm with a 2200-N (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.

(6) For a given size bearing, the speed at which three-point contact first occurs (transition speed) was found analytically to increase with either decreased arching, increased diametral play or increased thrust load.

REFERENCES

1. Brown, P. F., "Bearings and Dampers for Advanced Jet Engines," SAE Paper 700318, Apr. 1970.

2. Jones, A. B., "The Life of High-Speed Ball Bearings," Trans. ASME, Vol. 74, 1952, pp. 695-703.

3. Harris, T. A., "On the Effectiveness of Hollow Balls in High-Speed Thrust Bearings," ASLE Trans., Vol. 11, 1968, pp. 290-294.

4. Coe, H. H., Parker, R. J., and Scibbe, H. W., "Evaluation of Electron-Beam Welded Hollow Balls for High-Speed Ball Bearings," Journal of Lubrication Technology, Trans. ASME, Vol. 93, 1971, pp. 47-59.

5. Coe, H., Parker, R. J., and Scibbe, H. W., "Performance of 75-Millimeter-Bore Bearings Using Electron-Beam Welded Hollow Balls with a Diameter Ratio of 1.26," NASA TN D-7869, Jan. 1975.

6. Coe, H. H., Scibbe, H. W., and Anderson, W. J., "Evaluation of Cylindrically Hollow (Drilled) Balls in Ball Bearings at DN Values to 2.1 Million," NASA TN D-7007, 1971.

7. Holmes, P. W., "Evaluation of Drilled-Ball Bearings at DN Values to Three Million. I - Variable Oil Flow Tests," NASA CR-2004, Mar. 1972.

8. Holmes, P. W., "Evaluation of Driller-Ball Bearings at DN Values to Three Million. II - Experimental Skid Study and Endurance Tests," NASA CR-2005, Apr. 1972.

9. Nypan, L. J., Coe, H. H., and Parker, R. J., "Bending Stresses in Spherically Hollow Ball Bearing and Fatigue Experiments," ASME Paper 75-Lub-8, Oct. 1975.

10. Nypan, L. J., Coe, H. H., and Scibbe, H. W., "An Experimental Evaluation of the Stresses in Drilled Balls," Journal of Lubrication Technology, Trans. ASME, Vol. 97, 1975, pp. 533-538.

11. Wilcock, D. F., and Winn, L. W., "The Hybrid Boost Bearing - A Method of Obtaining Long Life in Rolling Contact Bearing Applications," Journal of Lubrication Technology, Trans. ASME, Vol. 92, 1970, pp. 406-414.

12. Anderson, W. J., Fleming, D. P., and Parker, R. J., "The Series Hybrid Bearing: A New High Speed Bearing Concept," Journal of Lubrication Technology, Trans. ASME, Vol. 94, Apr. 1972, pp. 117-124.

13. Nypan, L. J., et al., "Optimization of Conical Hydrostatic Bearing for Minimum Friction," Journal of Lubrication Technology, Trans. ASME, Vol. 94, Apr. 1972, pp. 136-142.

14. Gu, A., Eusepi, M., and Winn, L. W., "Evaluation of a Series Hybrid Thrust Bearing at DN Values to Three Million. I - Analysis and Design," NASA CR-2366, Jan. 1974.

15. Eusepi, M., and Winn, L. W., "Evaluation of a Series Hybrid Thrust Bearing at DN Values to Three Million. II - Fabrication and Testing," NASA CR-2542, June 1975.

16. Haines, D. J. and Edmonds, M. J., "A New Design of Angular Contact Ball Bearing," Proceedings of the Institute of Mechanical Engineers, Vol. 185, 1970-71, pp. 382-393.

17. Hamrock, B. J. and Anderson, W. J., "Analysis of an Arched Outer-Race Ball Bearing Considering Centrifugal Forces," Journal of Lubrication Technology, Trans. ASME, Vol. 95, 1973, pp. 265-276.

