NASA TECHNICAL NOTE

NASA TN D-7869

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PERFORMANCE OF 75-MILLIMETER-BORE BEARINGS USING ELECTRON-BEAM-WELDED HOLLOW BALLS WITH A DIAMETER RATIO OF 1.26

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NATIONAL AERONAUTICS AND SPACE ADMINISTRATION • WASHINGTON, D. C. • JANUARY 1975

1. Report No.
NASA TN D-7869

2. Government Accession No.

3. Recipient's Catalog No.

4. Title and Subtitle PERFORMANCE OF 75-MILLIMETER-BORE BEARINGS USING ELECTRON-BEAM-WELDED HOLLOW BALLS WITH A DIAMETER RATIO OF 1.26

5. Report Date
January 1975

6. Performing Organization Code

7. Author(s)
Harold H. Coe, Richard J. Parker, and Herbert W. Scibbe

8. Performing Organization Report No.
E-7961

10. Work Unit No.
501-24

9. Performing Organization Name and Address
Lewis Research Center
National Aeronautics and Space Administration
Cleveland, Ohio 44135

11. Contract or Grant No.

13. Type of Report and Period Covered
Technical Note

12. Sponsoring Agency Name, and Address
National Aeronautics and Space Administration
Washington, D.C. 20546

14. Sponsoring Agency Code

15. Supplementary Notes

16. Abstract
An experimental investigation was performed to determine the rolling-element fatigue life of electron-beam-welded hollow balls with a diameter ratio (o.d./i.d.) of 1.26 and to determine the operating characteristics of bearings using these hollow balls. Similar bearings with solid balls were also tested and the data compared. The bearings were operated at shaft speeds up to 28 000 rpm with a thrust load of 2200 N (500 lb). Ball failures during the bearing tests were due to flexure fatigue. The solid and hollow ball bearings tested showed little difference in outer-race temperatures and indicated the same bearing torque. The 17.5-mm (0.6875-in.) diameter balls were also tested in the five-ball fatigue tester and showed no significant difference in life when compared with the life of a solid ball.

17. Key Words (Suggested by Author(s))
Bearing
Ball bearing
Balls, hollow
Fatigue life
Electron beam welding

18. Distribution Statement
Unclassified - unlimited
STAR category 37 (rev.)

19. Security Classif. (of this report)	20. Security Classif. (of this page)	21. No. of Pages	22. Price*
Unclassified	Unclassified	21	$3.00

* For sale by the National Technical Information Service, Springfield, Virginia 22151

PERFORMANCE OF 75-MILLIMETER-BORE BEARINGS USING ELECTRON-BEAM-WELDED HOLLOW BALLS WITH A DIAMETER RATIO OF 1.26

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SUMMARY

These experiments compared the rolling-element fatigue life of electron-beam-welded hollow balls with 1.8-millimeter (0.070-in.) wall thickness with previously run solid balls and 1.5-millimeter (0.060-in.) wall thickness hollow balls. Also, the operating characteristics of ball bearings using 17.5-millimeter (0.6875-in.) diameter hollow balls were compared with similar bearings using solid balls.

The 75-millimeter-bore bearings were tested at a 2200-newton (500-lb)thrust load at shaft speeds up to 28 000 rpm or 2.1×10^{6} DN (bearing bore in millimeters times shaft speed in rpm). Each of the AISI M-50 steel hollow balls used in these bearings had a wall thickness of 1.8 millimeters (0.070 in.). These balls weighed 50 percent less than similar solid balls.

The fatigue life of the hollow balls, determined in the NASA five-ball fatigue tester at a maximum Hertz stress of 4.8×10^{9} pascal[1] (700 000 psi) was not significantly less than that of the previously run solid balls of the same material. The ball failures resulted from classical subsurface fatigue and did not occur in the weld area.

The results of the full-scale bearing tests showed that the outer-race temperatures of the bearings with the hollow balls were slightly lower than those with the solid balls. However, the bearing torque was about the same for all conditions tested for both the solid and the hollow ball bearings. The ball failures during the bearing tests were due to flexure fatigue. Most of the tracks on the balls that were examined were directly on the weld.

[1]Pascal = newton per square meter.

INTRODUCTION

Trends in gas turbine design have resulted in a requirement for higher shaft speeds and larger shaft diameters (ref. 1). Although 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), future engines may require bearings that can operate at DN values of 3 million or higher. However, when bearings are operating at high DN values, the balls produce significant centrifugal forces at the outer race. The resulting high Hertz stress at the outer-race contacts can shorten bearing fatigue life (ref. 2).

A possible solution to the high speed bearing problem is to reduce the mass of the ball and thus reduce the centrifugal force (ref. 3). One method of reducing the mass is to make the ball hollow. By reducing the mass 50 percent below that of a comparable solid ball, the bearing fatigue life might be improved by a factor of five for a lightly loaded, high speed application (ref. 3). A hollow ball can be fabricated by welding two hemispherically formed shells (refs. 4 and 5).

Electron-beam-welded hollow balls were evaluated for potential use as bearing balls in references 5 and 6. Also, some operating characteristics of a full-scale bearing using hollow balls were determined experimentally in reference 6. The results of these tests showed that the hollow ball is susceptible to flexure failure if the wall is sufficiently thin. The 12.7-millimeter (0.500-in.) diameter hollow balls (diameter ratio 1.67; 21.7-percent weight reduction) operated satisfactorily in a 75-millimeter bore bearing with no failures. The 17.5-millimeter (0.6875-in.) diameter hollow balls (diameter ratio, 1.21; 56.5-percent weight reduction) operated satisfactorily in a 75-millimeter bearing for only a short time until ball failures occurred. Subsequent examination revealed that these balls had failed by flexure fatigue in the area of the weld.

