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Ceramic Bearings for Use in Gas Turbine Engines[1]

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with Discussions by
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Abstract. Three decades of research by U.S. industry and government laboratories have produced a vast array of data related to the use of ceramic rolling element bearings and bearing components for aircraft gas turbine engines. Materials such as alumina, silicon carbide, titanium carbide, silicon nitride, and a crystallized glass ceramic have been investigated. Rolling element endurance tests and analysis of full complement bearings have been performed. Materials and bearing design methods have continuously improved over the years. This paper reviews a wide range of data and analyses with emphasis on how early NASA contributions as well as more recent data can enable the engineer or metallurgist to determine just where ceramic bearings are most applicable for gas turbines.

NOMENCLATURE

C = dynamic load capacity, N (lb)

$E = \left[\frac{1}{2} + \frac{Ys(1-\delta_c^2)}{2Yc(1-\delta_s^2)}\right]$

e = Weibull slope or modulus

$K_1 = \left[\frac{3P_oR}{8}\right]^{1/3}$ $N^{1/3} \cdot M^{1/3}$(in.$^{1/3}$-lb$^{1/3}$)

$K_2 = \frac{3}{8\pi}\left[\frac{P_o}{9R^2}\right]^{1/3}$ $N^{1/3} \cdot M^{1/3}$(psi$^{1/3}$)

$\bar{L}$ = life, hr, millions of inner-race revolutions or millions of stress cycles

$\bar{L}_{H_R}$ = relative life, hybrid bearing

m = temperature life exponent

$N = \frac{4(1-\delta^2)}{Y}$, M^2N^{-1}(psi^{-1})

n = stress life exponent

P = applied or equivalent bearing load, N (lb)

P_o = normal load, N (lb)

R = radius of a sphere, m (in.)

r = Hertzian contact radius, m (in.)

S = Hertzian contact stress, N/m^2 (psi)

T = temperature, K (°R)

V = stressed volume, m^3 (in.^{3})

Y = Young's modulus of elasticity, N/m^2 (psi)

Z = depth to maximum shear stress, m (in.)

δ = Poisson's ratio

τ = maximum shear stress, N/m^2 (psi)

Subscripts

a, b = bodies a and b

c = ceramic material

H = hybrid bearing

s = steel material or steel bearing

INTRODUCTION

Ceramic materials offer some potential advantages for rolling element bearing components because of their capability of operating over a wide temperature range and their low density relative to rolling element bearing steels. The low density of ceramics may make them attractive as ball or roller materials for very high speed bearings. This benefit is due to the fact that the fatigue life of very high speed ball bearings can be reduced as a result of excessive centrifugal force on

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the balls and subsequent increased stress at the outer race (Harris [1]). Lower mass balls can diminish this fatigue life reduction.

Ceramic materials generally maintain their strength and corrosion resistance over a range of temperatures much greater than typical rolling element bearing steels. Taylor et al. [2] were the first to evaluate hot pressed silicon carbide and hot pressed alumina for rolling element bearings to temperatures above 811 K (1000° F).

A crystallized glass ceramic was examined in rolling element fatigue by Carter and Zaretsky [3] and Zaretsky and Anderson [4] of NASA Lewis Research Center. The results of these early tests showed that the failure mode of ceramics was similar to that in bearing steels. That is, the failure was cycle dependent and apparently of subsurface origin, occurring at the depth of the maximum shear stress. The failure manifested itself as a spall that was limited to the depth of the maximum shear stress and in diameter to the width of the contact zone. The life of the ceramic material was less than 10% of that of a typical rolling element bearing steel under the same conditions of stress. However, the scatter in life (time to failure) was much less than that experienced by bearing steels. Appledoorn and Royle [5] confirmed these results in a later study.

Parker et al. [6] of NASA also conducted studies with three ceramics and one cermet for high temperature, rolling element bearing applications. The ceramic materials were hot pressed and cold pressed alumina, both 99% pure, and a two-phase sintered silicon carbide. Endurance tests were also conducted with the hot pressed alumina to 1367 K (2000° F). The time to failure for the ceramics was found to have less scatter than for bearing steels. The mode of failure was a spall which was attributed to a surface condition rather than subsurface shear stresses. Hot pressed alumina performed the best of the four materials. However, the life of this material was only 7% of that of a typical bearing steel at the same condition of stress. It was concluded that the life of these refractory ceramic materials was related to the porosity, surface finish, and homogeneity of the material [6].

Baughman and Bamberger [7] performed unlubricated high temperature bearing studies to 922 K (1200° F) in the rolling contact (RC) tester and full scale needle bearings. They tested the following materials: Star-J and Stellite 25 (super alloys) and silicon carbide and alumina ceramics. The silicon carbide was shown to be the most wear resistant material. However, there was a lack of silicon carbide material homogeneity which resulted in nonuniform results.