18. Hamrock, B. J., "Ball Motion and Sliding Friction in an Arched Outer-Race Ball Bearing," Journal of Lubrication Technology, Trans. ASME, Vol. 97, Apr. 1975, pp. 202-211.

19. Coe, H. H., "Experimental Performance of 75-Millimeter Bore Arched Outer-Race Ball Bearings to 2.1 Million DN," NASA TN D-8145, Jan. 1976.

20. Anderson, W. J., Macks, E. F., and Nemeth, Z. N., "Effect of Bronze and Nodular Iron Cage Materials on Cage Slip and Other Performance Characteristics of 75-Millimeter-Bore Cylindrical-Roller Bearings at DN Values to 2×10^6," NACA TN 3002, Sept. 1953.

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)	<1 000
2-ARCH	.51 (.020)	.056 (.0022)	<1 000
3-ARCH	.13 (.005)	.043 (.0017)	14 000
4-ARCH	.25 (.010)	.069 (.0027)	10 000
5-ARCH	.25 (.010)	.043 (.0017)	9 000
8-S	.00 (.000)	.051 (.0020)	------

TABLE III. - CAGE-TO-SHAFT SPEED RATIO

[Calculated using computer program from ref. 17. Bearing specifications per table I. Thrust load, 2200 N (500 lb).]

Diametral play, S_d, mm (in.)	Amount of arching, g, mm (in.)	Cage-to-shaft speed ratio, percent					
		Shaft speeds, rpm					
		18 000	20 000	22 000	24 000	26 000	28 000
0.025 (0.0010)	0.13 (0.005)	42.62	42.69	42.75	42.81	42.87	42.93
	.25 (.010)	42.31	42.35	42.40	42.45	42.50	42.55
	.38 (.015)	41.55	41.60	41.64	41.69	41.74	41.79
	.51 (.020)	40.33	40.38	40.43	40.48	40.54	40.59
0.051 (0.0020)	0.13 (0.005)	43.28	43.36	43.43	43.50	43.57	43.63
	.25 (.010)	42.98	43.04	43.09	43.14	43.20	43.25
	.38 (.015)	42.23	42.27	42.33	42.38	42.43	42.49
	.51 (.020)	41.00	41.05	41.11	41.17	41.23	41.29
0.076 (0.0030)	0.13 (0.005)	43.99	44.07	44.15	44.23	44.30	44.37
	.25 (.010)	43.70	43.76	43.82	43.87	43.93	43.99
	.38 (.015)	42.94	43.00	43.05	43.11	43.17	43.23
	.51 (.020)	41.71	41.77	41.83	41.89	41.96	42.03

TABLE IV. - EFFECT OF BEARING SIZE

[Calculated for extra-light series angular contact ball bearings operating at a constant value of 3 million dN.]

Bore diameter, d, mm	Shaft speed, N, rpm	Ball diameter, D, mm (in.)	Factor, D^3N^2	Dynamic capacity, C	Ratio, D^3N^2/C
50	60 000	8.73 (0.344)	2.39×10^{12}	5 010	4.8×10^{8}
100	30 000	14.29 (.562)	2.63	14 400	1.8
150	20 000	22.23 (.875)	4.39	31 210	1.4
200	15 000	31.75 (1.250)	7.20	54 790	1.3

