ROLLING-ELEMENT FATIGUE LIFE
OF SAE 52100 STEEL HOLLOW BALLS

by Herbert W. Scibbe, Richard J. Parker, and Erwin V. Zaretsky

Lewis Research Center

Cleveland, Ohio

ROLLING-ELEMENT FATIGUE LIFE OF SAE 52100 STEEL HOLLOW BALLS

By Herbert W. Scibbe, Richard J. Parker, and Erwin V. Zaretsky

Lewis Research Center
Cleveland, Ohio

ROLLING-ELEMENT FATIGUE LIFE OF SAE 52100 STEEL HOLLOW BALLS

by Herbert W. Scibbe, Richard J. Parker, and Erwin V. Zaretsky

Lewis Research Center

SUMMARY

The NASA five-ball fatigue tester was used to determine the rolling-element fatigue lives of hollow and solid 1/2-inch-diameter (12.7 mm) balls. The upper test balls fabricated from consumable vacuum melt (CVM) SAE 52100 steel were run against SAE 52100 steel lower support balls. Tests were conducted at a maximum Hertz stress of 800 000 psi (5.52×10^9 N/m^2) with no heat added and with a super-refined naphthenic mineral oil as the lubricant.

The hollow balls were fabricated by a technique that included rough-forming hemispherical shells, joining them together by electron-beam welding, heat treating, and finishing to an Anti-Friction Bearing Manufacturers Association (AFBMA) 10 specification.

The solid balls were fabricated by conventional methods to an AFBMA 10 specification from the same heat of material. The solid balls were fatigue tested to obtain comparative data for the hollow ball fatigue results.

A 10-percent fatigue life of 60 million stress cycles was obtained for the group of hollow balls and compared favorably to the 10-percent life of 93 million stress cycles for the solid balls. A confidence level of 66 percent indicated that the differences in fatigue life between the two groups of balls may be insignificant.

The probability of a fatigue spall occurring in the weld area was calculated as 6.5 percent for those tests in which the running track crossed the weld. One of the ten fatigue spalls that occurred on the hollow balls was in the weld. This spall was a classical subsurface type similar in appearance to those occurring in the area outside of the weld and in the conventional solid balls.

Only minor differences in hardness were measured on a section of the hollow ball wall cut through a diameter normal to the weld. A similar metallographic structure was also observed, on this section, between the weld zone and the parent material. These results indicated extremely good control during the manufacturing process.

INTRODUCTION

Two problem areas of special interest in ball-bearing applications are (1) high-speed operation, where ball centrifugal force is significant, and (2) high-load, low-speed operation, where ball load sharing is required.

The balls, in a bearing operating at DN values (bearing bore in mm times shaft speed in rpm) above 1.5 million, orbit rapidly within the bearing. High centrifugal force is developed, resulting in an appreciable increase in Hertz (compressive) stress at the outer race-ball contact. In ball bearings operating at high speed with a nominal thrust load, ball centrifugal force is an important factor that can seriously limit the operating range of the bearing because of both high stresses and high heat generation (refs. 1 and 2). These limitations are especially significant in large-bore ball bearings containing large diameter balls. The effect on heat generation at high speed of various bearing geometric design factors, such as ball diameter, number of balls, pitch diameter, contact angle, and race curvature, can be determined by an analysis similar to that given in references 1 and 2.

One solution to problems of ball-bearing centrifugal force is to use smaller diameter balls, since reducing the physical size of the ball reduces the ball centrifugal force. However, ball size can be reduced only a limited amount before the annular space available for the cage becomes too small. If the cage cross section is reduced excessively, it may break at high speed because of inadequate strength.

Another possible solution for reducing ball centrifugal force is to use hollow balls. Ball weight and, consequently, centrifugal force are reduced without reducing ball size and thereby affecting the internal bearing design geometry or the cage design.

In low-speed, high-load applications, such as large radar scanning antennas, large-bore ball bearings are employed. Fatigue failures, which may have resulted from uneven loadings, have been experienced in the antenna support bearings. Investigators in reference 3 have proposed that under large normal loads a hollow ball with a sufficiently thin wall thickness will deflect appreciably more than a solid ball of the same size. An improvement in load distribution and thus load capacity may be realized by using a bearing with hollow balls. The feasibility of this concept, however, has not been verified experimentally.