The 17.5-millimeter (0.6875-in.) diameter balls tested in reference 6 had a 1.5-millimeter (0.060-in.) wall thickness. However, the wall thickness can be increased by 17 percent to 1.8 millimeters (0.070 in.) and a diameter ratio of 1.26, and the ball will still weigh significantly less (50 percent) than a comparable solid ball. This investigation (1) determines the rolling-element fatigue life of hollow balls with a 1.8-millimeter (0.070-in.) wall thickness, (2) compares the hollow-ball life with the life of solid balls, (3) determines experimentally the operating characteristics of bearings using those hollow balls, and (4) compares the hollow-ball results with data from a similar bearing with solid balls.

The NASA five-ball fatigue tester was used to determine the fatigue life of the hollow balls. Tests were conducted at a maximum Hertz stress of 4.8×10^9 pascal (700 000 psi) at a shaft speed of 9700 rpm and a contact angle of 34.5°.

The bearing tests were conducted with 215-series, 75-millimeter bore, deep-groove ball bearings using both hollow and solid balls. The bearings were operated with 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

Five-Ball Fatigue Tester

A NASA five-ball fatigue tester was used for the hollow-ball fatigue tests. This apparatus (fig. 1) is identical to that used in reference 5 and is described in detail in reference 7. A fatigue tester consists essentially of an upper test ball pyramided on four lower support balls that are positioned by a separator and are free to rotate in an angular-contact raceway. Load is applied to the upper test ball through a vertical shaft that drives the ball assembly. In operation the upper test ball is analogous to the inner race of a bearing, while the separator, lower support balls, and the raceway function like the cage, balls, and outer race in a bearing. The axis of rotation of the upper test ball is fixed upon its insertion into the fatigue tester. However, the positioning of the ball is such that the orientation of the weld of the hollow ball is random.

Bearing Test Rig

A cutaway view of the bearing test apparatus is shown in figure 2. 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 2. 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 2. The estimated accuracy of the temperature measuring system was about ±1 K (±2 F^{0}).

The lubricant used for this investigation was a 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 and Hollow Balls

The hollow balls were fabricated by electron-beam welding two hemispheres of AISI M-50 consumable-electrode vacuum-melted (CVM) steel. The wall thickness of the finished ball was 1.8 millimeters (0.070 in.), which results in a weight reduction of 50 percent from that of a solid ball. A chemical analysis and heat-treatment schedule of the M-50 material is given in table I. Balls from a single batch of M-50 steel were used for both the five-ball fatigue tests and the bearing tests.

Test bearing specifications are listed in table II. The bearings were 75-millimeter bore, deep-groove ball bearings with 17.5-millimeter (0.6875-in.) diameter balls. The races and solid balls were made from AISI M-2 CVM steel. 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 the hollow ball bearings separable. A photograph of the bearing is shown in figure 3.

PROCEDURE

Fatigue Tests

Fatigue tests were conducted in the five-ball fatigue tester at a maximum Hertz stress of 4.82×10^{9} pascal (700 000 psi). The drive-shaft speed was 9700 rpm, and the contact angle (β in fig. 1) was 34.5°. Tests were run continuously for 300 hours, or until failure occurred in either the hollow upper test ball or a solid lower support ball. The outer-race temperature stabilized at about 336 K (145° F) with no heat added. The 17.5-millimeter (0.6875-in.) diameter hollow balls had a Rockwell C hardness of 61.5 to 62.5 and were run with 12.7-millimeter (0.500-in.) solid AISI M-50 lower support balls with a Rockwell C hardness of 64 to 64.5. The lubricant was superrefined naphthenic mineral oil.

Bearings Tests

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 speed was increased to 16 000 rpm, and the oil flow rate increased to 15×10^{-3} kilogram per second (2 lb/min). 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 solid 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 was 20 000 rmp, 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.

Posttest Inspection

After the fatigue tests the hollow balls with failures were cleaned and examined. The balls were electropolished to determine the location of the weld, the orientation of the weld relative to the ball track, and the proximity of the spall to the weld area. Some balls were sectioned through and parallel to the track. The sections were polished and photomicrographs were taken. After testing, the bearings were disassembled, cleaned, and inspected for damage.

RESULTS AND DISCUSSION

Fatigue Tests

The results of rolling-element fatigue tests with 17.5-millimeter (0.6875-in.) diameter electron-beam-welded hollow balls with a 1.8-millimeter (0.070-in.) wall thickness are shown in figure 4. The 10-percent life of these hollow balls is slightly less than that of the solid balls (ref. 8), but the differences are statistically insignificant, since the solid ball data fall within the 90 percent confidence bands of the hollow ball data, as determined by methods of reference 9.

Posttest Inspection of Fatigue Balls

The eleven 17.5-millimeter (0.6875-in.) diameter hollow balls that had been tested in the five-ball fatigue tester were electropolished to determine the position of the weld. On all 11 balls the running track that was in contact with the lower balls crossed the weld. Eight of the balls had fatigue spalls located randomly with respect to the weld, and no spalls were located directly on a weld. Tests with the remaining three balls were suspended at the 300 hour cutoff time without failure.

Four of the balls that failed and two of the balls that did not fail were sectioned parallel to and through the running track and etched. No cracks that could be related to the weld area were observed in any of the sections. The weld area in these balls was nearly identical in appearance to the 17.5-millimeter (0.6875-in.) diameter balls of reference 6, and the weld bead had formed a similar stress concentration on the inside surface of each ball.

The fatigue spalls were the result of classical rolling-element fatigue and originated near the outer surface with random distribution around the running track. It is concluded (1) that, under the load and stress conditions present in the five-ball fatigue tester, the hollow balls with 1.8-millimeter (0.070-in.) wall thickness did not fail due to flexure of the wall as did the 1.5-millimeter (0.060-in.) wall thickness balls of reference 6 and (2) that the weld area had a fatigue strength as good as the parent material, since no failures occurred in the weld area. This result agrees well with the results of reference 5. A summary of the hollow ball tests is shown in table III.

Bearing Tests

The results of the variable speed tests are shown in figure 5. The outer-race temperature tended to be lower for the hollow ball bearings over the entire speed range. As expected, larger differences occurred at the higher speeds. No significant differences in bearing torque were observed between the solid and hollow bearings, however, over the same speed range.