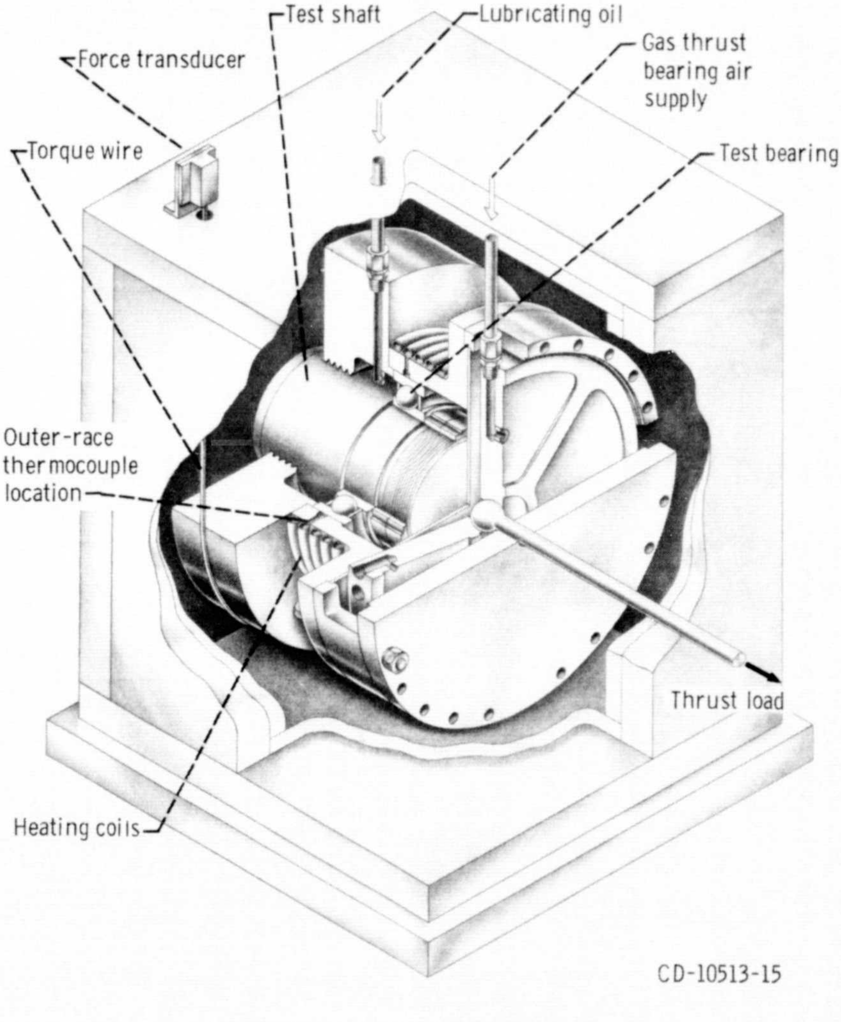


Figure 1. - Bearing test apparatus.

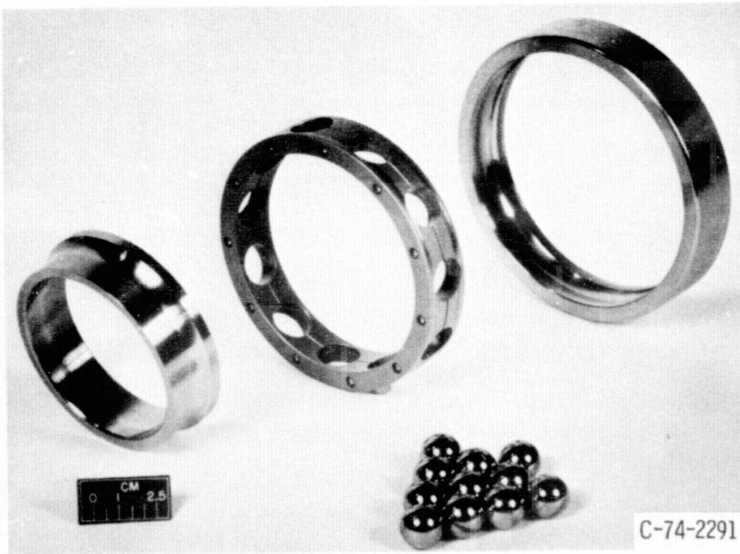
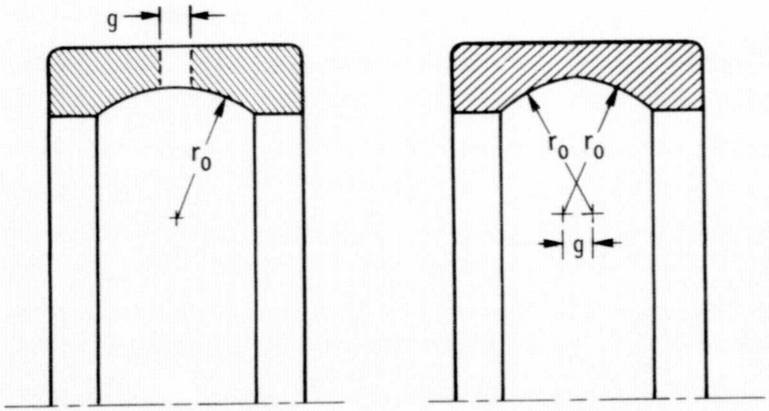