The results of an experimental investigation comparing the static deflection characteristics of 15/16-inch-diameter (23.8 mm) hollow and solid balls subjected to normal loads from 0 to 3500 pounds (0 to 1.56×10^4 N) are shown in figure 1 and reported in reference 4. The deflections of the hollow balls with wall thicknesses of 0.188 and 0.078 inch (4.78 and 1.98 mm) were compared with the solid ball deflections. At loads of 1000 pounds (4450 N) and above, the hollow ball deflections were approximately 10 percent and more than 35 percent greater, respectively, than the solid ball deflections.
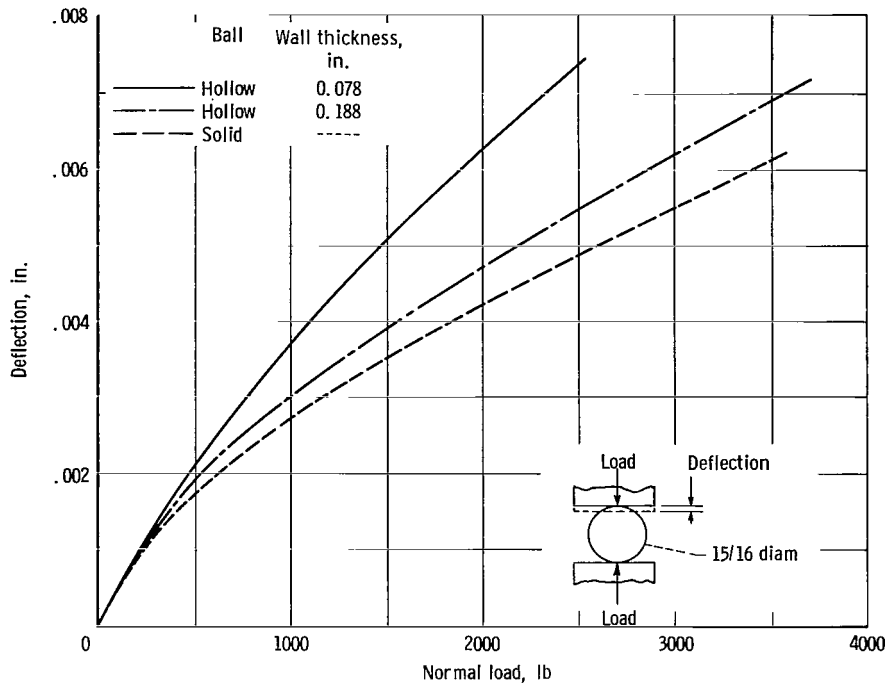
Figure 1. - Deflection characteristics of hollow and solid balls (ref. 4).

The objectives of this investigation were to determine experimentally (1) the fatigue life and the load capacity of SAE 52100 steel hollow balls relative to those of solid balls fabricated from the same heat of material, (2) whether the weld zones of the hollow balls are significantly weaker in fatigue, and (3) whether apparent differences in physical structure of the weld zone and parent metal, such as hardness and homogeneity, exist.

Fatigue tests were conducted with a group of 1/2-inch-diameter (12.7 mm) hollow balls with 0.100-inch (2.54 mm) wall thicknesses, and a group of 1/2-inch-diameter (12.7 mm) conventionally processed solid balls in the five-ball fatigue tester at a maximum Hertz stress of 800 000 psi (5.52×10^9 N/m^2), a drive-shaft speed of 10 600 rpm, a contact angle of 20°, and a nominal race temperature of 150° F (66° C). A super-refined naphthenic mineral oil was used as the lubricant. The fatigue lives of the hollow balls were compared to those of the solid balls. Both groups of balls had a nominal Rockwell C hardness of 65. All test results were obtained with a single batch of lubricant, the same heat of material for both hollow and solid upper test balls, and the same group of SAE 52100 steel lower support balls.

MATERIAL

The 1/2-inch-diameter (12.7 mm) ball specimens used for these tests were fabri-

cated from consumable electrode vacuum-melt (CVM) SAE 52100 steel. This steel was selected because of the large amount of bearing fatigue data obtained with this material and because of its widespread use in bearings. Both the hollow and solid balls were fabricated from the same heat of steel and finished to an Anti-Friction Bearing Manufacturers Association (AFBMA) grade 10 specification.

Fabrication and Heat-Treatment Specifications

The 1/2-inch-diameter (12.7 mm) hollow balls were cold-die forged into hemispherical shells with a nominal wall thickness of 0.100 inch (2.54 mm). The two hemispheres were matched and joined together by electron-beam welding. The electron-beam welding technique (ref. 5) has advantages over other types of fusion welding - namely, higher purity, higher depth to width ratio, less volume of molten metal to give off heat when the weld cools (thus minimizing distortion), and better heat concentration in a small area. The most important advantage is that the microstructure of the parent metal adjacent to the fusion zone is less affected by heat. After they were welded, the rough-formed spheres were stress relieved at 1000° F (538° C) and rough-ground prior to heat treatment.

The hollow balls were then heat treated according to the following schedule:

(1) Annealed to obtain a uniform structure in the weld, at 1475° F (802° C) for 7 hours in spent cast iron chips, furnace cooled to 300° F (149° C) at a rate not exceeding 20° F (11° C) per hour, and air cooled to room temperature

(2) Austenitized at 1550° F (843° C) for 12 minutes and oil quenched to room temperature to obtain a martensitic structure

(3) Tempered at 275° F (135° C) for 1 hour to obtain a nominal Rockwell C hardness of 65

The 1/2-inch-diameter (12.7 mm) solid balls were fabricated by conventional means and heat treated to a nominal Rockwell C hardness of 65 in accordance with the heat treatment schedule of the hollow balls with the exception of step (1), which applies to hollow balls only.