In 1970, hot pressed silicon nitride was proposed by Dee [8] for rolling element bearings as well as for journal bearings. Rolling element fatigue testing of hot pressed silicon nitride has resulted in seemingly contradictory results. Poor results were obtained in the limited tests reported in Scott et al. [9] and Scott and Blackwell [10]. The results reported in Baumgartner [11] and Baumgartner et al. [12] showed the rolling element fatigue life of hot pressed silicon nitride to exceed that of a typical rolling element bearing steel. Extrapolation of the experimental results of Parker and Zaretsky [13] to contact loads which result in stress levels typical of those in rolling element bearing applications indicate that hot pressed silicon nitride running against steel may be expected to yield fatigue lives comparable to or greater than those of bearing quality steel running against steel.

Concurrent with the work of Dee [8], Scott et al. [9], Scott and Blackwell [10], Baumgartner [11], Baumgartner et al. [12], and Parker and Zaretsky [13], hybrid bearings comprising silicon nitride rolling elements and steel races were manufactured and tested (Baumgartner et al. [12,14]; Baumgartner and Cowley [15]; Reddecliff and Valori [16]; and Miner et al. [17] as well as silicon nitride rolling elements and rings (Baumgartner et al. [12] Miner et al. [17]; Hosang [18]; and Bailey [19]).

In view of the aforementioned it is the objective of the work reported herein to summarize the data and analyses related to ceramic bearings for use in gas turbine engines. Emphasis is placed on how early NASA contributions as well as more recent data can enable the engineer or metallurgist to determine just where ceramic bearings are most applicable for future gas turbine engines.

EFFECT OF CONTACT STRESS

It has long been established that the rolling element fatigue life, $\bar{L}$, of a rolling element is inversely proportional to stress, S, to a power, n, that is:

$$\bar{L} \sim S^{-n} \tag{1}$$

For bearing steels the accepted value in the bearing industry for the stress life exponent is 9 to 10. However, variations in this value have been noted and may be a function of material processing (Parker and Zaretsky [20]). A summary of the stress life exponents for the ceramic and cermet materials tested for rolling element bearing application which were compiled from Carter and Zaretsky [3], Zaretsky and Anderson [4], Parker et al. [6], and Baumgartner et al. [12] are summarized in Table 1.

When steel and other metallic materials are tested at different stress levels, direct comparisons of fatigue lives can be made only by adjusting one of the lives

Table 1. Stress Life Exponent and Relative Dynamic Capacity of Materials for Rolling Element Bearing Application

Material	Stress Life Exponent, n	Relative Dynamic Capacity to Steel
Crystallized Glass ceramic[a]	10.5 to 13.8 (Average value, 11.6)	0.07
Hot pressed alumina[b]	9.4 to 10.8 (Average value, 10.6)	0.07
Cold pressed alumina[b]	6.0 to 8.1 (Average value, 7)	0.01
Self-bonded silicon carbide[b]	6.9 to 8.6 (Average value, 7.8)	0.01
Nickel-bonded titanium carbide[b]	9.7 to 10.5 (Average value, 10.2)	0.03
Silicon nitride[c]	16 to 16.2 (Average value, 16.1)	0.05 to 0.12
Bearing steel	9 to 10 (Accepted value, 9)	1.00

[a][3,4].
[b][6].
[c][12,13,24].

using the proper stress life exponent for that material (given in Table 1), or by making a comparison on the basis of dynamic load capacity. The dynamic load capacity, contact load, and life are related by the equation (Lundberg and Palmgren [21–23])

$$C = P\sqrt[n/3]{\bar{L}} \tag{2}$$

where

C = dynamic load capacity or load which will produce failure of 10% of test specimen in one million stress cycles, N (lb)
P = applied or equivalent bearing load, N (lb)
n = exponent relating stress and life, determined experimentally
$\bar{L}$ = life in millions of stress cycles which 90% of a group of specimens survive or within which time 10% fail

For steels, $n/3$ is usually taken as 3 (based on a stress life exponent of 9). The values of n for each of the materials can be obtained from Table 1. The relative dynamic capacities for each of the ceramic materials are then given based upon typical bearing steels. Knowing the dynamic capacity of an equivalent steel bearing, the value of the steel bearing can be multiplied by the relative dynamic capacity of the selected material from Table 1. Using the resultant value, the applicable stress life exponent, n, and the applied bearing load, P, in Eq. (1), an estimate of the ceramic bearing life, $\bar{L}$, in millions of inner-race revolutions, can be obtained.