Figure 2. - Deep-groove test bearing with inner race shoulder removed; two-piece machined cage construction.



(a) Conventional.

(b) Arched.

Figure 3. - Bearing outer-race geometries.

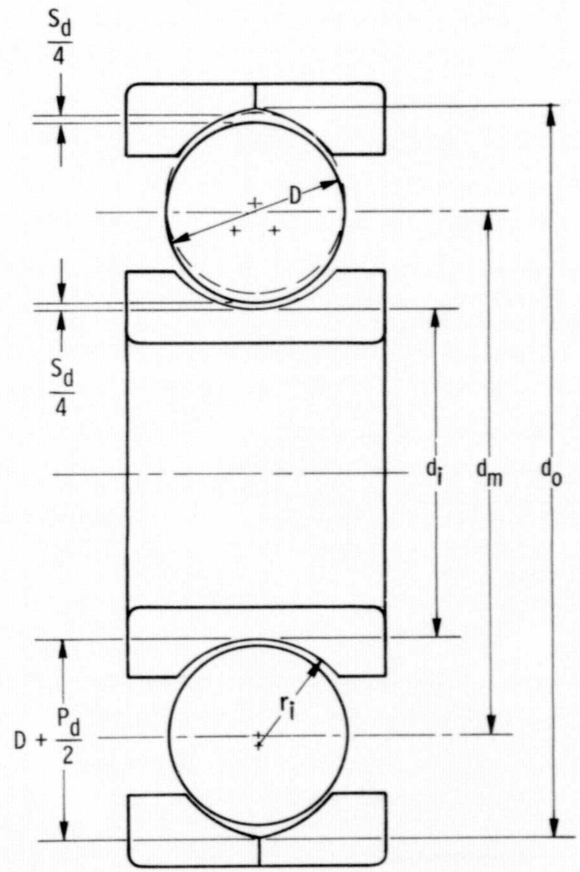
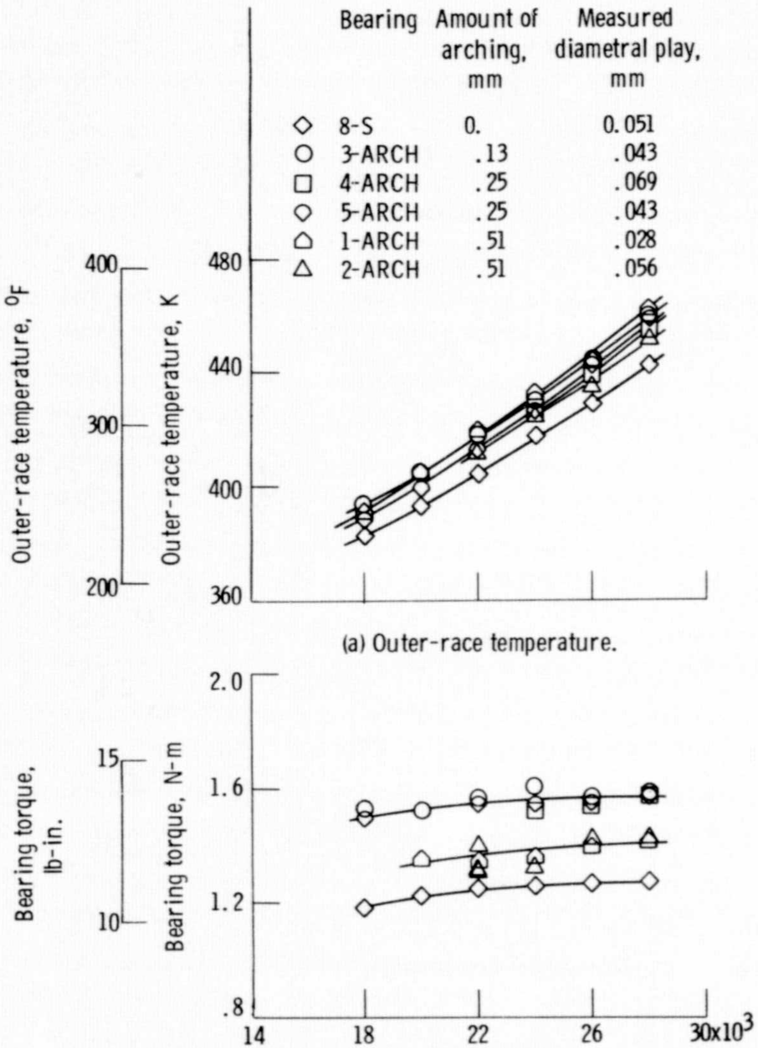