After the heat treatment, both the hollow and solid balls were finished to the required AFBMA 10 specifications.

Three rough-formed ball specimens were sectioned, polished, and etched in order to observe the depth of weld penetration and the fusion zone in the hollow ball wall. The specimens were sectioned on a plane normal to the weld band as shown in figure 2. The depth of weld penetration was approximately 75 percent of the wall thickness. This value was considered sufficient to provide adequate strength under high compressive stresses and deformations. A weld depth equal to the wall thickness will form a metal "flash"
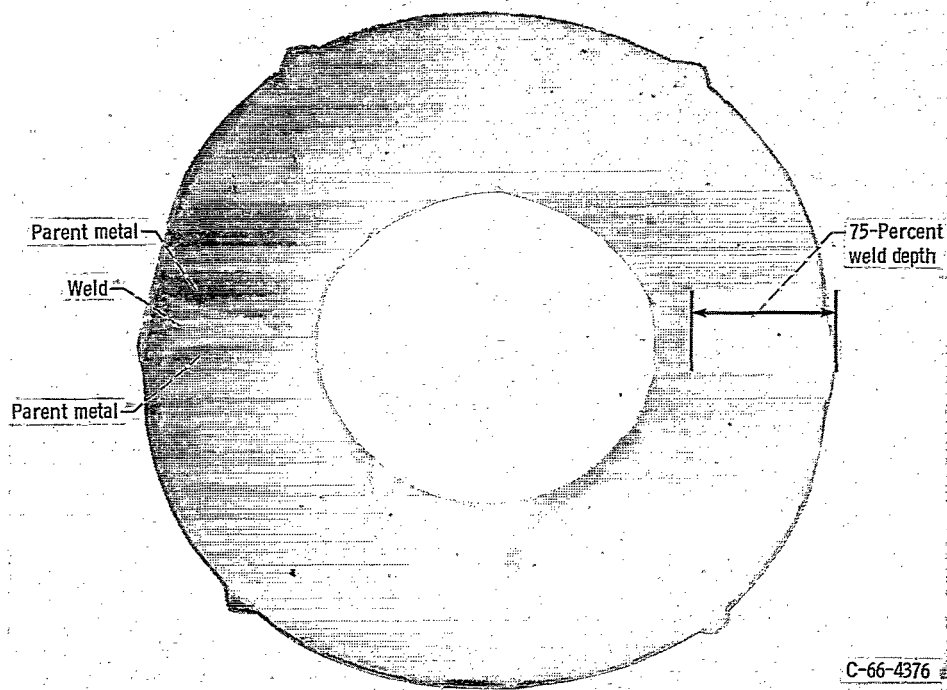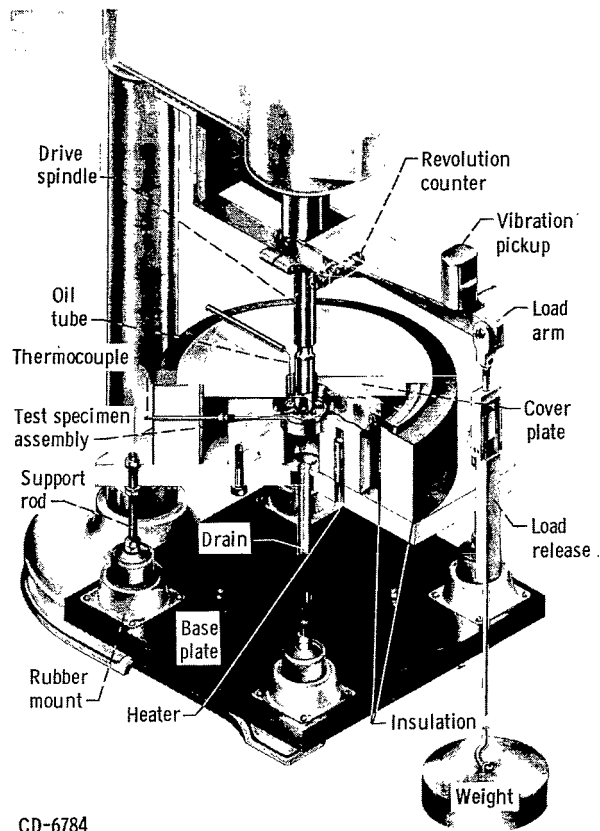
4

Figure 2. - Diametral section of rough-formed, welded, and annealed hollow ball illustrating depth of weld penetration and fusion zone in wall. X7.

