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EVALUATION OF CYLINDRICALLY HOLLOW (DRILLED) BALLS IN BALL BEARINGS AT DN VALUES TO 2.1 MILLION

by Harold H. Coe, Herbert W. Scibbe, and William J. Anderson

Lewis Research Center
Cleveland, Ohio 44135

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16. Abstract

An experimental investigation was conducted with 75-mm-bore bearings using hollow (drilled) balls as the rolling elements. Similar bearings using solid balls were also tested and the data compared. The bearings were operated with a 500-lb (2200-N) thrust load at shaft speeds up to 28 000 rpm (2.1×10^{6} DN). The torque and temperature of the bearings with the drilled balls (50 percent weight reduction) were significantly lower than those of the solid-ball bearings. No test bearing failures occurred with total running times of up to 107 hr.

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SUMMARY

An experimental investigation was conducted to evaluate a bearing with cylindrically hollow (drilled) balls as the rolling elements. Ball bearings having a 75-millimeter bore and either solid or drilled balls were tested with a thrust load of 500 pounds (2200 N) at shaft speeds up to 28 000 rpm or 2.1×10^{6} DN (bearing bore in mm times shaft speed in rpm). The 11/16-inch-(17.5-mm-) diameter hollow balls had a 0.42-inch-(10.7-mm-) diameter hole drilled through the center to effect a 50-percent weight reduction. Test data with the drilled-ball and solid-ball bearings were compared.

The test results showed that the drilled-ball bearing operated satisfactorily over the range of conditions investigated. The outer-race temperatures and torques of the drilled-ball bearings were significantly lower than those of the solid-ball bearings, when the lubricant flow rate was greater than 0.4 pound per minute (3×10^{-3} kg/sec). A drilled-ball bearing was still operating satisfactorily after 107 hours accumulated running time, including 66 hours at DN values of at least 1.5×10^{6}. It was concluded that the drilled-ball bearing concept offers potential for high-speed applications.

INTRODUCTION

Recent trends in gas turbine design and development have been toward engines with higher thrust-to-weight ratios, which result in a requirement for higher shaft speeds and larger shaft diameters (ref. 1). Bearings in current production aircraft turbine engines operate in the range from 1.5 to 2.0 million DN (bearing bore in mm times shaft speed in rpm). Engine designers anticipate that turbine bearing DN values will have to increase to 2.5 to 3.0 million by 1980. It is speculated that turbine engine developments after 1980 may require bearing DN values as high as 4.0 million.

When ball bearings are operated at DN values above 1.5 million, centrifugal forces produced by the balls can become significant. The resulting increase in Hertz stresses at the outer-race ball contacts can seriously shorten bearing fatigue life (ref. 2).

A possible solution to problems with high-speed bearings would be to reduce the mass of the ball and thereby reduce the centrifugal force (ref. 3). One method of reducing ball mass is to form a hollow ball by welding two hemispherical shells together (ref. 4). Tests were conducted with 75-millimeter-bore bearings with 11/16-inch-(17.5-mm-) diameter spherically hollow balls having a 0.060-inch (1.5-mm) wall (refs. 5 and 6). The hollow balls in these tests failed by flexure fatigue due to a stress concentration in the weld area. These balls had a weight reduction of 56 percent. Two other significant problems with spherically hollow balls, mentioned in references 5 and 6, are the difficulty in (1) maintaining uniform wall thickness and (2) controlling the weld penetration around the periphery of the ball. These problems would cause the balls to have an unbalance and/or a slightly different stiffness at the weld.

Another method of reducing ball mass is to machine a concentric hole through the ball. This method alleviates the possible problems of ball unbalance because the hole concentricity can be maintained very accurately. Additionally, a very smooth surface finish can be achieved, without the irregularities present in the area of a weld.

The objectives of this investigation were (1) to experimentally evaluate a ball bearing with "drilled" or cylindrically hollow balls as the rolling elements and (2) to compare torque and outer-race temperature data of a bearing with drilled balls with data of a similar bearing with solid balls over a range of operating conditions.

Tests were conducted with 75-millimeter-bore, deep-groove ball bearings using either solid or drilled balls. The bearings were operated at a thrust load of 500 pounds (2200 N) at speeds to 28 000 rpm (2.1 million DN) with either air-oil-mist or oil-jet lubrication. The bearings used 11/16-inch-(17.5-mm-) diameter balls. The drilled balls were made by electric-discharge machining (EDM) a 0.42-inch-(10.7-mm-) diameter hole through the center of a solid ball to effect a 50-percent weight reduction. Pins through the cage and the balls were used to prevent ball misorientation.