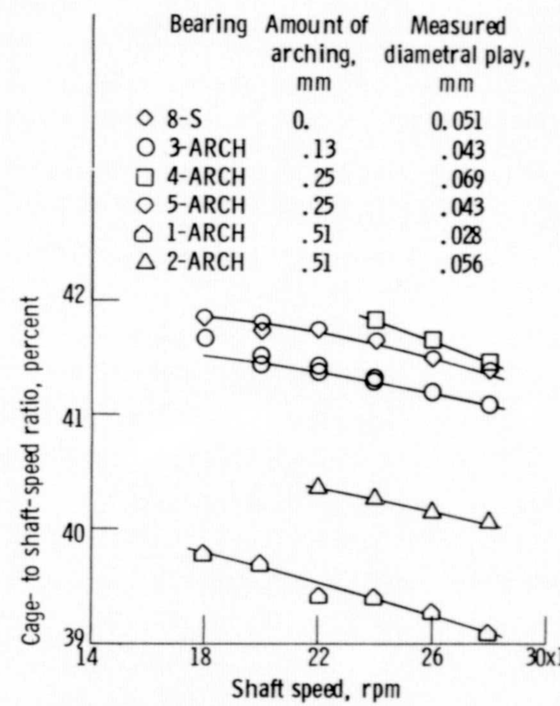
Figure 4. - Arched ball bearing in noncontacting position.



(a) Outer-race temperature.

(b) Bearing torque.

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^{0} F).



(c) Cage-to-shaft speed ratio.

Figure 5. - Concluded.