Results of the variable oil-flow tests are shown in figure 6. Outer race temperatures tended to be the same or just slightly lower for the hollow ball bearings over the flow range. The bearing torques were about the same for all bearings at each flow rate over the flow range (fig. 6(b)). Torque values doubled with the fivefold increase in oil flow rate.

For comparison analytical predictions of bearing performance were obtained using the computer program described in reference 10. This program is capable of calculating the thermal and kinematic performance of high-speed ball bearings. A comparison of the experimental data from solid ball bearing 8-S with the computed values is shown in figure 7. The temperature of the bearing housing and the test shaft, at both the oil-inlet and the oil-outlet ends, are required input for the computer program. Successive assumed values of these end temperatures allowed the computer predicted values of outer race temperature to become very close to the actual experimental data. The accompanying values of predicted bearing torque, however, were only about one-half the experimental values, although the trends seemed to be correct.

Bearing Posttest Inspection

Inspection of the bearings revealed extensive damage to some of the hollow balls. The total running times and damage to the hollow balls of each bearing are summarized in table IV.

The damaged balls were electropolished to determine the orientation of the weld relative to the running track. All of these tracks were virtually on top (within 10°) of the weld. This result indicated a preferential rotational axis and is probably related to the effect of the weld bead on the moment of inertia of the ball. A photograph of a typical damaged ball is shown in figure 8. The spall is located directly in the weld area.

Two of the balls were sectioned as shown in figure 9: One ball was sectioned parallel to and through the ball track (fig. 9(a)); the other ball was sectioned at right angles to the ball track (fig. 9(b)). Note the surface spalls, the irregularity of the weld bead, and the cracks across the weld. These specimens were polished and examined further; the results are shown in figure 10. These results are very similar to those obtained in reference 6, the main difference being that reference 6 showed most of the ball tracks crossing the weld. It was concluded that the present failures, like those in reference 6, were due to flexure fatigue.

CONCLUDING REMARKS

The 1.8-millimeter (0.070-in.) thick wall ball showed promise in the five-ball fatigue tests, but still failed by flexure when tested in an actual bearing. Because the number of ball stress cycles in the bearing tests was in the same range as those of the five-ball fatigue tests (100×10^{6} to 500×10^{6} stress cycles), it was concluded that the difference in results was due to the difference in the manner of ball loading in the two tests.

In the five-ball rig the upper test ball is loaded between a ring (the end of the drive shaft) and the four lower support balls. This is a considerably different loading arrangement from that of the normal contact loads in a ball bearing. Further, the normal load in the contact of the upper ball and each lower ball is 645 newtons (145 lb) for the conditions of these tests and of those of reference 6. The normal load in the contact of the ball with the outer race of the 75-millimeter-bore bearing at the 28 000 rpm, 2200-newton (500-lb) thrust load test condition is 1520 newtons (342 lb). It is apparent that the bearing tests impose the more severe stress condition at the inner surface of the ball.

It should be noted, however, that the fact that the balls with a 1.5-millimeter (0.060-in.) wall tested in the five-ball rig in reference 6 failed by flexure fatigue, while balls with a 1.8-millimeter (0.070-in.) thick wall, tested in the present work at identical conditions, did not fail by flexure, implies that the thicker wall significantly reduced the inner surface stresses.

Finally, based on the examination of the ball failures experienced in the bearing tests, it may be concluded that a diameter ratio of 1.26 results in a wall that is too thin for use as a bearing ball.

SUMMARY OF RESULTS

An experimental investigation was conducted to determine the operating characteristics of a full-size bearing using 17.5-millimeter (0.6875-in.) diameter hollow balls with a 1.8-millimeter (0.070-in.) wall (50 percent less weight than comparable solid balls), to determine the rolling-element-fatigue life of these hollow balls, and to compare the results with those of a solid ball and a similar hollow ball with a 1.5-millimeter (0.060-in.) wall thickness (56 percent less weight than comparable solid balls). The 75-millimeter-bore bearings were operated up to 28 000 rpm with a 2200-newton (500-lb) thrust load. The results were compared with data from a similar bearing using solid balls. The following results were obtained.

1. The hollow-ball failures in the bearing tests were due to flexure fatigue. Most of the tracks on the balls examined were directly on the weld.

2. The temperatures of the outer race were consistently lower for the bearings with hollow balls; however, the bearing torque was approximately the same at each condition tested for both the solid and the hollow ball bearings.

3. The rolling-element-fatigue life of the balls with 1.8-millimeter (0.070-in.) wall thickness, at 4.8×10^{9}-pascal (700 000-psi) maximum Hertz stress in a five-ball fatigue

tester, was not significantly less than previously run AISI M-50 solid steel balls. The ball failures were the result of classic subsurface fatigue and were not due to flexure fatigue.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, September 27, 1974,
501-24.

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TABLE I. - CHEMICAL ANALYSIS AND HEAT TREATMENT OF AISI M-50 CONSUMABLE-ELECTRODE VACUUM-MELTED STEEL HOLLOW BALLS

[Nominal Rockwell C hardness, 62±0.5.]

(a) Chemical analysis

Element	Content, wt. %	Element	Content, wt. %
Carbon	0.813	Vanadium	1.10
Silicon	.21	Molybdenum	4.18
Manganese	.25	Cobalt	.01
Sulphur	.002	Nickel	.08
Potassium	.009	Copper	.06
Tungsten	.03	Iron	Balance
Chromium	4.08		

(b) Heat treatment

Anneal after electron-beam welding	Heat to 1103 K (1525° F) in spent cast iron chip; hold for 8 hr; furnace cool to 811 K (1000° F) at 11 K/hr (20 F°/hr)
Austenitize	Preheat in salt bath at 1116 K (1550° F) for 6 min; austenitize in bath at 1394 K (2050° F) for 6 min
Quench	Molten salt to 811 K (1000° F) for 6 min
Temper	811 K (1000° F) for 120 min; repeat for second and third temper

TABLE II. - 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
Race and solid ball material	AISI M-2[b]
Hollow ball material	AISI M-50[b]
Hollow ball wall thickness, mm (in.)	1.8 (0.070)
Hollow ball outside/inside diameter ratio	1.26