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 and cantilevered at the driven end by two oil-jet-lubricated ball bearings. The test bearing was thrust loaded by a pneumatic cylinder through an externally pressurized gas thrust bearing. The gas bearing was used so that the 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. The estimated accuracy of the data recording system was ± 0.05 pound-inch (0.006 N-m).

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

Test Bearing Lubrication System

The test bearing was lubricated by either an air-oil mist or an oil jet introduced to the bearing as shown in figure 1. The air-oil-mist generating system is shown in figure 2. The pressurized tank fed oil through the capillary tube into the air line downstream from a Venturi. A jet of high-velocity air generated by the Venturi atomized a metered flow of oil from the capillary into an air-oil mist. The mist then flowed to the test bearing lubricator through plastic tubing. After the initial tests with the air-oil mist, a recirculating oil lubrication system was installed. The oil flow rates for the air-oil mist tests ranged from 0.01 to 0.07 pound per minute (0.8×10^{-4} to 5×10^{-4} kg/sec). The flow rates for the oil-jet lubrication tests ranged from 0.4 to 2.0 pounds per minute (3×10^{-3} to 15×10^{-3} kg/sec).

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

Test Bearings and Drilled Balls

The test bearing specifications are listed in table I. The bearings were 75-millimeter-bore, deep-groove ball bearings with 11/16-inch-(17.5-mm-) diameter balls. The two-piece, machined cages were outer-race riding. One shoulder of the inner race was removed for the drilled-ball bearings to make the bearing separable. Photographs of the bearings are shown in figure 3.

A section view of the modified bearing showing details of the drilled ball and the restraining pin is presented in figure 4. Each ball has a 0.42-inch-(10.7-mm-) diameter concentric hole machined through the center. This size hole results in a weight reduction of 50 percent from that of a solid ball. Each ball is retained in the cage by a 0.125-inch-

(3.2-mm-) diameter pin made from stainless-steel tubing which prevents the edge of the hole from riding on the race groove. The pin is sized to allow the ball to turn to its contact angle. The pin is located at approximately the center of the ball pocket at the pitch diameter of the bearing.

PROCEDURE

The bearings were tested with both air-oil-mist and oil-jet lubrication. Each test bearing was run in for 1 hour with a 250-pound (1100-N) thrust load, a shaft speed of 6000 rpm, and an air-oil-mist flow rate of about 0.010 pound per minute (0.8×10^{-4} kg/sec). After run-in for the air-oil-mist tests, the flow rate was increased to 0.037 pound per minute (2.8×10^{-4} kg/sec), the thrust load was set at 500 pounds (2200-N), and the shaft speed was increased to 10 000 rpm. Each bearing was operated at this initial condition until temperature equilibrium was achieved. Equilibrium was assumed for each data point when the bearing outer-race temperature reading had not changed more than 1° F (2 K) over a 10-minute interval.

After the initial data point was taken at a speed of 10 000 rpm and a thrust load of 500 pounds (2200 N), the shaft speed was increased in increments of 2000 rpm while the load was maintained constant. The maximum hertz stress of the solid-ball bearing at 28 000 rpm was approximately 250 000 psi (1.7×10^{9} N/m^2) at the outer-race - ball contact. The lubricant flow rate was increased with the shaft speed, as shown in table II. The technique described in reference 7 was used to determine the flow rates required. Outer-race temperature and bearing torque were recorded for each shaft speed. Data points were taken until outer-race temperature equilibrium could not be attained.

Two types of tests were conducted with oil-jet lubrication. In the first type, the procedure just given was repeated with a constant oil flow rate of about 1 pound per minute (8×10^{-3} kg/sec). For the second type of tests, the procedure was used until the shaft speed was 20 000 rpm. Then, at a constant shaft speed, the oil flow rate was changed. The oil flow rate was first increased to about 2 pounds per minute (15×10^{-3} kg/sec) and then decreased to about 0.4 pound per minute (3×10^{-3} kg/sec), in about 10 increments. Equilibrium data were taken at each flow rate. A final check point was then taken at about 1 pound per minute (8×10^{-3} kg/sec) to make certain that the bearing operating characteristics had not changed.