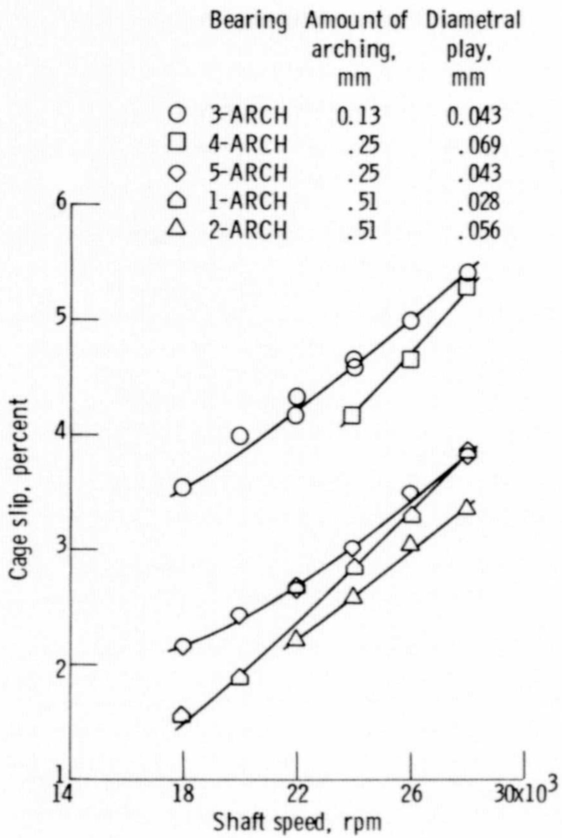


Figure 6. - Cage slip as function of shaft speed. Theoretical cage speed values computed from reference 17. Oil flow rate, 15×10^{-3} kilogram per second (2 lb/min); oil inlet temperature, 316 K (110^0 F); thrust load, 2200 newtons (500 lb).

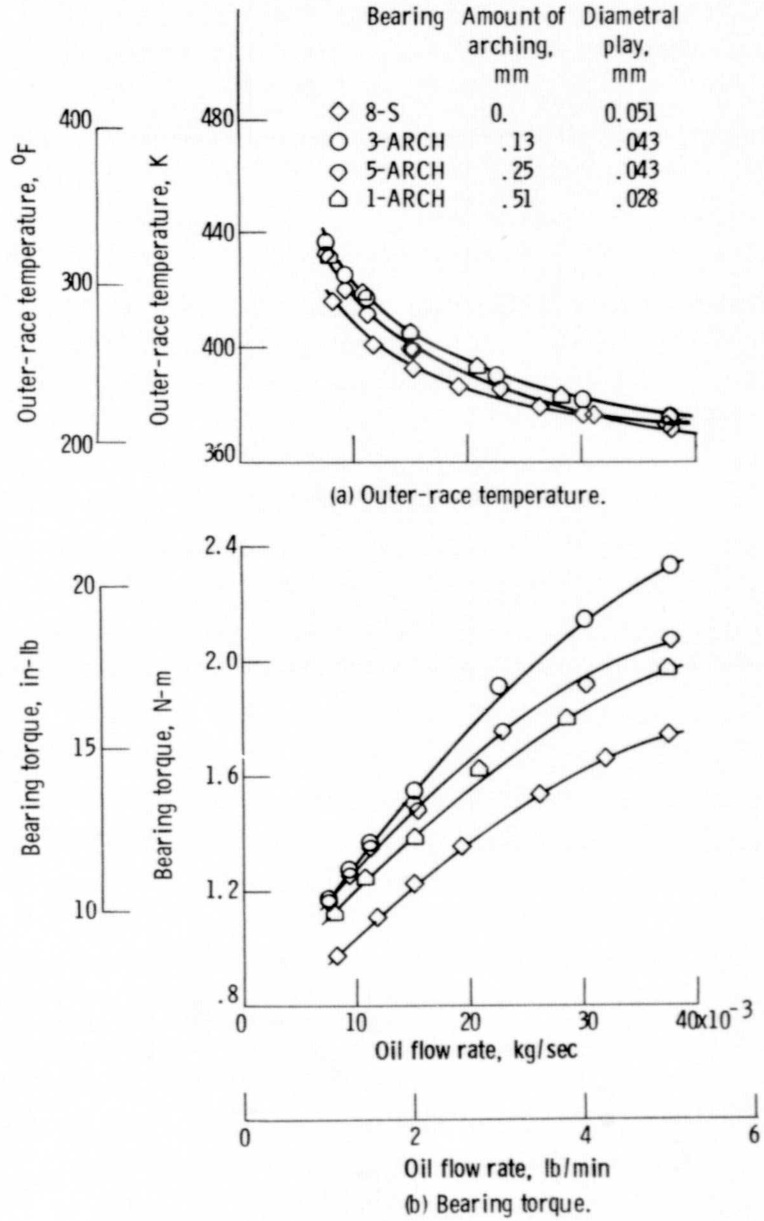
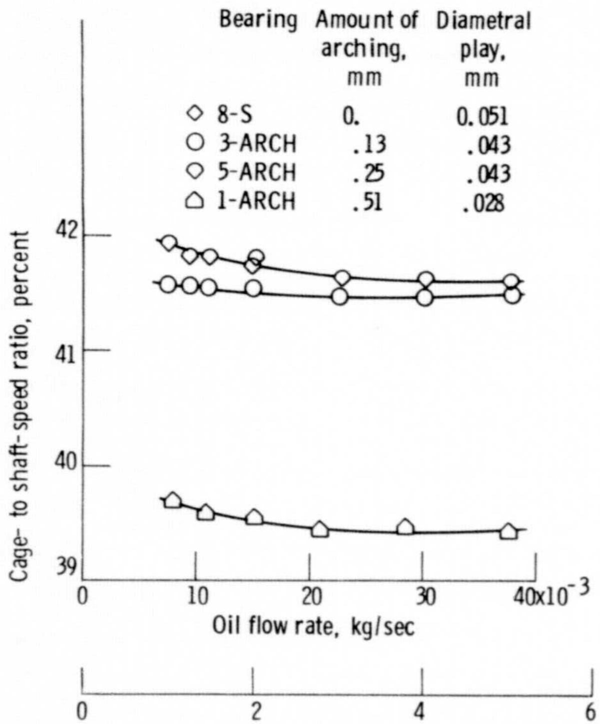
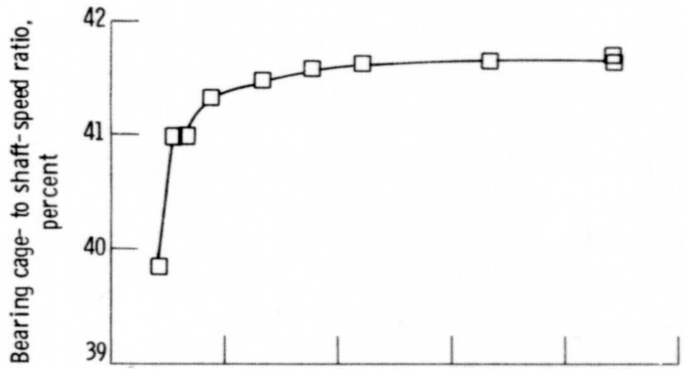