on the inner wall which may result in a severe imbalance condition of the balls at high rotative speeds. The 75-percent weld depth will decrease somewhat in the finishing process as a result of grinding and lapping operations on the outer surfaces of the balls. The zone of complete fusion was observed to be the depth of the weld penetration through the walls (fig. 2).
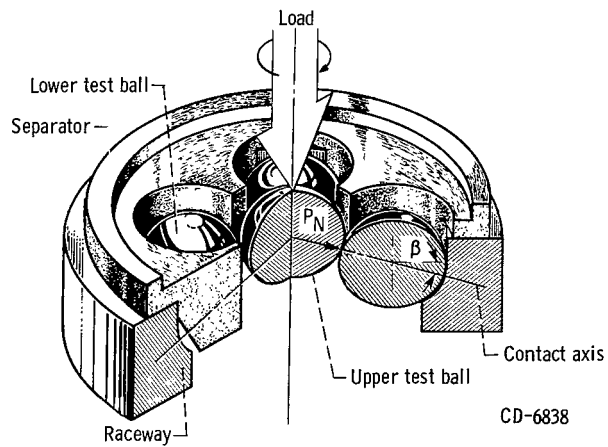
APPARATUS AND PROCEDURE

Five-Ball Fatigue Tester

The NASA five-ball fatigue tester was used for all tests conducted. The apparatus is shown schematically in figures 3(a) and (b) (p. 6) and is described in detail in reference 6. This fatigue tester consists essentially of an upper test ball pyramided upon four lower support balls that are positioned by a separator and are free to rotate in an angular contact raceway. System loading and drive are supplied through a vertical drive shaft. For every revolution of the drive shaft, the upper test ball received three stress cycles. The upper test ball and raceway are analogous in operation to the inner and

Drive
spindle

Revolution
counter

Vibration
pickup

Oil
tube

Load
arm

Thermocouple

Test specimen
assembly

Cover
plate

Support
rod

Drain

Load
release

Base
plate

Rubber
mount

Heater

Insulation

Weight

CD-6784

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

Load

Lower test ball

Separator

P_N

β

Contact axis

Upper test ball

Raceway

CD-6838

(b) Schematic of five-ball tester.

Figure 3. – Test apparatus.

6

outer races of a bearing, respectively. The separator and the lower support balls function in a manner similar to the cage and the balls in a bearing.

Fatigue Testing

Before they were assembled in the five-ball fatigue tester, all test-section components were flushed and scrubbed with ethyl alcohol and wiped dry with clean cheesecloth. The specimens were examined for imperfections at a magnification of 15 diameters. The lower support balls were fabricated from a different heat of material than that of the upper test balls. The support balls were grouped in sets of four with diameters matched within 20 microinches. This selective grouping ensured loading of the upper test specimen at all four contact points. After examination, all specimens were coated with test lubricant to prevent corrosion and wear at startup. A new set of lower support balls was used with each upper test ball specimen. If a lower support ball failed during the tests, the support ball set would be replaced and the test continued. The speed, outer-race temperature, and oil flow were monitored and recorded at regular intervals. After each test, or when the support ball set was replaced, the outer race of the five-ball system was examined visually for damage. If any damage was discovered, the race would be replaced prior to further testing.

Fatigue tests were conducted in the five-ball fatigue tester at a maximum Hertz stress of 800 000 psi (5.52×10^9 N/m^2), at a drive-shaft speed of 10 600 rpm, and at a contact angle of 20° (indicated by β in fig. 3(b)). The outer-race temperature stabilized at approximately 150° F (66° C) with no heat added. The stress that was developed in the contact area was calculated by using the Hertz formulas given in reference 7.

Method of Presenting Fatigue Results

The total test time for each specimen was recorded and converted to total stress cycles. The statistical methods of reference 8 for analyzing rolling-element fatigue data were used to obtain a log-log plot of the reciprocal of the probability of survival as a function of the log of stress cycles to failure (Weibull coordinates). For convenience, the ordinate is graduated in statistical percent of specimens failed. From these plots, the number of stress cycles necessary to fail any given portion of the specimen group may be determined. Where high reliability is of paramount importance, the main interest is in early failures. For purposes of comparison, the 10-percent life on the Weibull plot was used. The 10-percent life is the number of stress cycles within which 10 per-

cent of the specimens can be expected to fail; this 10-percent life is equivalent to a 90-percent probability of survival. The failure index indicates the number of specimens that failed out of those tested.

Posttest Inspection

After the tests, all the hollow balls and those solid balls with failures were etched with a 50-percent hydrochloric acid solution (at 140° F, 60° C) for approximately 3 minutes to determine the locations of the polar areas. The polar areas are two areas on a ball, 180° apart, where the fiber or grain flow is perpendicular to the surface (ref. 9). An additional etching time of 2 to 3 minutes on the hollow balls was required to locate the weld bands. After the etching, the weld bands and running tracks on the hollow balls were marked with a pencil. The angle of intersection of the weld band and the running track was also measured.