RESULTS AND DISCUSSION

Two 75-millimeter-bore bearings with cylindrically hollow (drilled) balls were operated with a 500-pound (2200-N) thrust load over a range of shaft speeds to 28 000 rpm

(2.1 million DN), using either air-oil-mist or oil-jet lubrication. Similar bearings with solid balls were also tested under the same conditions. Outer-race temperature and torque data for the drilled-ball bearings were compared with similar data for the solid-ball bearings.

Air-Oil-Mist Lubrication Tests

The results of the bearing tests with air-oil-mist lubrication are presented in figure 5. The drilled-ball bearing 4SRH had a significantly lower outer-race temperature than the solid-ball bearing (1S). The temperature of bearing 3SRH was also lower at the lower shaft speeds but was the same as that of the solid-ball bearing at the higher speeds. The difference in the outer-race temperatures of the two drilled-ball bearings was probably due to the difference in radial internal clearance (see table III). The differences in torque were considered insignificant.

Oil-Jet Lubrication Tests

Two types of tests were conducted with oil-jet lubrication. In the first type, the oil flow rate was held constant while the shaft speed was varied. In the second type, the shaft speed was maintained constant while the oil flow rate was varied.

Variable shaft-speed tests. - The results of the variable shaft speed tests with oil-jet lubrication are presented in figure 6. The outer-race temperature and bearing torque are significantly lower for the drilled-ball bearings than for the solid-ball bearings.

The tests were run several times, and the values presented are the averages of all runs. The scatter in the temperature data was very small, as shown in figure 6. The agreement between the two drilled-ball bearings was very good over the range tested, although the torque of bearing 3SRH was deviating toward higher values at the higher shaft speeds.

During testing, a critical speed of the test shaft was encountered at about 8000 rpm (as the speed was increased from 6000 to 10 000 rpm), which caused a rise in the recorded bearing torque. This influence on bearing torque still existed somewhat at 10 000 rpm, as the values for all three bearings were slightly high at that point.

The lower outer-race temperatures for the drilled-ball bearings reflected lower surface temperatures of the balls and were probably due to two factors. These were (1) the lower ball mass, which resulted in lower heat generation, and (2) additional cooling surface, which was available with the hole through the interior of the ball. Comparison of figures 5 and 6 shows that the bearing temperatures with oil-jet lubrication averaged about 70° F (40 K) cooler than with air-oil-mist lubrication over the same speed range.

Comparison of figures 5 and 6 also shows that the bearing torques increased about 3 or 4 times with the increase in the oil flow rate of from 15 to 100 times when going from mist to jet lubrication. The bearing torque was more sensitive to shaft speed with jet lubrication because of the additional oil churning losses.

Variable oil flow tests. - The results of the variable oil flow rate tests are presented in figure 7. The torque and temperature of the drilled-ball bearing remained lower than those of the solid-ball bearing over the range tested. The changes in outer-race temperature were greater at the lower flow rates, as would be expected, since the oil was functioning as a coolant (flow rate much greater than that required for lubrication). With cooling, the outer-race temperature varied inversely with coolant flow rate. Therefore, a small change in oil flow rate will produce a greater change in outer-race temperature at low flow rates than at high flow rates.

The bearing torque doubled with a fivefold increase in oil flow rate. The rate of increase in torque was lower at the higher flow rates, as the bearing was becoming saturated with oil.

Bearing Post-Test Inspection

Visual inspection of the drilled-ball bearings showed that the balls and races were still in good condition. There was no evidence of abnormal ball-race tracking or skidding in either bearing.

The cages were also in good condition, although there was slight wear on the outside diameter in the area of the land, as shown in figure 8. The wear was not considered excessive. Wear of the pins was observed, however, as shown in figures 9 and 10. Most of the pins from bearing 3SRH showed moderate wear, but two pins had heavier wear. Pins from bearing 3SRH are shown in figure 9. Most of the pins from bearing 4SRH had light wear, but three had very heavy wear. Typical pins from bearing 4SRH are shown in figure 10. It is assumed that the wear was caused by the edge of the concentric hole. The stainless-steel tubing pins were free to turn in the cage.

The total running times for each drilled-ball bearing are shown in tables IV and V. Note that bearing 3SRH (table IV) was still operating satisfactorily after 107 hours accumulated running time, including 66 hours at DN values of at least 1.5×10^6. This compares with the fact that the same size spherically hollow ball bearings in reference 5 failed after 4 to 10 hours running time.

CONCLUDING REMARKS

It was expedient to modify an existing bearing for these tests to evaluate the drilled-ball bearing. Therefore, the bearing in this report should obviously not be considered an optimum design.