Figure 7. - Bearing performance as function of oil flow rate. Thrust load, 2200 newtons (500 lb); inlet oil temperature, 316 K (110^0 F); shaft speed, 20 000 rpm.

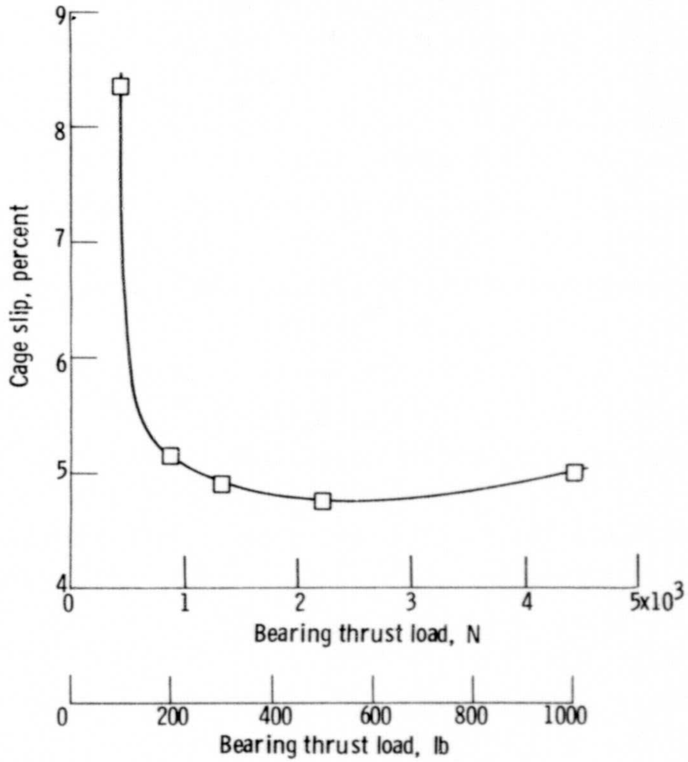


(c) Cage-to-shaft speed ratio.