There is conflicting data with regard to the life of the silicon nitride material. Figure 1 (Parker and Zaretsky [24]) shows a comparison of life data for hot pressed silicon nitride and for typical bearing steels, consumable electrode vacuum melted (CVM) AISI 52100 and AISI M-50 (Parker and Zaretsky [36]) at a maximum Hertz stress of 5.52×10^9 N/m^2 (800,000 psi). The 10% fatigue life of the silicon nitride balls was approximately one-eighth that of the AISI 52100 balls and approximatley one-fifth that of the AISI M-50 balls. Figure 2 (Baumgartner et al. [12]) shows results from the rolling contact fatigue (RC) tester for silicon nitride and AISI M-50. These results show that the life of the AISI M-50 material was approximately one-eighth the life of the silicon nitride material at a maximum Hertz stress of 4.83×10^9 N/m^2 (700,000 psi). However, the data for Baumgartner et al. [12] and Parker and Zaretsky [24] show a stress life exponent, n, of 16.2 and 16, respectively. Based upon both sets of data, the dynamic capacity (or load carrying ability) of the silicon nitride material would be significantly less than that of a typical bearing steel. This is shown in Table 1 where relative C values have been calculated and compared to a bearing steel.

EFFECT OF ELASTIC PROPERTIES

The physical and thermal properties of ceramic and cermet materials considered for rolling element bear-

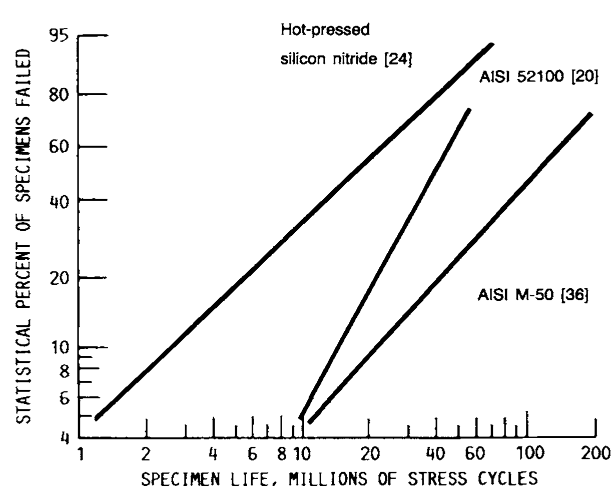

Fig. 1. Rolling element fatigue life of hot pressed silicon nitride balls and steel balls in five-ball fatigue tester. Maximum Hertz stress, 5.52×10^9 N/m^2 (800000 psi); shaft speed, 9400 rpm; race temperature, 328 K (130° F); contact angle, 30 deg; lubricant, superrefined naphthenic mineral oil. [24]

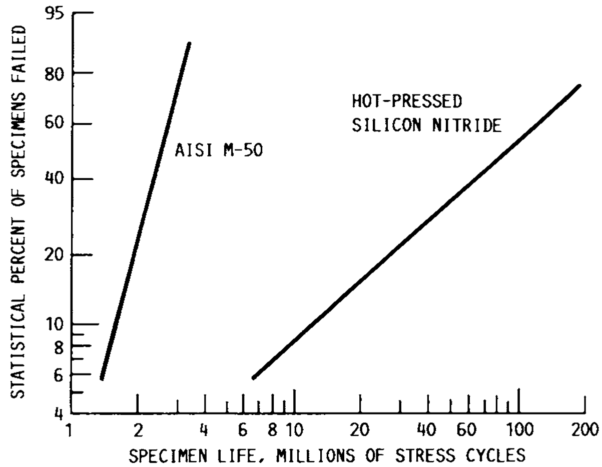

Fig. 2. Rolling element fatigue life of hot pressed silicon nitride and steel rollers in rolling contact (RC) fatigue tester. Maximum Hertz stress, 4.83×10^9 N/m^2 (700000 psi); roller speed, 10000 rpm; temperature, ambient; lubricant, triester (MIL-L-23699B). [12]

ing application are given in Table 2. The properties are a compilation from (Carter and Zaretsky [3]; Parker et al. [6]; Parker and Zaretsky [24]; Parker et al. [25,26]; Sibley et al. [27]; and Bhushan and Sibley [28]). The elastic modulus of most ceramics is much greater than that of a bearing steel. Consequently, the resultant contact or Hertz stress will be different for a given loading of ceramic on ceramic, ceramic on steel, or steel on steel. This was first recognized by Carter and Zaretsky [3] in their work with a crystallized glass ceramic. For the case of a ceramic on steel, it is assumed that the ceramic material will have infinite life and that the steel races will be the element to fail from rolling element fatigue. Since life is inversely proportional to stress to a power, the life of a hybrid bearing (ceramic rolling element on steel races) will generally be lower than that for a full complement steel bearing. This can be illustrated as follows:

From Hertz theory for two spheres of radii R_a and R_b in contact (Jones [29]), the maximum compressive stress is

$$S_{\max} = \frac{3P_o}{2\pi r^2} \tag{3}$$

where

P = normal load
r = contact radius

For two spheres in contact, the Hertzian contact area is a circle with radius r.

$$r = \left[\frac{3P_o(N_a + N_b)}{8\left(\dfrac{2}{R_a} + \dfrac{2}{R_b}\right)}\right]^{1/3} \tag{4}$$