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

TABLE III. - SUMMARY OF HOLLOW BALL TESTS

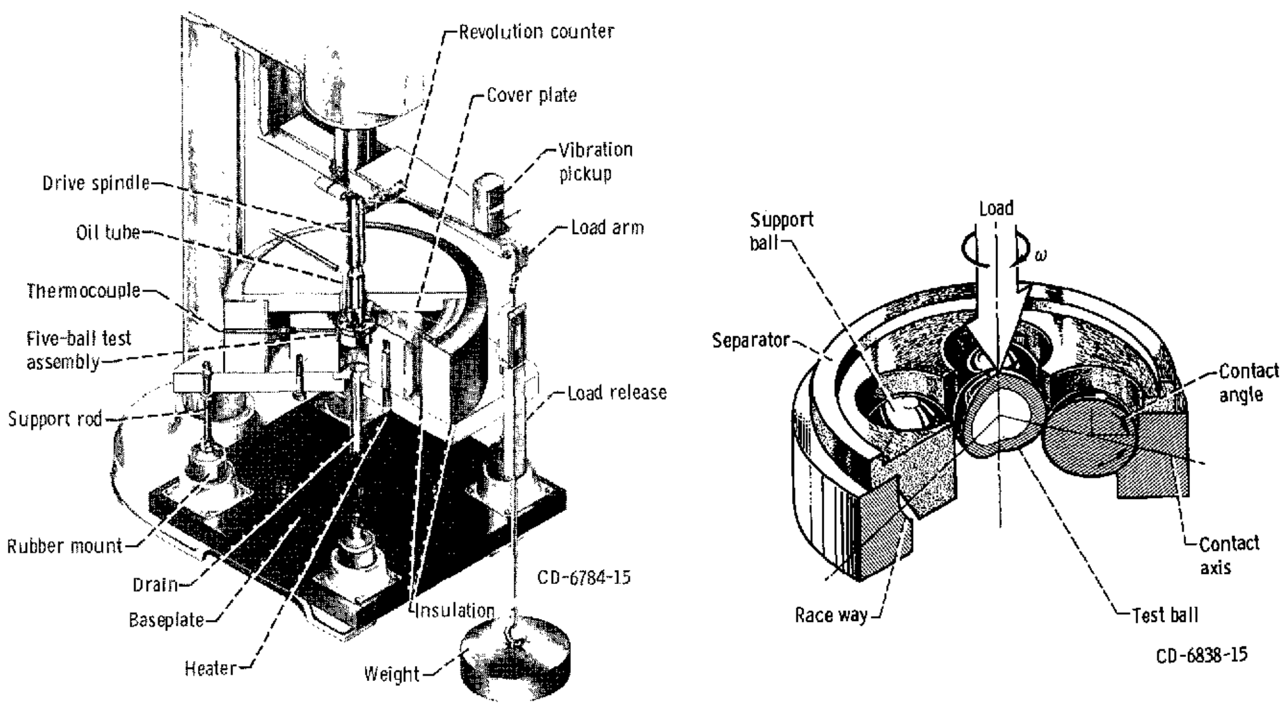
Ball diameter		Wall thickness		Outside to inside diameter ratio	Weight reduction, percent	Results	
mm	in.	mm	in.			Five-ball rig tests	Bearing tests
12.7	0.500	2.5	0.100	1.67	21.7	Normal-subsurface fatigue[a]	No failures[a]
17.5	.6875	1.5	.060	1.21	56.5	Flexure failures[a]	Flexure failures[a]
17.5	.6875	1.8	.070	1.26	50.0	Normal-subsurface fatigue	Flexure failures

[a]Results are from reference 6.

TABLE IV. - SUMMARY OF HOLLOW-BALL BEARING DAMAGE

Bearing	Total running time, hr	Number of spalled balls[a]
14-H	75.7	3
17-H	31.9	2
20-H	19.6	None

[a]Spalls initiated from cracks originating at the inside surface of the hollow ball and propagating to the outside surface.



(a) Cutaway view of five-ball fatigue tester.

(b) Five-ball test assembly.

Figure 1. - Test apparatus.

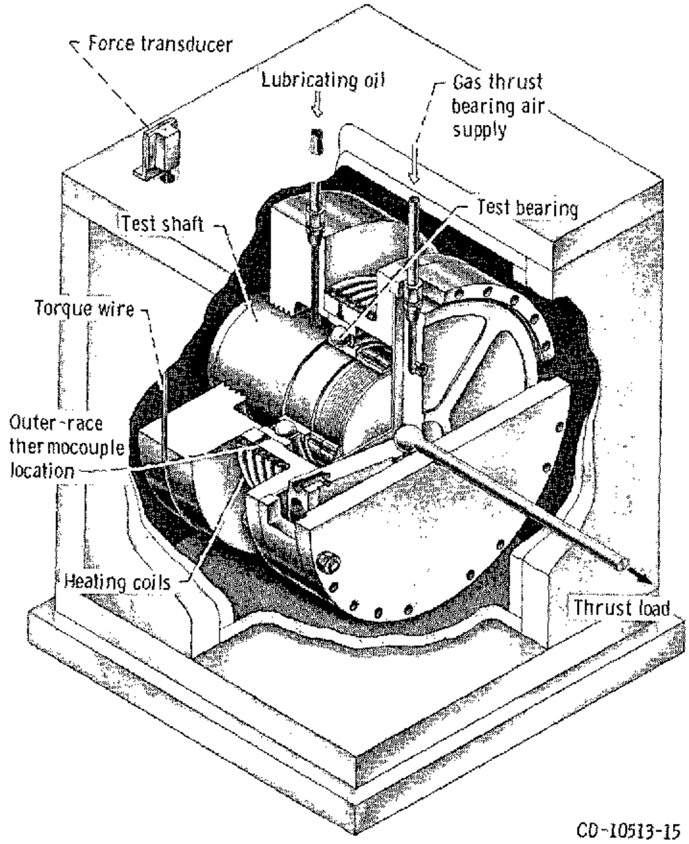


Figure 2. - Bearing test apparatus.

