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BENDING STRESSES IN SPHERICALLY HOLLOW BALL
BEARING AND FATIGUE EXPERIMENTS

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ABSTRACT

Spherically hollow balls of 21.7, 50.0, and 56.5 percent mass reduction have been operated in ball bearings and in a five-ball fatigue tester with differing outcomes. Available theoretical and experimental treatments of stresses in spherically hollow balls are reviewed and compared. Bending stresses are estimated for these spherically hollow balls to better understand the differences in ball bearing and fatigue test experience.

INTRODUCTION

Commercial aircraft gas turbines currently operate in a speed range of about 1.5 to 2 million DN (bearing bore in mm times shaft speed in rpm). Trends in turbine design have resulted in requirements for higher shaft speeds and larger shaft diameters [1][1] and it is estimated that future engines may require bearings to operate at DN values of three million or higher. However, the high centrifugal forces developed at the outer race can seriously reduce the bearing fatigue life. One of the solutions for this problem would be to reduce the mass of the balls and thus reduce the centrifugal force. A method for reducing the ball mass is to make it hollow. A hollow bearing ball can be fabricated by welding two hemispherically formed shells [2]. Three different sized electron-beam welded hollow balls were evaluated for potential use as bearing balls in Refs. [2] and [3]. The 12.7 mm (0.500 in.) diameter balls (diameter ratio 1.67) operated satisfactorily in a 75 mm bore bearing with no failures [2]. The 17.5 mm (0.6875 in.) diameter balls (diameter ratio 1.26) of Ref. [3] experienced flexure failures in bearing tests but only classic fatigue spalls in the fatigue tests. The 17.5 mm balls (diameter ratio 1.21) of Ref. [2] experienced flexure fatigue in both the bearing and the fatigue tests. The stresses at the inner surfaces of these balls however, have not been calculated, nor has the theory permitting such a calculation been experimentally verified.

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[1]Numbers in brackets designate references at end of paper.

To better evaluate the results of the hollow ball tests, the research reported herein was undertaken. The objective of this investigation was: (1) to employ strain gage techniques to measure strains on the inner surface of spherically hollow ball models under static loading, (2) to compare the experimental results with theoretical predictions, and (3) to apply the results to the actual bearing balls previously tested.

EXPERIMENTAL DETERMINATION OF STRESSES IN BALLS

Actual sized bearing balls would be very difficult to instrument for experimental stress analysis. Therefore large models of spherically hollow balls proportioned for mass reductions of 40, 50, and 60 percent were fabricated by joining hemispheres made from mild steel bar stock. The hemispheres were turned with a radius cutting tool to a 63.5 mm (2.5 in.) radius spherical exterior contour, and then turned to a spherical interior radius calculated to provide the desired mass reduction. Strain gage rosettes incorporating 1 mm (0.04 in.) gage length gages were mounted inside one of the hemispheres with lead wires brought out through a 7 mm (0.27 in.) diameter hole in the center of the hemisphere opposite the gages.

The mild steel material simplified metal cutting. Care was taken to ensure strains were within the elastic range so lack of hardness created no difficulty. The hemispheres were bonded with epoxy adhesive. Analysis of the strain gage data indicated that the bond line transmitted forces and moments very well. Five rosettes were installed over a 120° arc of a great circle of the model. Loads were applied at four different locations over the great circle arc of gages by repositioning the model in a universal testing machine. A sketch of a typical ball model showing rosette and load locations is shown in Fig. 1. A photograph of an instrumented ball positioned for loading is shown in Fig. 2. A more complete description of the models and procedures is given in Ref. [4].