For two spheres of equal radii

$$R_a = R_b = R$$

Then

$$r = \left[\frac{3P_o R(N_a + N_b)}{32}\right]^{1/3} \tag{5}$$

where

$$N_a = \frac{4(1 - \delta_a^2)}{Y_a}$$
$$N_b = \frac{4(1 - \delta_b^2)}{Y_b}$$

and

Y = modulus of elasticity
δ = Poisson's ratio

Then

$$r = K_1\left[\frac{1 - \delta_a^2}{Y_a} + \frac{1 - \delta_b^2}{Y_b}\right]^{1/3} = K_1\left[\frac{Y_b(1 - \delta_a^2) + Y_a(1 - \delta_b^2)}{Y_aY_b}\right]^{1/3} \tag{6}$$

where

$$K_1 = \left[\frac{3P_oR}{8}\right]^{1/3}$$

Substituting Eq. (6) into Eq. (3),

$$S_{\max} = K_2\left[\frac{Y_aY_b}{Y_b(1 - \delta_a^2) + Y_a(1 - \delta_b^2)}\right]^{2/3} \tag{7}$$

where

$$K_2 = \frac{3}{8\pi}\left[\frac{P_o}{9R^2}\right]^{1/3}$$

For steel on steel

$$S_{\max_s} = K_2\left[\frac{Y_s}{2(1 - \delta_s^2)}\right]^{2/3} \tag{8}$$

Table 2. Typical Physical and Thermal Properties of Materials for Rolling-Element Bearing Application [3,4,12,28,26,6,13,24]

Material	Rockwell C Hardness at 294 K (70° F)	Estimated[a] Maximum Usable Bearing Temperature, K (°F)	Density, gm/cc	Elastic Modulus at 294 K (70° F), GPa (10^6psi), Y	Poisson's Ratio, w	Thermal Conductivity, w/mK (BTu ft/hr-ft^2-F) at 294 K (70° F)	Thermal Conductivity, w/mK (BTu ft/hr-ft^2-F) at 1073 K (1471° F)	Coefficient of Thermal Expansion, 10^{-6} K (10^{-6}/°F) 273 to 1073 K (32 to 1471° F)	Weibull[b] Slope or Modulus, e
Crystallized glass ceramic	53	>644 (700)	2.5	87 (12.5)	0.25	1.6 (0.9)	2.0 (1.2) at 873 K (1112° F)	0.4 (0.2)	3.3
Alumina	85	N1376 (2000)	3.9	350 (51)	0.25	7.2 (4.2)	1.7 (1.0)	8.5 (4.7)	2.7
Silicon carbide	90	<1367 (2000)	3.2	410 (59)	0.25	35 (20)	12 (6.9)	5.0 (2.8)	2.1
Nickel-bonded titanium carbide	67	<867 (1100)	6.3	300 (57)	0.23	14 (8.1)	6.8 (3.9)	10.7 (5.9)	1.4
Silicon nitride	78	N1367 (2000)	3.11 to 3.24	310 (45)	0.26	7.3 (4.2)	4.7 (2.7)	2.9 (1.6)	1.7
Bearing steel (AISI M-50)	63	>589 (600)	7.6	190 (28)	0.28	13.4 (7.7)	...	12.3 (6.8)	1.1

[a]Based primarily on hardness retention and test experience.
[b]Based upon rolling element fatigue testing.

The relative life, $\bar{L}_{H_R}$, for the same load on a steel bearing and a hybrid bearing would be as follows:

$$\frac{\bar{L}_{H_R}}{\bar{L}_s} = \left(\frac{S_{\max_s}}{S_{\max_H}}\right)^n \tag{9}$$

Letting $\bar{L}_s = 1$ and $n = 9$ and substituting Eq. (8) in Eq. (9),

$$\bar{L}_{H_R} = \left[\frac{1}{2} + \frac{Y_s(1 - \delta_c^2)}{2Y_c(1 - \delta_s^2)}\right]^6 \tag{10}$$

Equation (10) is a stress correction factor based upon a ninth power of the ratio of the Hertz stress in the contact of a ceramic or cermet ball or roller and a steel race to that of a steel rolling element on a steel race for identical contact load and geometry. The factor $\bar{L}_{H_R}$ for various rolling element materials is listed in Table 3. $\bar{L}_{H_R}$ can be multiplied by the calculated life of a full complement or all steel rolling element bearing to obtain the life of a hybrid bearing using the applicable material in Table 3 as the rolling elements. This simplified correction neglects the effect of stressed volume on fatigue life. However, this effect is found to be small relative to the stress effect. In fact, it is easily shown that the small stressed volume effect is nearly offset by the additional affect of the depth of the maximum shear stress (Parker and Zaretsky [13]). From the Lundberg-Palmgren analysis [21–23],

$$\frac{\bar{L}_{H_R}}{\bar{L}_s} = \left(\frac{Z_H}{Z_s}\right)^{2.1}\left(\frac{V_s}{V_H}\right)^{0.9}\left(\frac{\tau_s}{\tau_H}\right)^{9.3} \tag{11}$$

where

$\bar{L}$ contact life
Z depth to maximum shear stress
V stressed volume
τ maximum shear stress