The wear of the pins could certainly have been reduced by using a hardened material. The results with the stainless-steel tubing however, do point out that pin wear could be a problem.

It should be noted that the tests reported herein were limited to the speed range of the test apparatus and not limited by the test bearings. Since the drilled-ball bearing performed quite well over the range of conditions tested, it was concluded to be not only a workable concept, but one worthy of further investigation. The drilled-ball bearing has potential for high-speed application.

SUMMARY OF RESULTS

An experimental investigation was conducted to evaluate a drilled-ball bearing. Two 75-millimeter-bore bearings having balls with a 50-percent weight reduction were operated with a 500-pound (2200-N) thrust load at speeds to 28 000 rpm (2.1 million DN). Weight reduction was achieved by drilling a 0.42-inch- (10.7-mm-) diameter concentric hole through each 11/16-inch- (17.5-mm-) diameter ball. Test results were evaluated by comparing the torque and temperature of the drilled-ball bearing with data of a similar bearing with solid balls over a range of operating conditions. The following results were obtained.

1. The drilled-ball bearings operated satisfactorily over the range of conditions investigated with both mist and jet lubrication.

2. The torques and outer-race temperatures of the drilled-ball bearing were significantly lower than those of a similar bearing with solid balls when the lubricant flow rates were greater than 0.4 pound per minute (3×10^{-3} kg/sec).

3. A drilled-ball bearing was still operating satisfactorily after 107 hours accumulated running time, including 66 hours at DN values of at least 1.5×10^{6}.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, October 14, 1970,
126-15.

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TABLE I. - BEARING SPECIFICATIONS

Bore diameter, mm	75
Radial clearance, in. (mm)	0.0016 to 0.0024 (0.041 to 0.061)
Number of balls	11
Ball diameter,[a] in. (mm)	0.6875 (17.5)
Ball and race material,[b, c]	AISI M-2
Race curvature, inner and outer	0.53
Retainer locating surface	Outer race
Retainer material	Annealed AISI M-2

[a]Hole size of drilled balls was 0.420 in. (10.7 mm).
[b]Consumable-electrode vacuum melted.
[c]Ball material for one drilled-ball bearing was SAE 52100.

TABLE II. - LUBRICANT FLOW RATE

Shaft speed, rpm	Oil flow rate[a]	
	lb/min	kg/sec
10 000	0.037	0.00028
12 000	.037	.00028
14 000	.037	.00028
16 000	.046	.00035
18 000	.062	.00047
20 000	.073	.00055
22 000	.073	.00055

[a]Oil introduced to bearing as an air-oil mist.

TABLE III. - RADIAL INTERNAL CLEARANCE MEASURED AFTER TEST

Bearing number	Ball type	Clearance	
		in.	mm
1S	Solid	0.0023	0.058
6S	Solid	.0024	.061
3SRH	Drilled	.0036	.091
4SRH	Drilled	.0024	.061

TABLE IV. - ACCUMULATED RUNNING TIME FOR BEARING 3SRH

Shaft speed, rpm	Time at speed, min		
	Air-oil-mist runs	Oil-jet runs	Total
6 000	337	128	465
10 000	241	424	665
12 000	178	111	289
14 000	213	66	279
16 000	313	78	391
17 000	----	11	11
18 000	269	78	347
20 000	3724	114	3838
22 000	----	140	140
Total	5275	1150	6425

TABLE V. - ACCUMULATED RUNNING TIME FOR BEARING 4SRH

Shaft speed, rpm	Time at speed, min		
	Air-oil-mist runs	Oil-jet runs	Total
6 000	75	288	363
10 000	78	799	877
12 000	110	250	360
14 000	9	320	329
15 000	---	8	8
16 000	36	288	324
18 000	37	321	358
20 000	77	922	999
22 000	---	52	52
24 000	---	45	45
25 000	---	46	46
26 000	---	75	75
27 000	---	54	54
28 000	---	51	51
29 000	---	14	14
Total	422	3533	3955