Principal strains and stresses were calculated for each rosette location. The principal stresses were then used to calculate dimensionless stress coefficients applicable to spherically hollow balls of any size proportioned for the same mass reduction percentage from

$$K_{1,2} = \frac{\sigma_{1,2}}{\frac{P}{\pi R_0^2}} \tag{1}$$

where $K_{1,2}$ are the principal stress coefficients, $\sigma_{1,2}$ are the maximum and minimum stresses at the rosette locations, P is the applied contact load, and R_0 is the outer radius of the ball. Dimensionless stress coefficients are plotted against angle from the applied load for the 40, 50, and 60 percent mass reduction cases in Figs. 3, 4, and 5.

Theoretical stress calculations for hollow balls by Golecki in Ref. [6] reported a theory of elasticity Legendre polynomial series solution for the state of stress in hollow spheres, and used five terms of the series to obtain values of dimensionless stress coefficients for the interior surface of a sphere of radius ratio (inside radius over outside radius) 0.25, corresponding to a mass reduction of 1.56 percent.

Pih and Vanderveldt [7] in reporting the result of a three-dimensional photoelasticity investigation found good agreement between Golecki's analytical solution and their experimental result within the region of the cavity for spheres of radius ratio 0.25 and 0.33. However, they also noted that the Legendre polynomial series was oscillatory for radial distances greater than 0.5 times the outer radius and that it diverges near the outer boundary. Golecki pointed out in a discussion to Pih and Vanderveldt's paper that his result was intended to apply to the interior surface, and that with special attention to slowly convergent or divergent parts of the series, it could be used for points between the cavity and outer surface of hollow spheres.

Using Golecki's analytical solution, stresses at the interior surface of spheres proportioned for mass reductions of 40, 50, and 60 percent were calculated though it was necessary to sum 40 terms of the series for the 60 percent mass reduction case. The radius ratios were 0.736, 0.794, and 0.544, respectively. Dimensionless stress coefficients were calculated from Golecki's theoretical results and are superimposed on the experimental stress coefficients plotted in Figs. 3, 4, and 5. It can be seen that the agreement between the analytical and experimental results is very good. Maximum dimensionless stress coefficients are 43.1, 75.9, and 142.8 in tension and 13.6, 20.4, and 33.6 in compression for the 40, 50, and 60 percent mass reduction cases.

Rumbarger, Herrick, and Eklund [8] reported the result of a finite element computer analysis of a hollow ball contacting a flat plate. The ball model had a diameter of 25.4 mm (1 in.), a wall thickness of 2.03 mm (0.08 in.) and was loaded to 4450 N (1000 lb). They reported calculated interior bending stresses of 1305 MN/m^2 (150 000 psi) directly under the load and 34.5 MN/m^2 (5000 psi) at 90° to the load. The radius ratio for this case was 0.84.

From this it may be inferred that the ball was proportioned to a 59.3 percent mass reduction and that dimensionless stress coefficients would be 117.8 and 3.9 at 0° and 90°. These compare with values of 138.1 and 3.09 interpolated between Golecki's series solutions for a 59.3 percent mass reduction ball. The values are of an appropriate magnitude, but do not agree as well as do the experimental results. The reason for the lack of agreement between Rumbarger et al., and Golecki is not clear.

Ling, Chow, and Rivera [9] have presented an analytical solution for the elastodynamic response of hollow balls. However, this analysis was not applied to the statically loaded 127 mm (5 in) diameter ball models.

Chiu [10] has considered the case of an elastic hollow sphere in-
denting a cup, which permits calculation of elastic deflections, contact
radius, and pressure.

FLEXURAL STRESSES IN FATIGUE TESTS AND BEARING EXPERIMENTS

Hollow balls having weight reductions of 21.7, 50.0, and 56.5-
percent have been tested in fatigue tests and bearing experiments in
Refs. [2] and [3]. The flexural stress levels on the inner surface of
these hollow balls may be estimated from Eq. (1). The contact forces
for the balls were determined by calculation using the highest load and
speed condition indicated in the reference. The maximum dimensionless
stress coefficients were calculated using Golecki's Legendre polynomial
series solution. Using the ball contact forces and outer radii with
these coefficients, the bending stresses were obtained. Values of the
loads, stresses, sizes, operating conditions, and test results of the
balls are summarized in Table 1.