From the Hertzian equations for line contact (Jones [29]) where the ellipticity ratio is 0 and from Eq. (10) letting

$$E = \left[{}^1\!/_2 + \frac{Y_s(1 - \delta_c^2)}{2Y_c(1 - \delta_s^2)}\right] \tag{12}$$

then

$$\frac{Z_H}{Z_s} = E^{2/3} \tag{13}$$

$$\frac{V_s}{V_H} = E^{-2/3} \tag{14}$$

$$\frac{\tau_s}{\tau_H} = E^{2/3} \tag{15}$$

Substituting Eqs. (12) to (15) into Eq. (11) and letting $\bar{L}_s = 1$, then for line contact,

$$\bar{L}_{H_R} = \left[{}^1\!/_2 + \frac{Y_s(1 - \delta_c^2)}{2Y_c(1 - \delta_s^2)}\right]^{6.97} \tag{16}$$

For point contact where the ellipticity ratio is 1 (Jones [29]),

$$\frac{Z_H}{Z_s} = E^{1/3} \tag{17}$$

$$\frac{V_s}{V_H} = E^{-2/3} \tag{18}$$

$$\frac{\tau_s}{\tau_H} = E^{2/3} \tag{19}$$

Equations (18) and (19) are identical to Eqs. (14) and (15), respectively. Substituting Eqs. (12) and (17) to (19) into Eq. (11) and letting $\bar{L}_s = 1$, for point contact,

$$\bar{L}_{H_R} = \left[{}^1\!/_2 + \frac{Y_s(1 - \delta_c^2)}{2Y_c(1 - \delta_s^2)}\right]^{6.3} \tag{20}$$

Using Eq. (10) and the elastic properties listed in Table 2, the relative lives and dynamic capacities of a hybrid bearing comprising rolling elements of the materials listed are given in Table 3.

Hybrid 57 mm bore cylindrical roller bearings each containing 20 each crowned, 7.1 mm diameter silicon

Table 3. Relative Life and Dynamic Capacity of Hybrid Bearing with Various Rolling Element Materials

Raceway Material	Rolling Element Material	Relative Life from Eq. (10),[a] $\bar{L}_{HR}$	Relative Dynamic Capacity to Steel from Eq. (2),[a] $n = 9$
Steel	Crystallized glass ceramic	18	2.6
	Alumina	0.22	0.6
	Silicon carbide	0.16	0.5
	Nickel-bonded titanium carbide	0.18	0.6
	Silicon nitride	0.29	0.7
	Steel	1	1

[a]Based upon failure of steel raceway and assuming no failure of rolling elements.

nitride rollers were endurance tested (Baumgartner and Cowley [15]). The bearing rings were consumable electrode vacuum melted (CVM) AISI M-50. Test conditions were at an outer-race temperature 366 to 380 K (200 to 225° F), a radial load of 22,464 N (5050 lb) producing a maximum Hertz stress of 2.82×10^9 N/m^2 (408,000 psi), a shaft speed of 5400 rpm and a triester lubricant (MIL-L-23699B). The catalog life of an equivalent steel bearing was calculated to be 21.6 hr without any life adjustment factors (Baumgartner and Cowley [15]). Using the ASME design guide (Bamberger et al. [30]), the following life adjustment factors are obtained: (a) material factor, 2; (b) processing factor, 3; and (c) lubricant factor, 1.5. Combining these factors ($2 \times 3 \times 1.5$), a life adjustment factor of 9 can be used for a CVM AISI M-50 steel bearing under these operating conditions. The adjusted predicted life of an all-steel bearing would equal (9×21.6) 194.4 hr at a 90% probability of survival. From Eq. (10) and Table 3 for silicon nitride, the life adjustment factor for the hybrid bearing is 0.29. Hence, the predicted life of the hybrid roller bearing would be (0.29×194.4) 56.4 hr. Using Eq. (16) for line contact, and Eq. (20) for point contact, the predicted lives would be (0.24×194.4) 46.7 and (0.28×194.4) 53.5 hr, respectively. From Figure 3, the life of the hybrid bearing was approximately 48 hr. There were an unusual number of silicon nitride roller failures and roller damage in many of the failed bearings including roller fracture. These results would indicate that the high Hertzian stress at which the bearing was run exceeded the capability of the silicon nitride material.