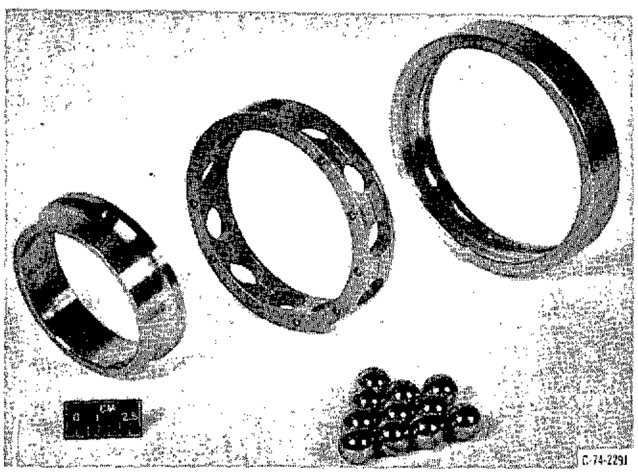


Figure 3. - Deep groove test bearing with inner race shoulder removed; two-piece machined cage construction.

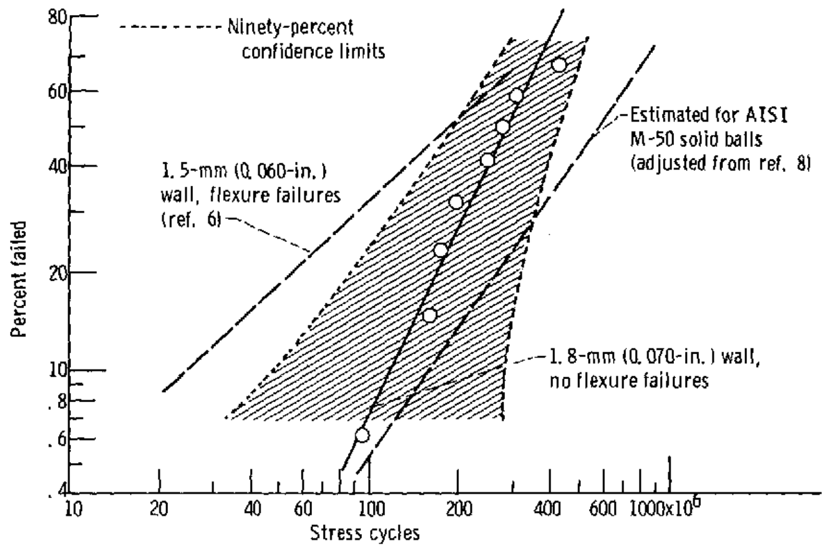


Figure 4. - Rolling-element fatigue life of AISI M-50 consumable-electrode vacuum-melted steel balls in five-ball fatigue tester. Maximum hertz stress, 4.8×10^9 pascal (700 000 psi); speed, 9700 rpm; no heat added; lubricant, super-refined naphtenic mineral oil.

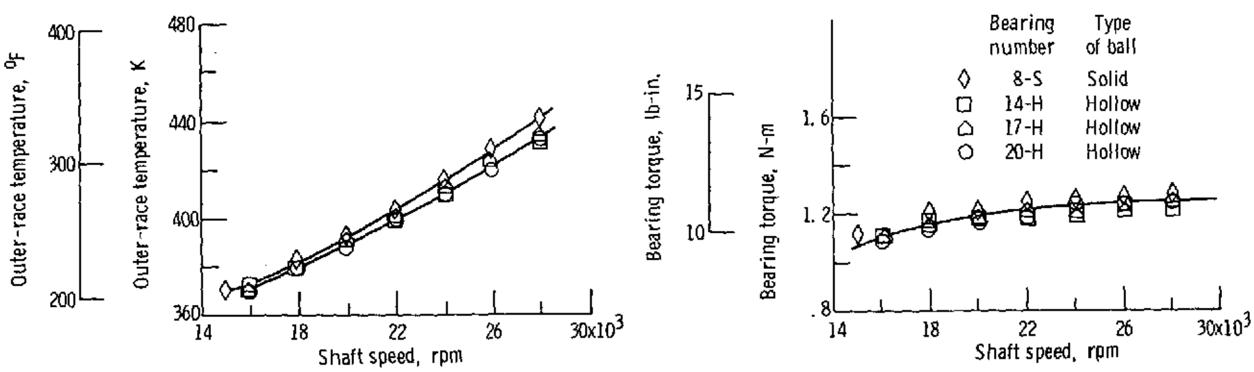
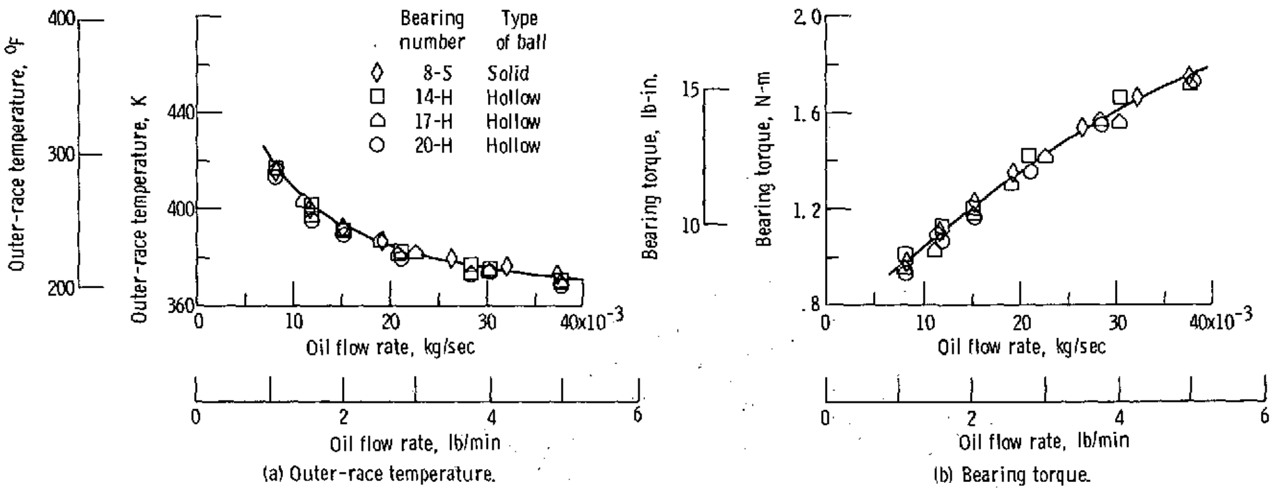


Figure 5. - Bearing performance as function of shaft speed. Thrust load, 2200 newtons (500 lb); oil flow rate, 15×10^{-3} kilograms per second (2 lb/min); oil inlet temperature, 316 K (110° F).