First, from Table 1 note that: (1) the 21.7-percent reduction ball
experienced no flexure failures, (2) the 56.5-percent reduction ball had
flexural fatigue failures in both the fatigue tests and the bearing
tests, and (3) the 50-percent reduction ball had no flexure failures in
the fatigue tests, but did have flexural fatigue failures during the
bearing tests. Therefore, since some balls failed by flexure and some
did not, further analysis of the data in Table 1 should be informative.

The maximum bending stresses for the 21.7-percent reduction ball
were only 130 MN/m^2 (18 800 psi) in tension and 52 MN/m^2 (7600 psi) in
compression in the bearing tests, and slightly less in the fatigue
tests. The actual stresses for this ball were apparently below the mate-
rial endurance limit, since there were no flexure failures. In fact, the
endurance limit in bending for smooth specimens of the SAE 52100 and AISI
M-50 bearing materials used in Refs. [2] and [3] is about 620 MN/m^2
(90 000 psi) according to Ref. [11]. However, it should be remembered
that the hollow bearing balls were electron-beam welded and that stress
raisers thus existed on the inner surface. Therefore, the actual stresses
in the bearing and fatigue tests would certainly be higher than the cal-
culated stresses, due to the effect of these stress raisers.

The stresses calculated for the 56.5-percent reduction ball were
about three times those for the 21.7-percent reduction ball. The maximum
values were 298 MN/m^2 (43 200 psi) in tension and 72 MN/m^2 (10 400 psi)
in compression in the fatigue tests, and were considerably higher in the
bearing tests. Since flexure failures occurred in both the fatigue tests
and the bearing tests, it is apparent that the actual stress exceeded the
endurance limit in both of these cases. It is not unlikely that a stress
concentration factor of at least two was present at the weld area, due to
the effect of the weld bead.

The bending stresses calculated for the 50-percent reduction ball were, as expected, intermediate between the low values of the 21.7-percent ball and the higher values of the 56.5-percent ball, for the respective tests. However, perhaps of more significance is the fact that the calculated stress was 207 MN/m^2 (30 100 psi) in tension and 56 MN/m^2 (8200 psi) in compression in the fatigue tests where no flexure failures occurred, and was 381 MN/m^2 (55 300 psi) in tension and 105 MN/m^2 (15 200 psi) in compression in the bearing tests where flexure failures did occur. Also, since the balls were the same, the stress concentration factor should be similar.

Another interesting evaluation is to compare the calculated stress values for the 56.5-percent ball (with flexure failure) and the 50-percent ball (without flexure failures) for the fatigue tests. From this comparison it would appear that flexure failures would occur for these welded hollow balls if the calculated bending stress in tension is much more than say 200 MN/m^2 (30 000 psi). Use of the maximum bending stress in tension as a criteria for failure would imply a stress concentration factor at the weld of about 3 for the actual stress to exceed the endurance limit.

Based on the above, the bending stresses at the inner surface of the hollow bearing balls, calculated using Golecki's theoretical solution, appear to be consistent with the results of using the hollow balls in actual application.

A direction for further work toward a more thorough understanding of the fatigue stress environment in hollow bearing balls might be to consider the stress range that a given point may experience. The stress range that a single point can experience might be taken as the difference between the maximum tensile stress and the numerically maximum compressive stress, assuming that the ball does not rotate about a single axis. Calculated stresses might be correlated to conventional fatigue tests based on the effective stress concept.

CONCLUSION

Dimensionless stress coefficients for flexural stresses at the interior of hollow spheres calculated from Golecki's analytical result are in good agreement with experimental stress analysis measurements, and seem consistent with the behavior of hollow balls in actual application. Similar calculations are not difficult to carry out and should be a useful first step in evaluating future hollow ball applications.