A hollow and solid ball were each sectioned on a diametral plane through the polar areas and normal to the equator (weld band on the hollow ball). The sections were pol-
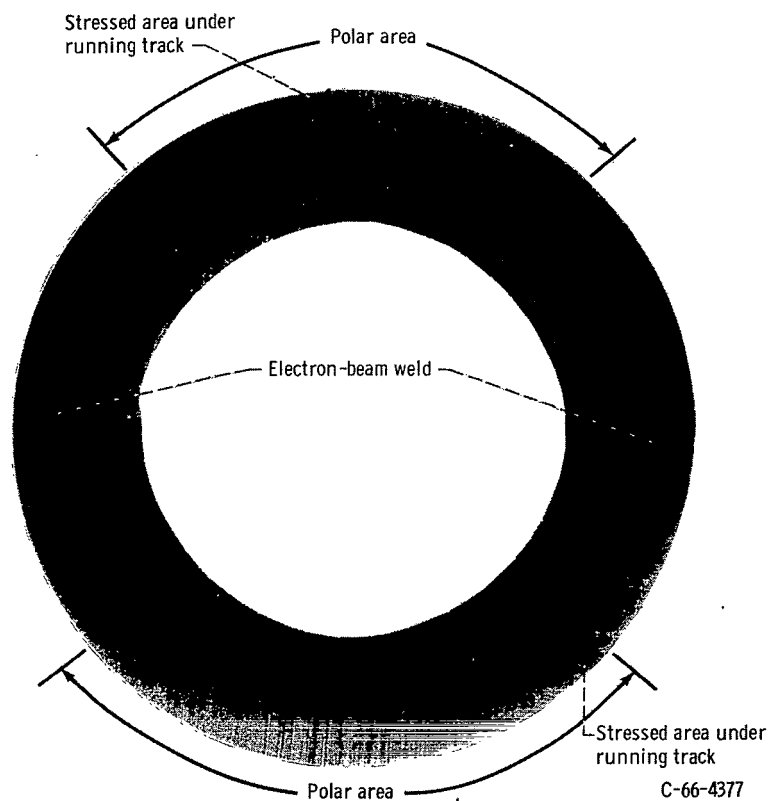

Figure 4. - Etched cross section of finished hollow ball showing fiber flow lines, polar areas, and weld zone. X7.

ished and etched with a 1.5-percent nital solution. Photomacrographs were made in order to identify the fiber flow lines and to measure the polar areas. A cross section of a hollow ball showing the fiber flow lines and polar areas is presented in figure 4. The width of the weld band measured approximately 0.04 inch (1.0 mm). Similarly, the fiber flow lines and the polar areas for a solid ball cross section are shown in figure 5. The polar areas for the solid and hollow balls measured approximately the same.

Posttest hardness measurements were made on the test ball specimens and on the lower support balls. The hollow test balls, the solid test balls, and the lower support balls had average Rockwell C hardnesses of 64.8, 64.7, and 62.6, respectively. These hardness readings indicated a nominal differential hardness value ΔH (hardness of the lower support balls minus the hardness of the test ball, refs. 10 and 11) of -2 for both the hollow and solid test balls.

The 1/2-inch-diameter (12.7 mm) hollow test balls with 0.100-inch (2.54 mm) wall thicknesses weighed 0.0143 pound (6.42 grams) each, whereas the 1/2-inch-diameter (12.7 mm) solid balls weighed 0.0183 pound (8.29 grams) each. The hollow ball weight was thus 77 percent of the solid ball weight.
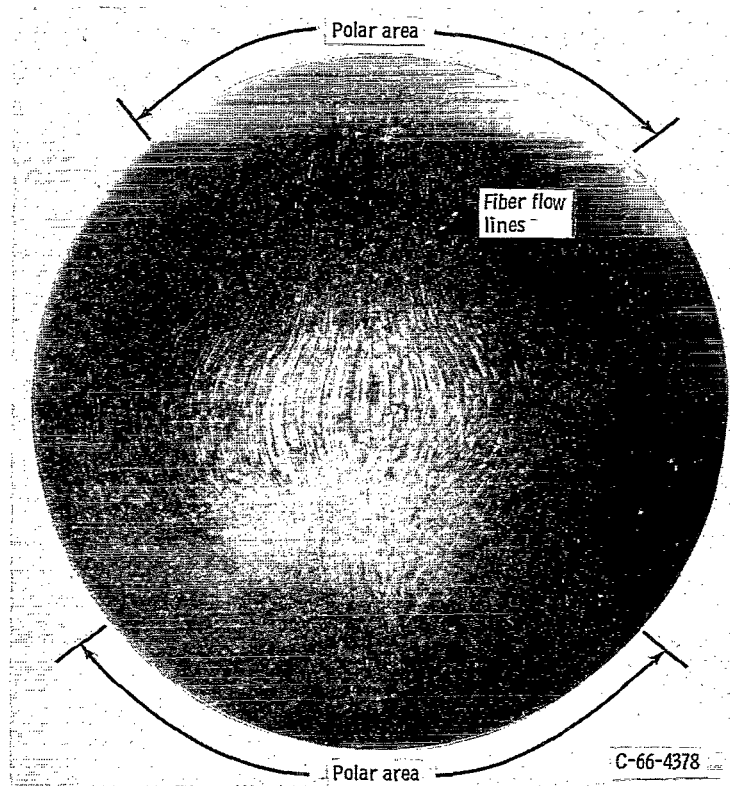

Figure 5. – Etched cross section of solid ball showing fiber flow lines and polar areas. X7.