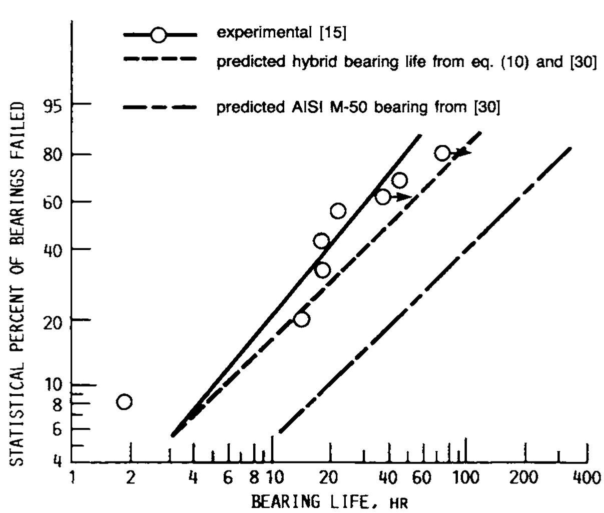

Fig. 3. Comparison of experimental and predicted lives of hybrid 57 mm bore cylindrical roller bearing. Radial load, 22464 N (5050 lb); maximum Hertz stress, 2.82×10^9 N/m^2 (408000 psi); shaft speed, 5400 rpm; lubricant, triester; outer-race temperature, 380 K (255° F).

EFFECT OF TEMPERATURE

Liquid and solid lubricated rolling element tests were conducted in the NASA five-ball fatigue tester with the crystallized glass ceramic, hot pressed alumina, cold pressed alumina, self-bonded silicon carbide, and nickel-bonded titanium carbide at temperatures to 1366 K (2000° F) (Carter and Zaretsky [1]; Zaretsky and Anderson [4]; and Parker et al. [6,25,26]). The crystallized ceramic was only tested to 644 K (700° F) [3,4]. There are no fatigue results reported for silicon nitride beyond nominal temperature under lubricated conditions. Shorter lives were exhibited at 644 K (700° F) for these materials. This decrease in life may be accounted for by a change in lubricant viscosity with temperature. As the viscosity of the liquid lubricant decreases, any elastohydrodynamic film separating the rolling elements will decrease. The life of a rolling element in addition to its own material properties is a function of the elastohydrodynamic film thickness (Bamberger et al. [30]). Hence, the life of these materials may be affected in a manner similar to steel's.

Three materials were run in a modified five-ball fatigue tester to temperatures of 1366 K (2000° F) with molybdenum disulfide–argon mist lubrication (Parker et al. [6,25,26]). The results of these tests indicated that the hot pressed alumina was capable of operating to temperatures of 1366 K (2000° F). However, tests with the cold pressed alumina and silicon carbide at 1366 K (2000° F) and maximum Hertz stresses of 1.66×10^9 N/m^2 (270,000 psi) resulted in general track deterioration unlike the failure pits or spalls observed at 300 and 644 K (80 and 700° F). Titanium carbide cermet at temperatures beyond 866 K (1100° F) and a maximum Hertz stress of 1.91×10^9 N/m^2 (310,000 psi) exhibited excess cumulative plastic deformation, which indicated that this material is limited to less severe conditions of temperature and stress.

Surface failure data with hot pressed alumina tested at 1366 K (2000° F) and a maximum Hertz stress of 3.39×10^9 N/m^2 (550,000 psi) are given in Figure 4(a) together with the experimental lives at 300 and 644 K (80 and 700° F) (at the same stress) but with a mineral oil lubricant. Figure 4(b) is a plot of the 10% and 50% lives of the material as a function of temperature. While the mode of lubrication at the lower temperatures and 1366 K (2000° F) is different, the figure provides not only a relative indication of the life performance of the hot pressed alumina to 1366 K (2000° F), but also of the other refractory materials with temperature where $\bar{L} \sim 1/T^m$. From Figure 4(b), $m = 1.8$; where temperature is in Kelvin or Rankine.

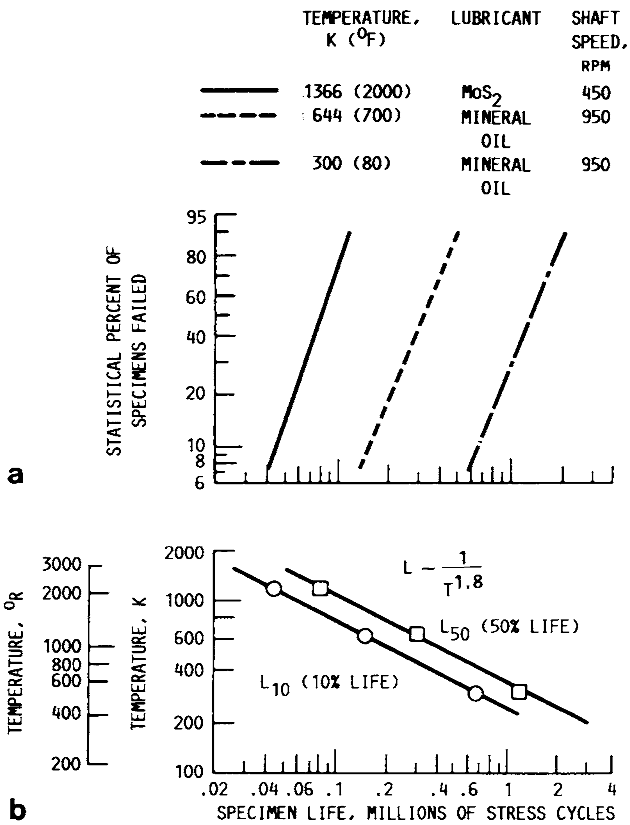

Fig. 4. Rolling element fatigue life of hot pressed alumina balls in five-ball fatigue tester. Maximum Hertz stress, 3.39 $\times$ 10^9 N/m^2 (550000 psi); contact angle, 20 deg. (a) Life distribution at temperature, (b) effect of temperature. [6]