REFERENCES

1. Brown, P. F., "Discussion of paper previously published in ASLE Transactions," Vol. 12, July 1969, pp. 204-205.

2. 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, Series F, Vol. 93, Jan. 1971, pp. 47-59.

3. Coe, H. H., Parker, R. J., and Scibbe, H. W., "Performance of 75-mm Bore Bearings Using Electron-Beam-Welded Hollow Balls with a Diameter Ratio of 1.26," NASA TN D-7869, National Aeronautics and Space Administration, Washington, D.C., Jan. 1975.

4. Nypan, L. J., "Experimental Evaluation of Stresses in Spherically Hollow Balls," NASA CR-2445, National Aeronautics and Space Administration, Washington, D.C., Aug. 1974.

5. Dally, J. W., and Riley, W. F., "Experimental Stress Analysis," McGraw-Hill New York, 1965.

6. Golecki, J., "The Sphere Weakened by a Concentric Inclusion of Different Elastic Properties Under Concentrated Loads," Archiwum Mechaniki Stosowanej, Vol. 9, 1957, pp. 301-317.

7. Pih, H., and Vanderveldt, H., "Stresses in Spheres with Concentric Spherical Cavities Under Diametral Compression by Three-Dimensional Photoelasticity," Experimental Mechanics, Vol. 6, May 1966, pp. 244-250.

8. Rumbarger, J. H., Herrick, R. C., and Eklund, P. R., "Analysis of the Elastic Contact of a Hollow Ball with a Flat Plate," Journal of Lubrication Technology, Trans. ASME, Series F, Vol. 92, Jan. 1970, pp. 138-144.

9. Ling, F. F., Chow, C. Y., and Rivera, M. E., "Elastodynamic Response of Hollow Balls," ASLE Preprint 74LC-4A-1.

10. Chiu, Y. P., "On the Axial Symmetric Contact Problem Involving Elastic Hollow Spheres," ASME Paper 74-Lub-23.

11. Sachs, G., Sell, R., and Weiss, V., "Tension, Compression and Fatigue Properties of Several SAE 52100 and Tool Steels Used for Ball Bearings." NASA TN D-239, National Aeronautics and Space Administration, Washington, D.C., Feb. 1960.

TABLE 1. – STRESSES IN FATIGUE TESTS AND BEARING EXPERIMENTS

	21.7		56.5		50.0	
Mass reduction, percent	21.7		56.5		50.0	
Ball diameter, mm (in.)	12.7	(0.500)	17.5	(0.6875)	17.5	(0.6875)
Wall thickness, mm (in.)	2.5	(0.10)	1.5	(0.06)	1.8	(0.07)
Golecki stress coefficient	+16.8	−6.8	+110.7	−26.6	+77.1	−20.9
Fatigue tests						
Ball load, N (lb)	756	(170)	645	(145)	645	(145)
Maximum Hertz stress, MN/m^2 (ksi)	5500	(800)	4800	(700)	4800	(700)
Tension bending stress, MN/m^2 (ksi)	100	(14.5)	298	[a](43.2)	207	(30.1)
Compression bending stress, MN/m^2 (ksi)	41	(5.9)	72	[a](10.4)	56	(8.2)
Bearing tests						
Bearing load, N (lb)	4450	(1000)	2220	(500)	2220	(500)
Shaft speed, rpm	18 000		24 000		28 000	
Ball load, N (lb)	979	(220)	890	(200)	1200	(270)
Maximum Hertz stress, MN/m^2 (ksi)	1650	(240)	1280	(185)	1410	(205)
Tension bending stress, MN/m^2 (ksi)	130	(18.8)	454	[a](65.8)	381	[a](55.3)
Compression bending stress, MN/m^2 (ksi)	52	(7.6)	109	[a](15.8)	105	[a](15.2)

[a]Flexural fatigue failures were observed.