RESULTS AND DISCUSSION

Life Results and Load Capacity

Groups of 1/2-inch-diameter (12.7 mm) hollow and conventional solid balls fabricated from the same heat of SAE 52100 CVM steel were tested as upper test ball specimens against SAE 52100 lower support balls in the NASA five-ball fatigue tester. Standard test conditions were ambient temperature (i.e., no heat added) and a maximum Hertz stress of 800 000 psi (5.52×10^9 N/m^2) at a 20° contact angle. A super-refined naphthenic mineral oil was used as the lubricant. The nominal wall thickness of the hollow balls was 0.100 inch (2.54 mm). The results of the fatigue tests for the upper test specimens are presented in figure 6 and are summarized in table I (p. 11). The 10-percent life for the conventional solid test balls was 93 million stress cycles as opposed to 60 million stress cycles for the hollow ball test specimens. In order to determine the significance of the apparent higher life obtained with the solid test balls, the data were evaluated statistically by using the methods of reference 8. It was determined that a 66-percent or approximately a one-sigma confidence existed at this life difference (a one-sigma confidence is approximately 68 percent). This means that approximately 66 out of 100 times the fatigue lives of a group of solid conventional balls will exceed those of an equivalent group of hollow balls operated under identical test conditions. In general, however, a confidence number of 66 is not considered significant in order to state that the welded hollow balls are inferior in terms of fatigue life to conventional balls.

An important criterion of rolling-element operation is bearing or specimen load-carrying capacity, usually termed capacity. This is the load that will theoretically give a ball specimen life of one million stress cycles or a bearing life of one million inner-race revolutions with a 90-percent probability of survival. The load-carrying capacity may be calculated from the fatigue life results that are summarized in table I. The equation for the load-carrying capacity is

$$C = P_N \sqrt[3]{L}$$

where C is the load capcity, P_N is the normal load between the upper test ball and the lower support ball, and L is the 10-percent life of the upper test ball in millions of stress cycles. The load capacity obtained for each group of specimens tested is given in table I. The capacities for the hollow and solid balls are 669 and 775 pounds, respectively.
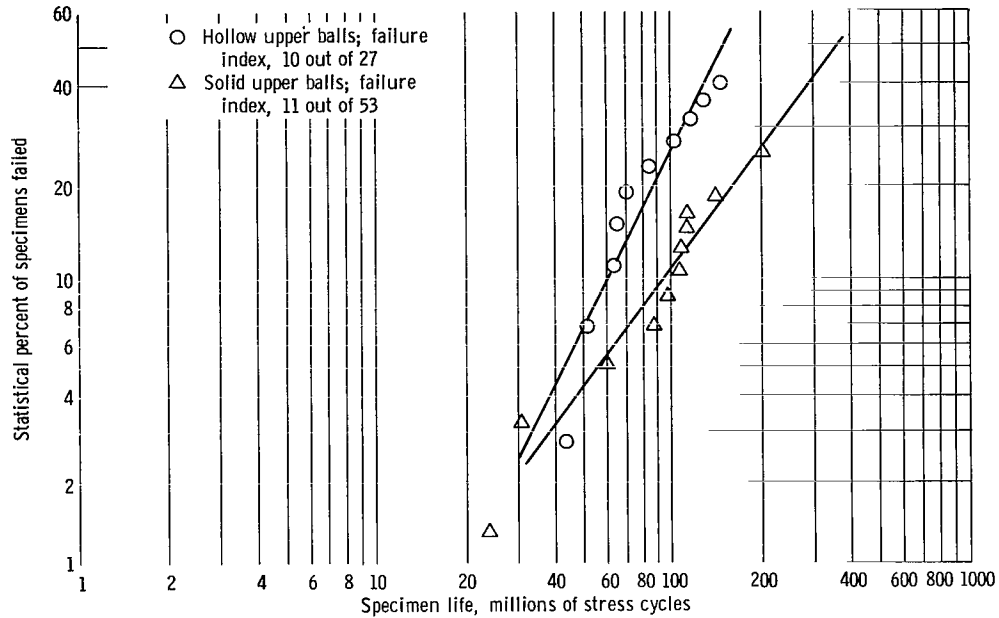
Figure 6. – Rolling-element fatigue life of SAE 52100 CVM steel balls of nominal Rockwell C hardness 65 run with lower support balls of SAE 52100 steel of nominal Rockwell C hardness 63. Initial maximum Hertz stress, 800 000 psi (5.52x10^9 N/m^2); speed, 10 600 rpm; outer-race temperature, 150° F (66° C); lubricant, super-refined naphthenic mineral oil.

TABLE I. – FATIGUE LIFE AND LOAD CAPACITY FOR HOLLOW AND SOLID BALLS

[Ball diameter, 1/2 inch (12.7 mm); material, SAE 52100 steel; initial maximum Hertz stress, 800 000 psi (5.52×10^9 N/m^2); contact angle, 20°; nominal race temperature, 150° F (66° C); lubricant, super-refined naphthenic mineral oil; nominal ΔH, -2 points Rockwell C.]