(a) Outer-race temperature.

(b) Bearing torque.

Figure 6. - Bearing performance as function of oil flow rate. Thrust load, 2200 newtons (500 lb); inlet oil temperature, 316 K (110° F); shaft speed, 20 000 rpm.

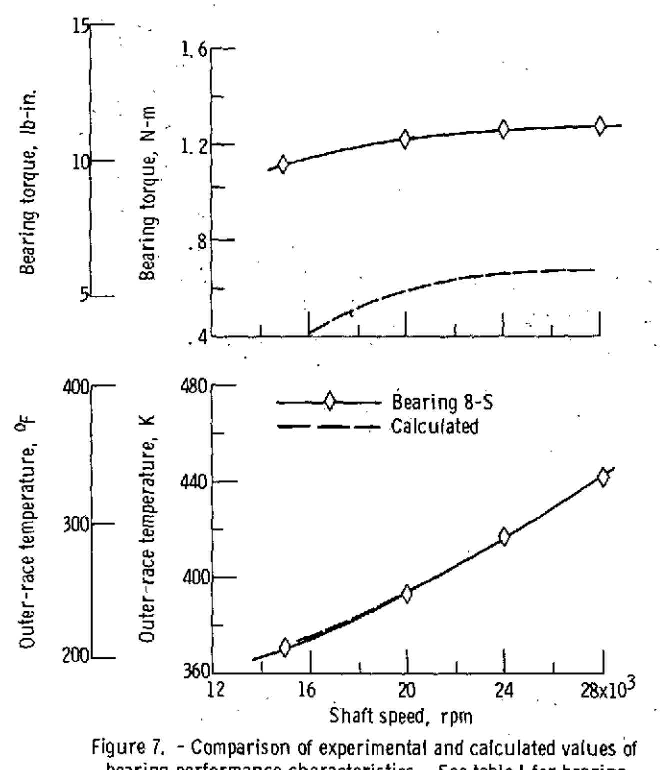


Figure 7. - Comparison of experimental and calculated values of bearing performance characteristics. See table I for bearing specifications. Load, 2200 newtons (500 lb); oil flow 15×10^{-3} kilogram per second (2 lb/min).

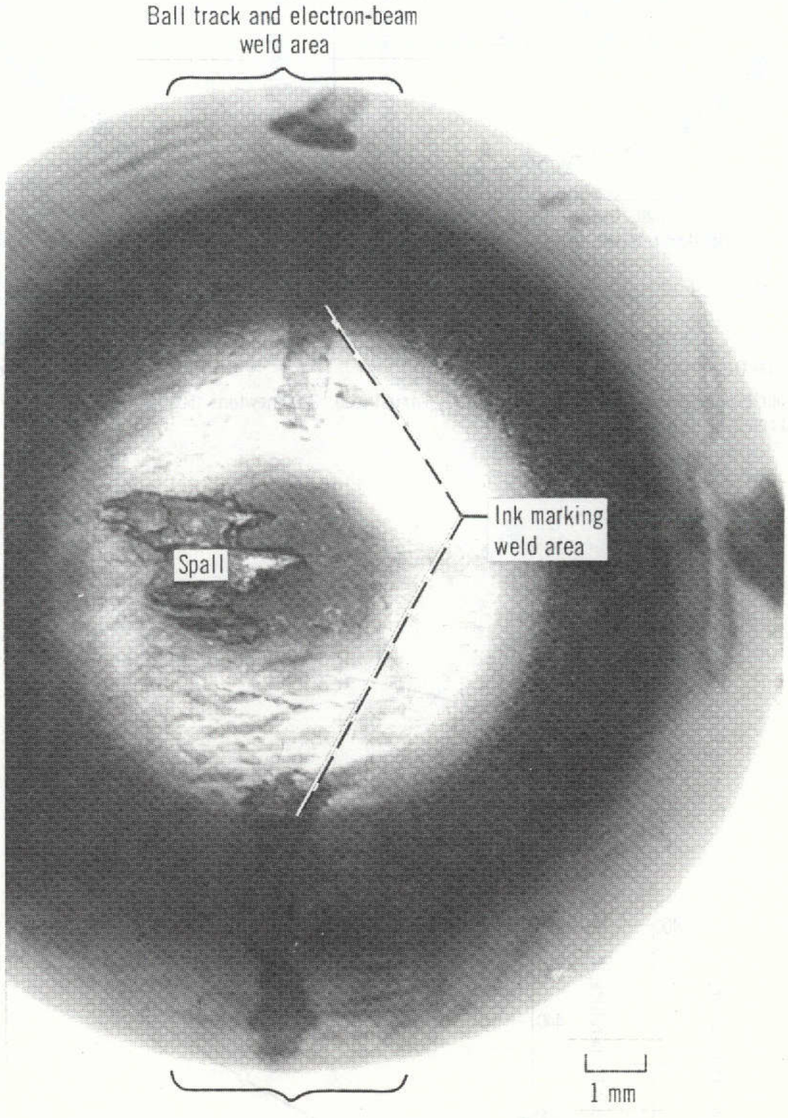
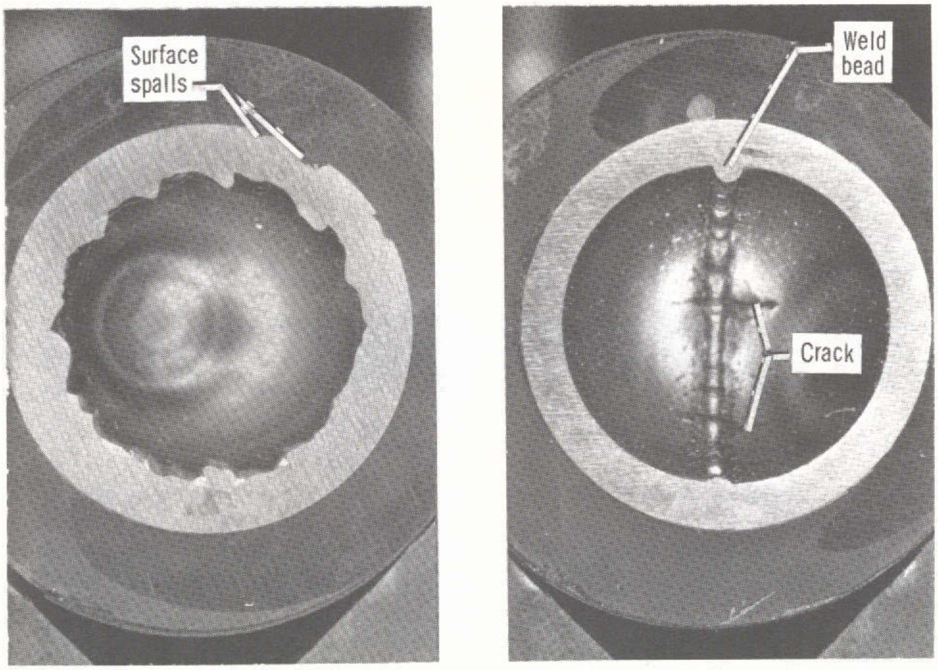