◫ STRAIN GAGE ROSETTE LOCATION

LOAD LOCATION 3 △
LOAD LOCATION 2 ◇
LOAD LOCATION 4 ▽
LOAD LOCATION 1 □

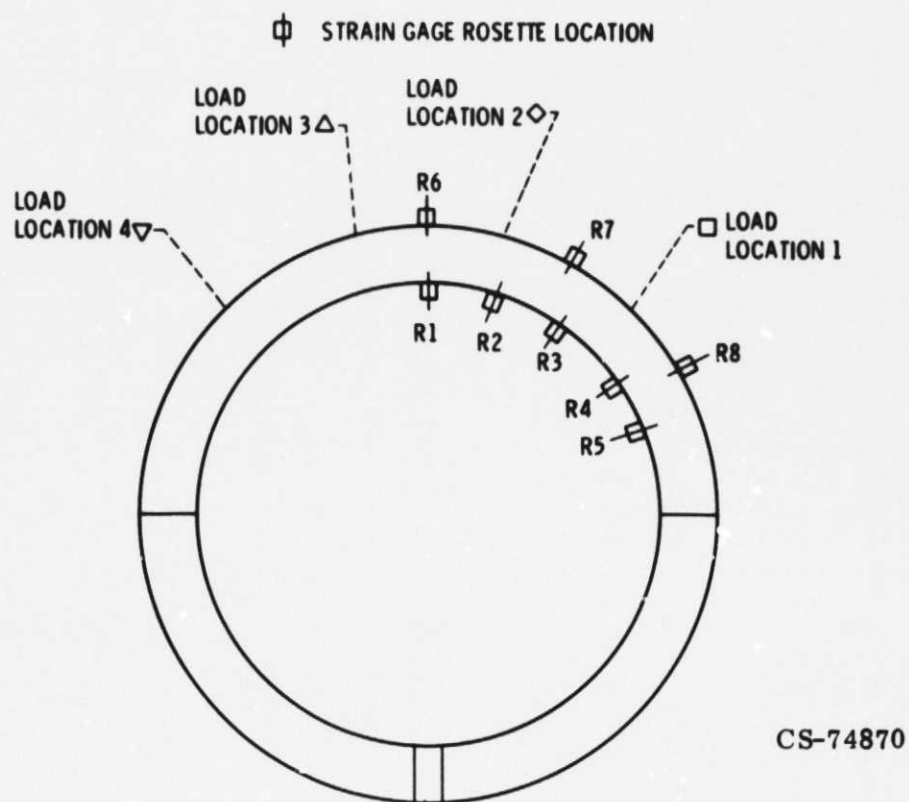R6
R7
R1
R2
R3
R8
R4
R5

CS-74870

Figure 1. - Typical rosette and load locations.
50% mass reduction model.