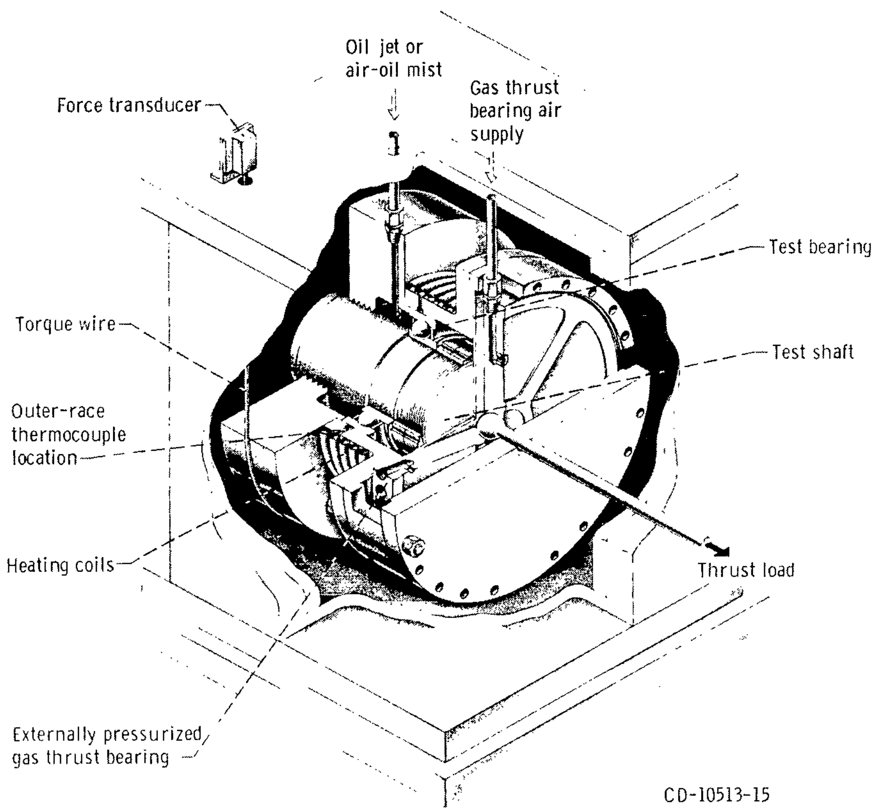


Figure 1. - Bearing test apparatus.

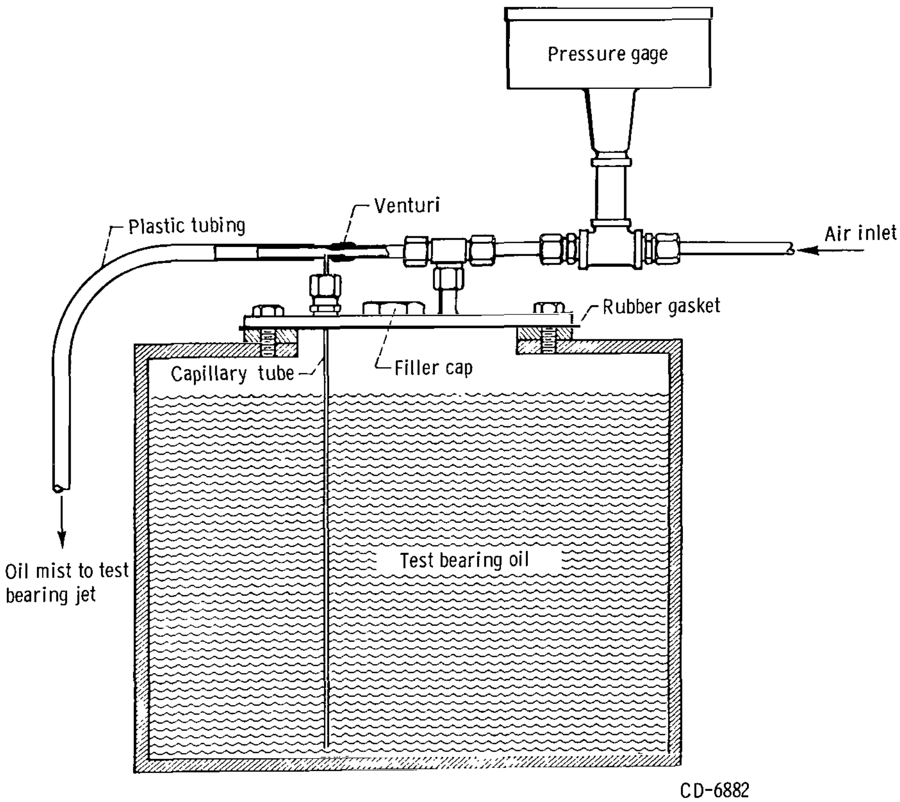
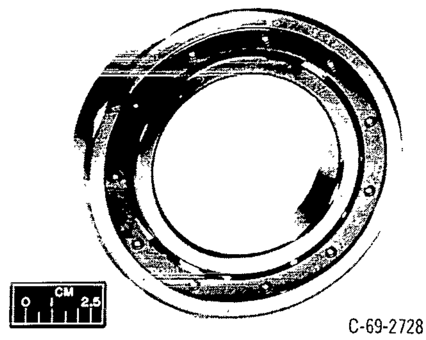
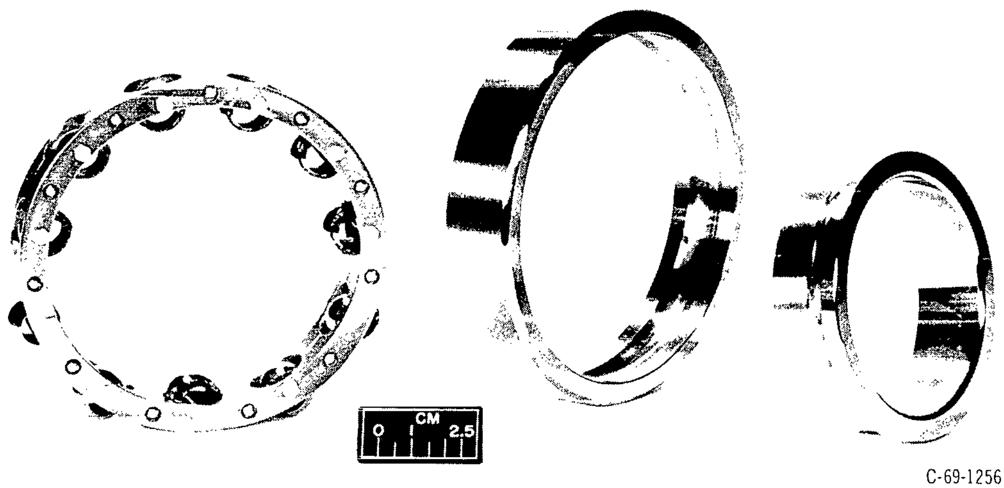