Figure 7. - Concluded.



(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.).

E-8504

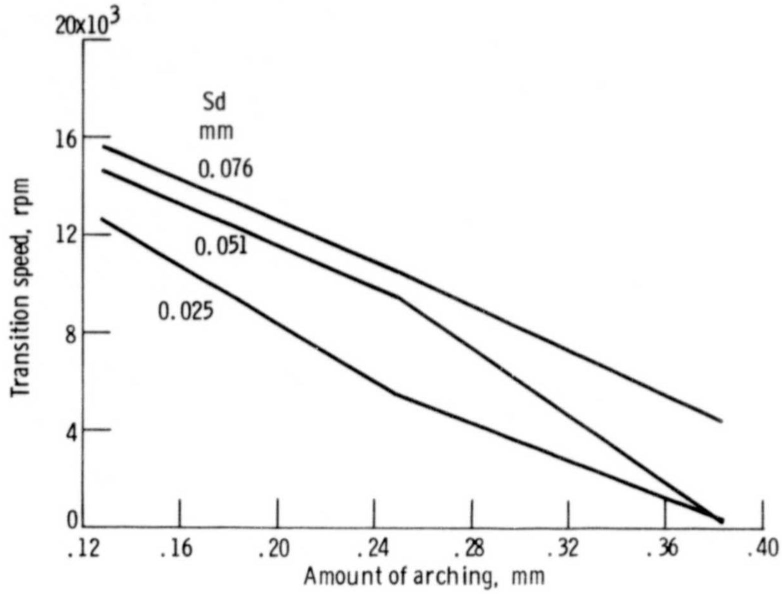


Figure 9. - Arched bearing transition speed as a function of the amount of arching, for three values of diametral play. Thrust load, 2200 N (500 lb).

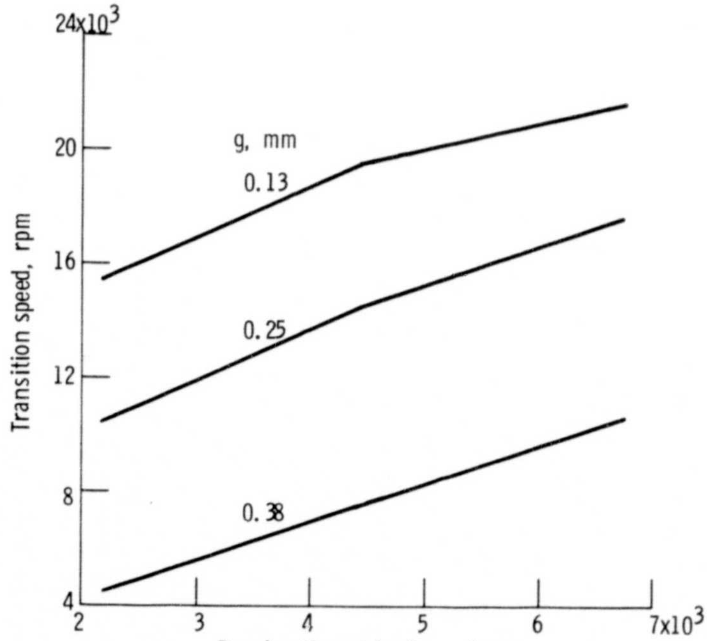


Figure 10. - Arched bearing transition speed as a function of bearing thrust load for three values of arching. Diametral play, 0.076 mm (0.0030 in.).