CS-74869
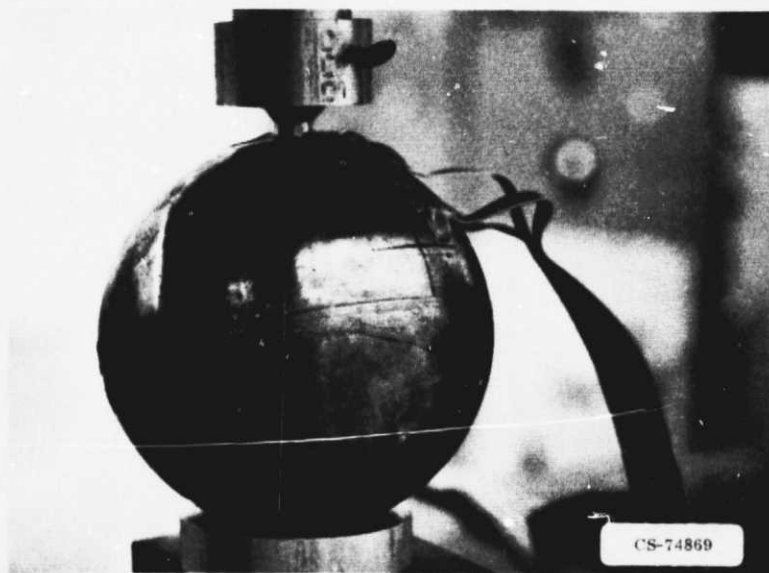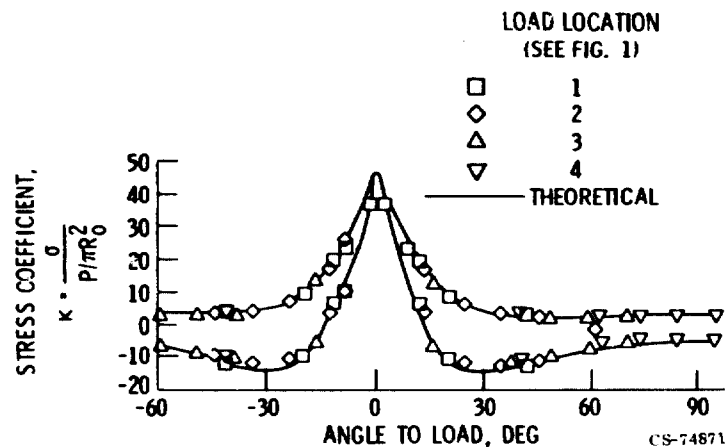
Figure 2. - Model positioned for testing.

E-8582

Figure 3. – Dimensionless principal stress coefficients for 40% mass reduction model.
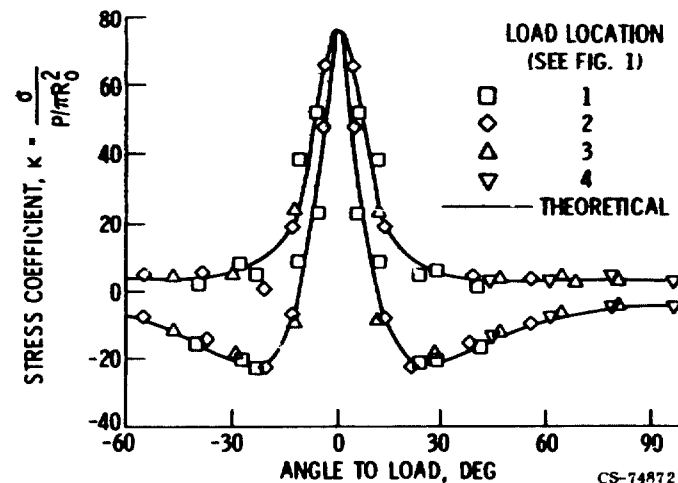
CS-74871

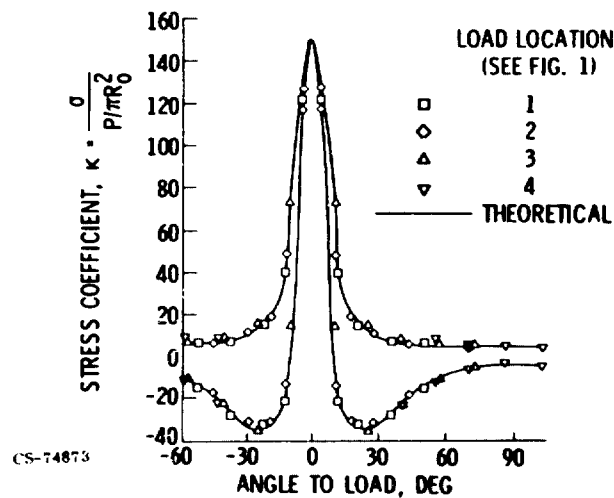Figure 4. – Dimensionless principal stress coefficients for 50% mass reduction model.

CS-74872

Figure 5. – Dimensionless principal stress coefficients for 60% mass reduction model.

CS-74873