Figure 2. - Air-oil-mist test bearing lubrication system.



(a) Assembled solid-ball bearing.



(b) Disassembled cylindrically hollow (drilled) ball bearing.

Figure 3. - Test bearings. Type, deep groove; material, AISI M-2 tool steel; cage construction, two-piece machined.

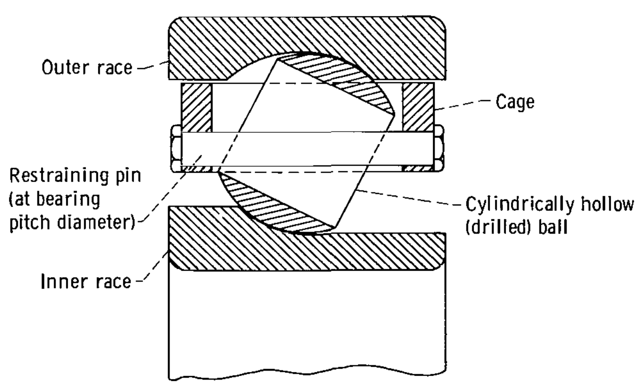


Figure 4. - Section view of test bearing with hollow ball.

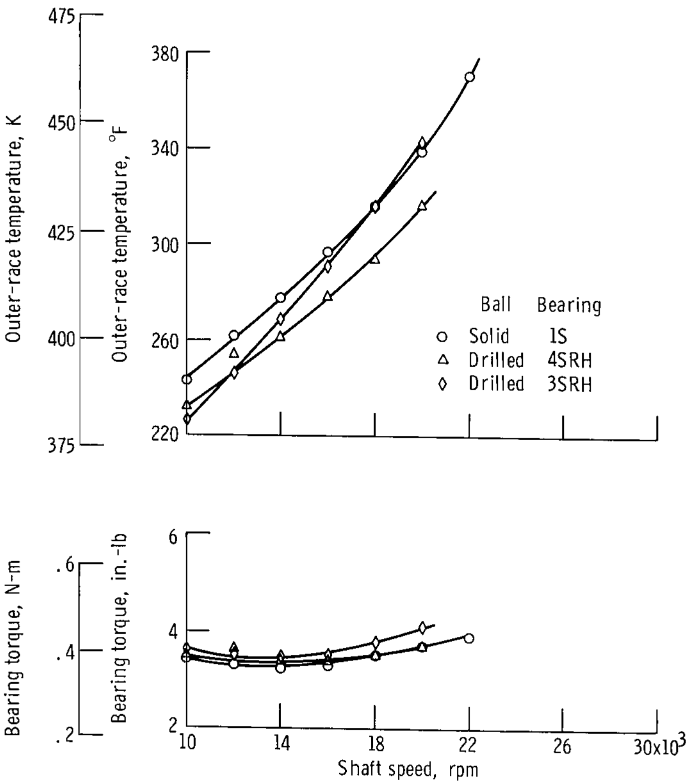