EFFECT OF SPEED ON LIFE

At high aircraft turbine engine speeds, the effect of centrifugal loading of the rolling elements of a bearing against the bearing outer race becomes extremely important. Theoretical life calculations for a 150 mm bore angular-contact ball bearing operating at 3 million *DN* (20,000 rpm) (*DN* is bearing speed in revolutions/minute $\times$ bearing bore in millimeters) predict that this bearing has approximately 20% AFBMA calculated life (Scibbe and Zaretsky [31]). The decrease in predicted life is due to the increased stress in the outer race caused by centrifugal effects. The expected final result is extremely short bearing life at speeds much above 2 million *DN* both in actual running time (hours) and in total bearing inner-race revolutions. In order to reduce the centrifugal force effects, concepts such as hollow and hollowed rolling elements and lightweight ceramic rolling elements have been considered as a substitute for the conventional rolling elements contained within the bearing. Silicon nitride has been one such material.

Computer analysis of the dynamic performance characteristics of ball bearings was used to evaluate the effect of the low mass silicon nitride balls on 120 mm bore angular-contact ball bearing fatigue life (Parker and Zaretsky [13]). The analysis was performed with both steel and silicon nitride balls with steel inner and outer races. Figure 5 is a summary of the results for three thrust loads. In general, this analysis indicates that the use of silicon nitride balls to replace steel balls in high speed bearings will not yield an improvement in fatigue life over the speed range of anticipated advanced air-breathing engine main-shaft ball bearings up to 3 million *DN*. However, at some

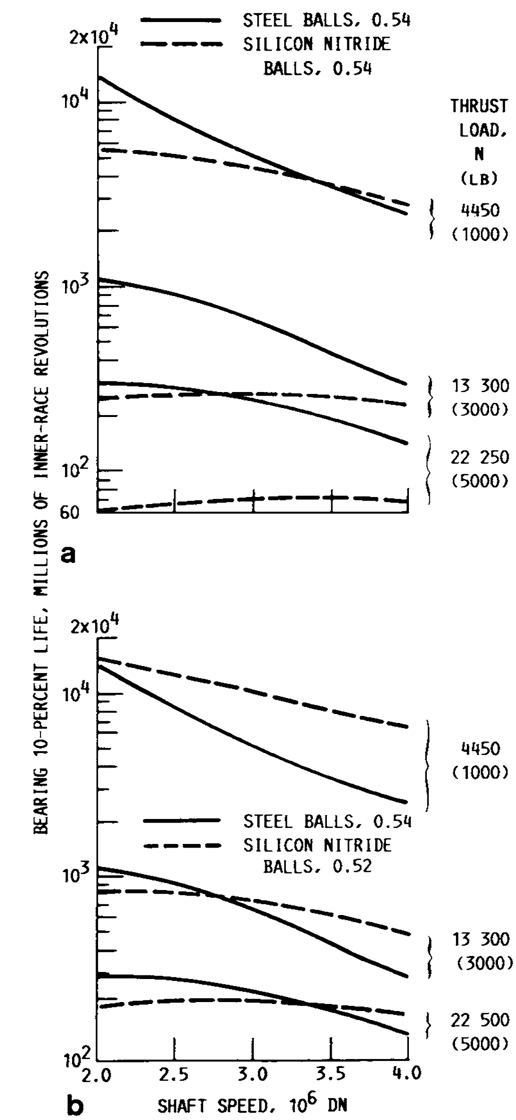

Fig. 5. Predicted life of 120 mm-bore angular-contact ball bearing with silicon nitride balls. Ball diameter, 20.65 mm (0.8125 in.). (a) Steel balls and silicon nitride balls with inner-race curvature of 0.54, (b) Steel balls and silicon nitride balls with inner-race curvatures of 0.54 and 0.52, respectively. [13]

conditions of very high speeds and light loads, modest life improvements are indicated, but only if modifications are made in bearing internal geometry (inner-race curvature, for example) [13].

Bearing life calculations were made with reduced curvature at the inner race (0.52 as opposed to 0.54) [13]. With the exception of very low loads and very high speeds, the life improvements over the steel ball cases are small. For the case of 3 million *DN* and 13,300 N (3000 lb) thrust load, the life improvement is less than 14% [13].