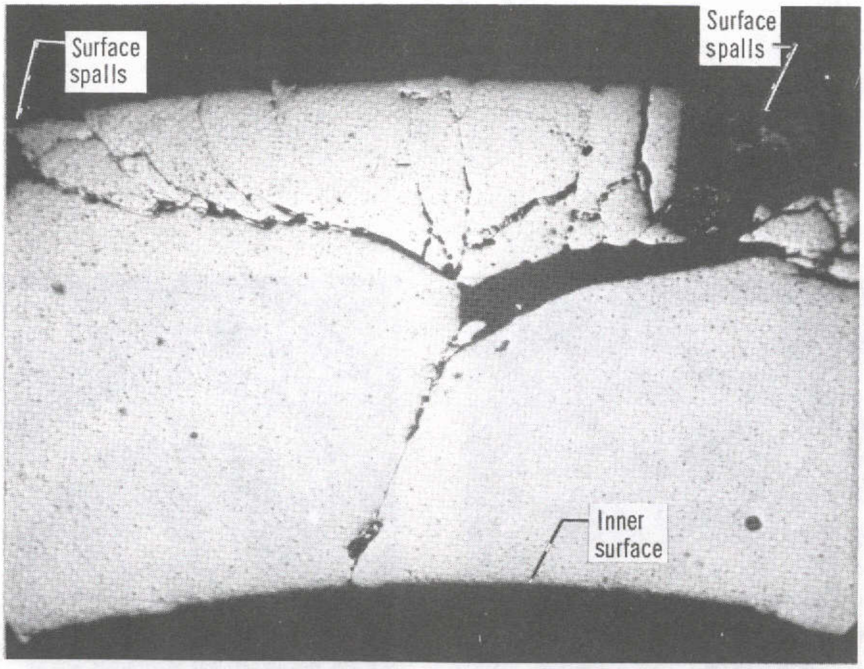
Figure 8. - View of external surface showing typical spall. Ball track directly on weld area.



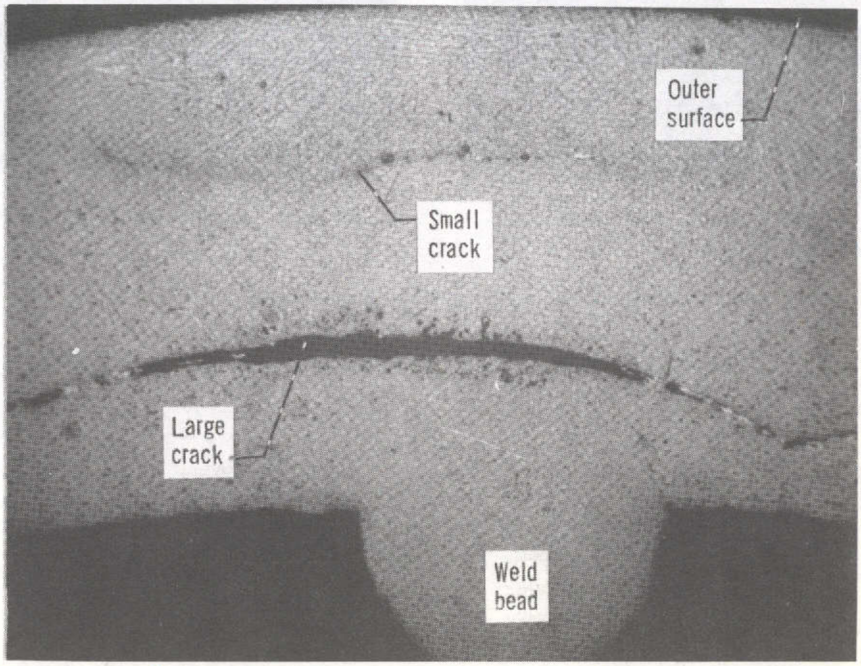
(a) Section parallel to and through ball track.

(b) Section perpendicular to ball track.

Figure 9. - Sectioned specimens of typical failed balls.