Figure 5. - Comparison of performance data of 75-millimeter-bore bearings using both solid and cylindrically hollow (drilled) balls with air-oil-mist lubrication. Oil flow rate, 0.01 to 0.07 pound per minute (0.8×10^{-4} to 5×10^{-4} kg/sec).

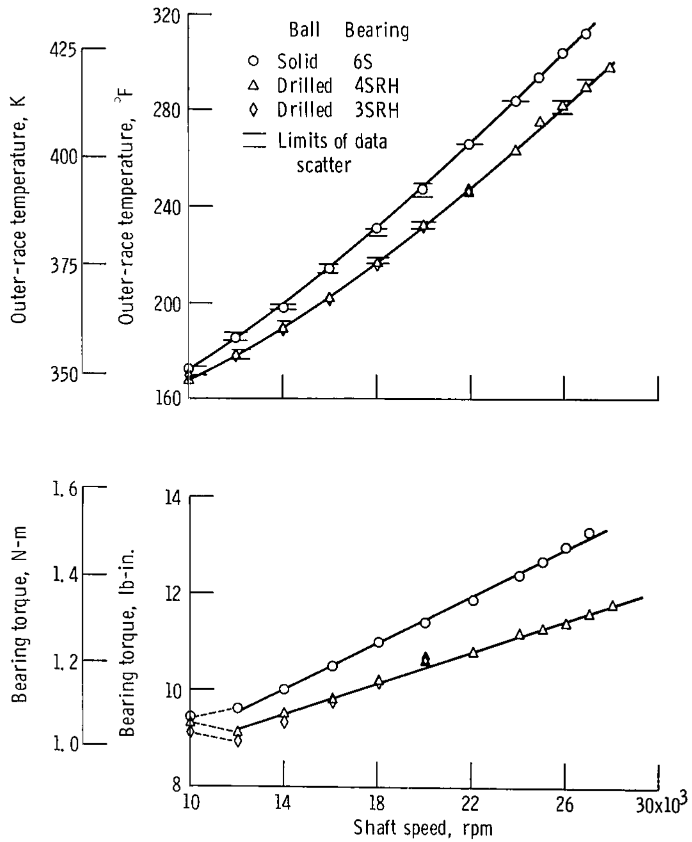


Figure 6. - Comparison of performance data of 75-millimeter-bore bearings using both solid and drilled balls, with jet lubrication. Oil flow rate, 0.95 pound per minute (7.2×10^{-3} kg/sec).

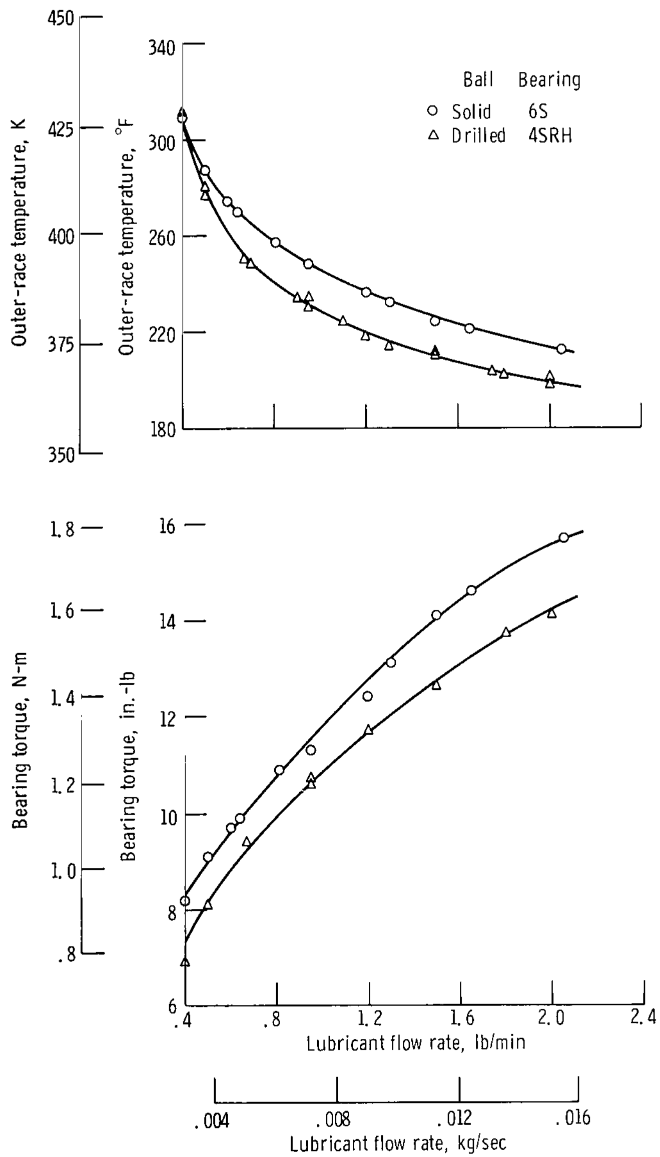
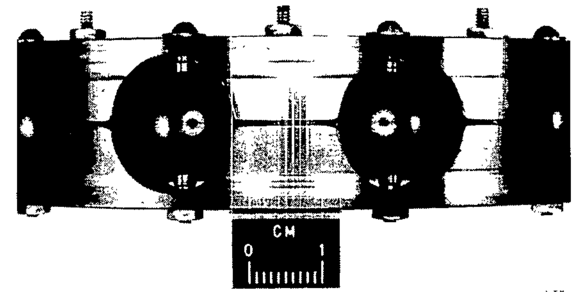