Test specimen	Normal ball load, lb (N)	10-Percent fatigue life, millions of stress cycles	50-Percent fatigue life, millions of stress cycles	Normal load capacity, lb (N)	Confidence number, percent	Failure index	Weibull slope
Hollow balls	169 (752)	60	149	669 (2980)	--	10 out of 27	2.05
Solid balls	169 (752)	93	369	775 (3450)	66	11 out of 53	1.38

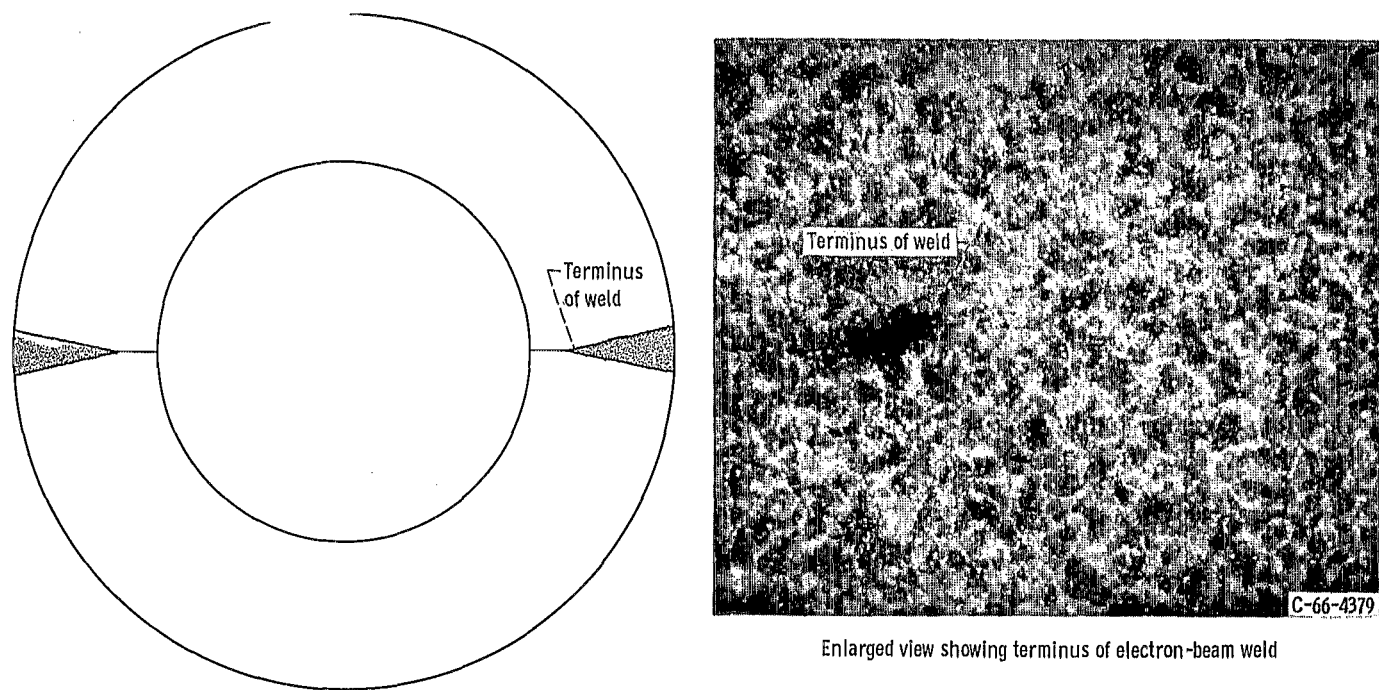Enlarged view showing terminus of electron-beam weld

Figure 7. – Cross section of hollow ball illustrating grain structure and homogeniety at terminus of electron-beam weld.   Etchant, 1.5-percent nital.   X250.

Effect of Weld Band on Hollow Ball Fatigue Life

Examination of the 27 hollow balls tested revealed that 10 had failed in fatigue. Of the 10 fatigue failures, 4 failures had occurred in the polar areas (fig. 4, p. 8). Five failures occurred between the weld and the polar areas and 1 failure occurred in the weld. For the purposes of comparison, in the solid ball group, 5 failures out of 11 occurred in the polar areas (fig. 5, p. 9).

The probability of a failure occurring in the weld, if the weld area is assumed to be of equal fatigue strength to the area outside the weld, is the ratio of the length of the running track in the weld to the total track length on the ball. The track length in the weld for each hollow ball, in which the running track crossed the weld, was measured. The probability of a weld failure was determined to be approximately 6.5 percent for the group of hollow balls. One failure, out of a total of ten, occurring in the weld would therefore not be unexpected.

From a statistical viewpoint, it is conceivable that the effect the weld area has on decreasing the fatigue life of the hollow ball can be minimized. Minimizing can be done by either increasing the size of the ball while maintaining the same weld area, or by decreasing the weld area for the same ball size.

The fatigue spall that occurred on the weld of a hollow ball was a classical subsurface type. It was similar in appearance to those occurring in the area outside of the weld and in the conventional solid balls.