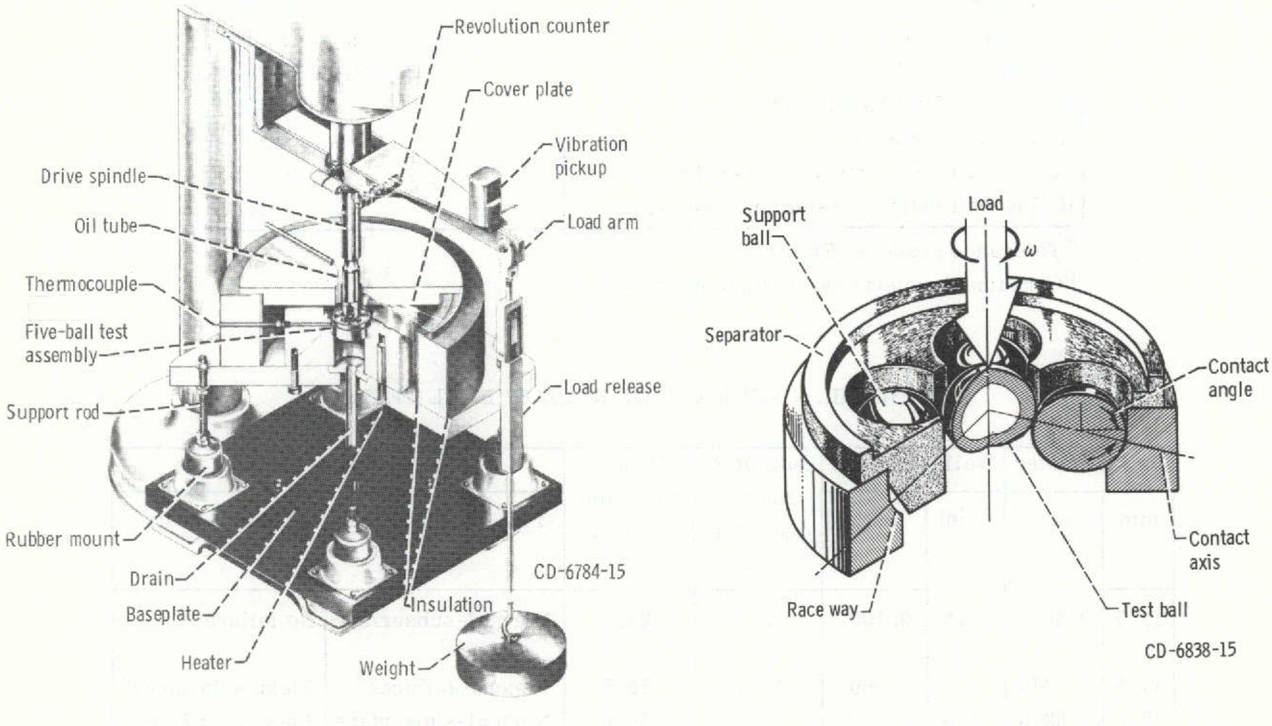
(a) Typical sectional view of specimen from figure 9(a).



(b) Typical sectional view of specimen from figure 9(b).

Figure 10. - Photomicrographs of polished specimens from figure 9.

NASA-Langley, 1975 E-7961



(a) Cutaway view of five-ball fatigue tester.

(b) Five-ball test assembly.

Figure 1. - Test apparatus.

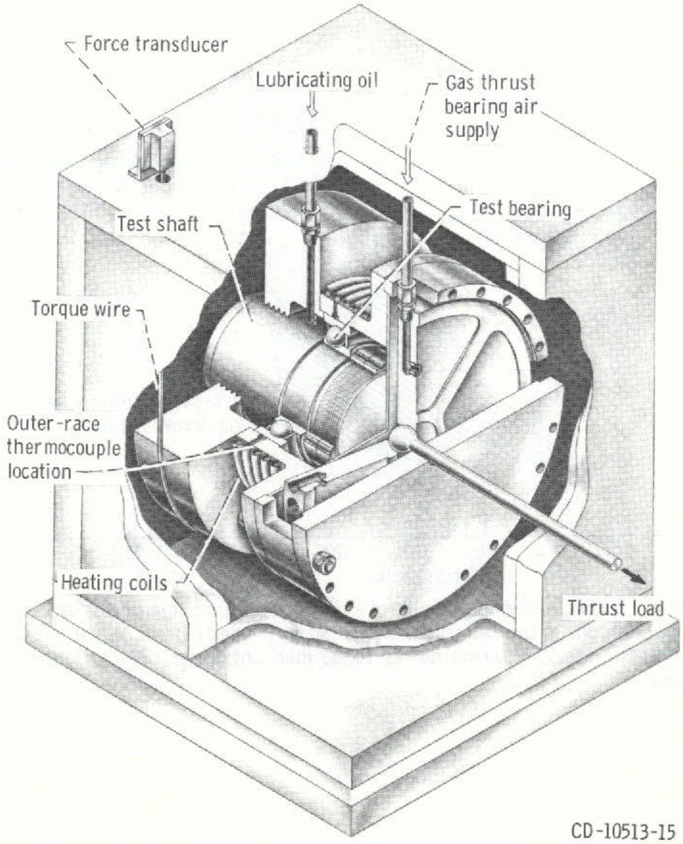


Figure 2. - Bearing test apparatus.

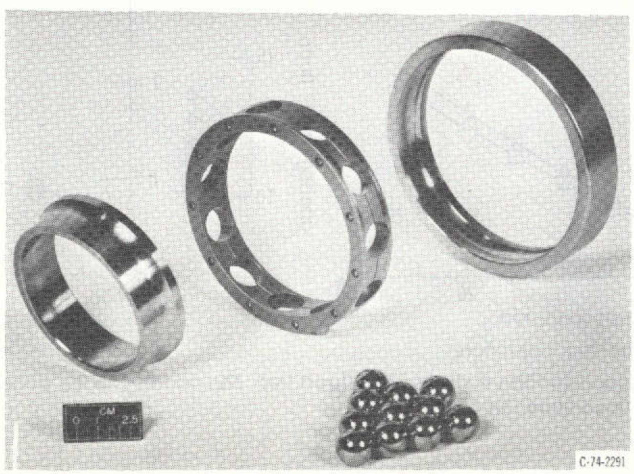


Figure 3. - Deep groove test bearing with inner race shoulder removed; two-piece machined cage construction.