Figure 7. - Bearing performance as function of lubricant flow rate. Oil inlet temperature, 110° F (316 K); shaft speed, 20 000 rpm; thrust load, 500 pounds (2200 N).



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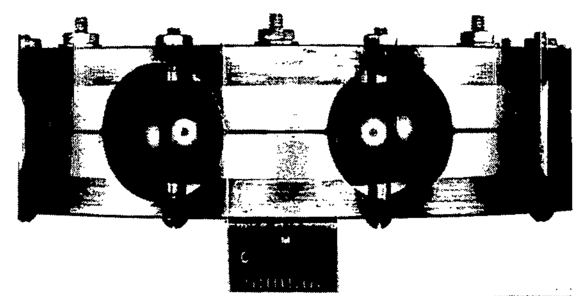
(a) Bearing 3SRH.



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(b) Bearing 4SRH.

Figure 8. - Post-test view of drilled-ball bearing retainers showing area of light wear from outer-race land contact.

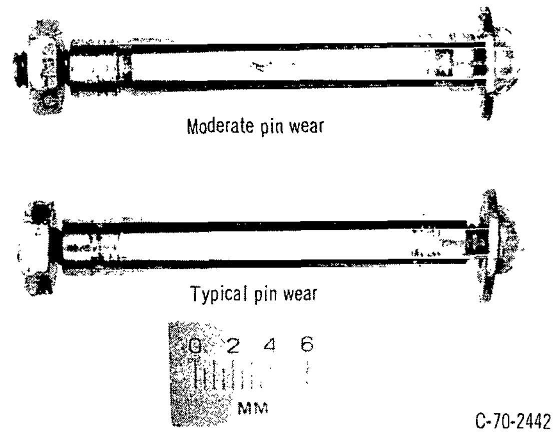

Figure 9. - Pins from bearing 3SRH after 107 hours of running, including 66 hours at DN values of at least 1.5×10^6. Maximum speed, 22 000 rpm.

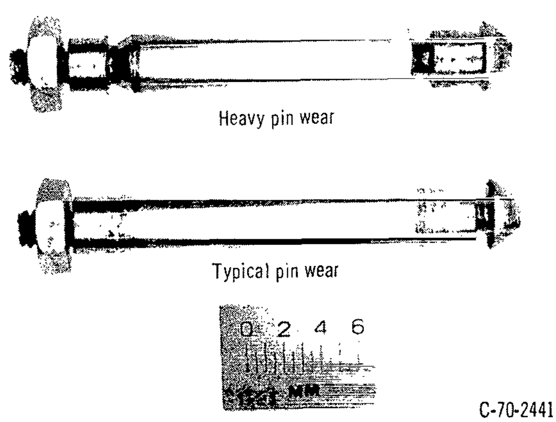

Figure 10. - Pins from bearing 4SRH after 66 hours of running, including 22 hours at DN values of at least 1.5×10^6. Maximum speed, 29 000 rpm.

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