A photomicrograph of the cross section of a hollow ball at the terminus of the electron-beam weld is presented in figure 7. The weld microstructure indicates a microstructure very similar to that of the parent material. Hardness measurements were made across the weld zone and the adjacent parent material on the sectioned ball. A variation in Rockwell C hardness of less than 1 point was measured. These hardness measurements indicated extremely good control during the manufacturing process.

SUMMARY OF RESULTS

The NASA five-ball fatigue tester was used to determine the rolling-element fatigue life of hollow and solid 1/2-inch-diameter (12.7 mm) SAE 52100 CVM steel upper test balls run against SAE 52100 steel lower test balls. Tests were conducted at a maximum Hertz stress of 800 000 psi (5.52×10^9 N/m^2), with no heat added. A super-refined naphthenic mineral oil was used as the lubricant. The fatigue results with hollow balls fabricated by an electron-beam welding technique were compared with fatigue results obtained for solid balls fabricated by conventional methods. Both the hollow and solid ball specimens were from the same heat of material. The following results were obtained:

1. The fatigue life obtained for the hollow balls was not significantly less than that obtained for the solid conventional balls.

2. Of the 10 fatigue spalls occurring on the 27 hollow ball specimens tested, only one spall occurred on a weld.

3. The fatigue spall on the weld of a hollow ball was a classical subsurface type similar in appearance to those occurring in the area outside of the weld and in the conventional solid balls.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, November 25, 1966,
720-03-01-01-22.

REFERENCES

1. Bisson, Edmond E.; and Anderson, William J.: Advanced Bearing Technology. NASA SP-38, 1964, pp. 311-321.

2. Scibbe, Herbert W.; and Anderson, William J.: Evaluation of Ball-Bearing Performance in Liquid Hydrogen at DN Values to 1.6 Million. Trans. ASLE, vol. 5, no. 1, Apr. 1962, pp. 220-232.

3. Perkins, Dave: Extensive Defense Potential Seen in Hollow Bearing Balls. Metalworking News, vol. 6, no. 265, Sept. 6, 1965, pp. 1, 5.

4. Hanau, Heinz, et al.: New Concepts in Bearing Designs and Applications. Industrial Tectonics, Inc., Compton, Calif., 1965, pp. 20-23.

5. Bunshah, R. F.: High-Power Electron Beams. Int. Science Tech. no. 4, Apr. 1962, pp. 30-38.

6. Carter, Thomas L.; Zaretsky, Erwin V.; and Anderson, William J.: Effect of Hardness and Other Mechanical Properties on Rolling-Contact Fatigue Life of Four High-Temperature Bearing Steels. NASA TN D-270, 1960.

7. Jones, A. B.: New Departure-Analysis of Stresses and Deflections. Vols. I and II. New Departure Div., General Motors Corp., Bristol (Conn), 1946.

8. Johnson, Leonard G.: The Statistical Treatment of Fatigue Experiments. Rep. No. GMR-202, General Motors Corp., Res. Labs., Apr. 1959.

9. Carter, Thomas L.: A Study of Some Factors Affecting Rolling-Contact Fatigue Life. NASA TR R-60, 1960.

10. Zaretsky, Erwin V.; Parker, Richard J.; and Anderson, William J.: Effect of Component Differential Hardnesses on Rolling-Contact Fatigue and Load Capacity. NASA TN D-2640, 1965.

11. Zaretsky, E. V.; Parker, R. J.; and Anderson, W. J.: Component Hardness Differences and Their Effect on Bearing Fatigue. Paper No. 65-Lub-7, ASME, 1965.

"The aeronautical and space activities of the United States shall be conducted so as to contribute . . . to the expansion of human knowledge of phenomena in the atmosphere and space. The Administration shall provide for the widest practicable and appropriate dissemination of information concerning its activities and the results thereof."

—NATIONAL AERONAUTICS AND SPACE ACT OF 1958

NASA SCIENTIFIC AND TECHNICAL PUBLICATIONS

TECHNICAL REPORTS: Scientific and technical information considered important, complete, and a lasting contribution to existing knowledge.

TECHNICAL NOTES: Information less broad in scope but nevertheless of importance as a contribution to existing knowledge.

TECHNICAL MEMORANDUMS: Information receiving limited distribution because of preliminary data, security classification, or other reasons.

CONTRACTOR REPORTS: Technical information generated in connection with a NASA contract or grant and released under NASA auspices.

TECHNICAL TRANSLATIONS: Information published in a foreign language considered to merit NASA distribution in English.

TECHNICAL REPRINTS: Information derived from NASA activities and initially published in the form of journal articles.

SPECIAL PUBLICATIONS: Information derived from or of value to NASA activities but not necessarily reporting the results of individual NASA-programmed scientific efforts. Publications include conference proceedings, monographs, data compilations, handbooks, sourcebooks, and special bibliographies.

Details on the availability of these publications may be obtained from:

SCIENTIFIC AND TECHNICAL INFORMATION DIVISION
NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
Washington, D.C. 20546