HEAT GENERATION AND TEMPERATURE

In a discussion to Reddecliff and Valori [16], Coe and Zaretsky reported results obtained with a 115 series, 75 mm bore ball bearing with steel balls operated at a 4400 N (1000 lb) thrust load up to 2.0 × 10^6 *DN* at NASA. The steel balls were then replaced with silicon nitride balls and the test repeated.

Figure 6(a) is a comparison of the outer-race temperature for both bearings over a range of shaft speed. The outer-race temperature was almost the same for both silicon nitride and the steel balls. However, Figure 6(b) is a comparison of the bearing torque over the same speed range. It is apparent that the bearing with silicon nitride balls showed significantly higher torque than the same bearing with steel balls. The torque was measured directly by a force transducer connected to the periphery of the bearing housing. The higher torque with the silicon nitride balls can be explained in part by the fact that the traction coefficient of a lubricant is a function of the viscosity of the oil under the contact pressure (Loewenthal and Zaretsky [32]). Since for a given load, the contact stress with the silicon nitride balls are higher than with the steel balls, the viscosity of the oil in the contact zone is higher. Because of the higher viscosity, the traction in the contact zone of the bearing must be accordingly higher.

It was reported by Reddecliff and Valori [16] that a 33% reduction in axial preload to prevent skidding in a 35 mm bore angular-contact ball bearing was achieved by substituting silicon nitride balls for steel balls. This would tend to substantiate that a higher traction force in the bearing may exist with a given load as a result of the higher viscosity with higher contact stress. Higher traction forces should result in higher heat generation with the silicon nitride balls. However, data reported in [16] (Fig. 7) show that with the silicon nitride balls in the 35 mm bearing, the heat generation was lower [Fig. 7(a)] even though the outer-race temperatures were almost identical [Fig. 7(b)].

There are possible explanations for the difference

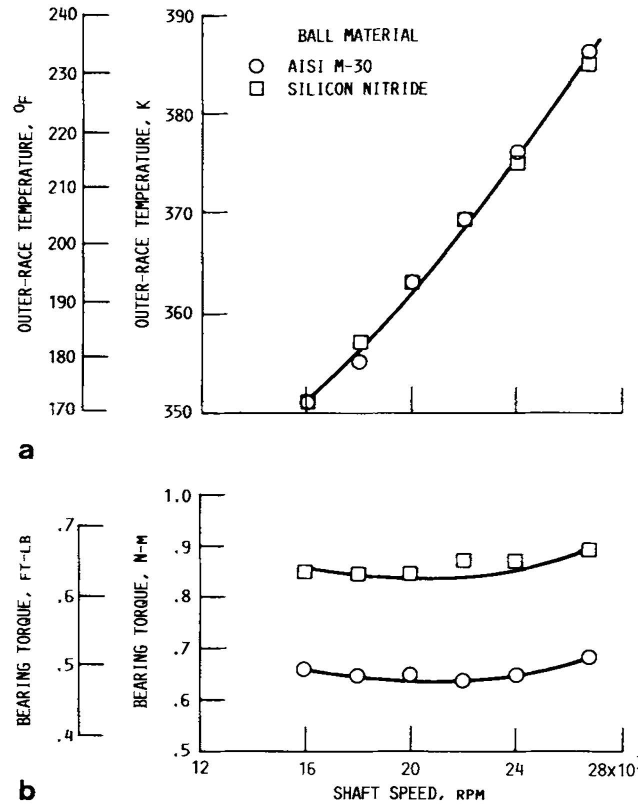

Fig. 6. Performance of 75 mm-bore angular-contact ball bearing as function of shaft speed. Thrust load, 4400 N (989 lb); oil flow rate, 0.9 kg/min (2 lb/min); oil inlet temperature, 316 K (109° F). (a) Outer-race temperature, (b) bearing torque.

in these results. There could, of course, be an effect due to bearing size. For example, the computer program used in Parker and Zaretsky [13] indicates that about 60% of the calculated heat generation was due to ball spin for a 35 mm bore bearing, whereas it was about 50% for a 120 mm bore bearing. Further, the oil used for the 75 mm bearing tests was super-refined naphthenic mineral oil with undoubtedly different viscosity characteristics from those of the oil used in Reddecliff and Valori [16]. Also, the 75 mm bearings were lubricated directly by oil jet, not through the race as were those of Reddecliff and Valori. Finally, it should be noted that the diameter of the silicon nitride balls used in the 75 mm bearing differed less than 0.5 μm (20 μin.) from the steel balls, at room temperature. Therefore, at operating temperature, the bearings were slightly different, due to the lower coefficient of expansion of the silicon nitride.

Data were reported for AISI M-50 35 mm roller bearings (Baumgartner et al. [14]) comparing the heat generated in a hybrid roller bearing with the same size bearing having steel rollers and approximately the same radial load. It was concluded in [14] that the heat gen-

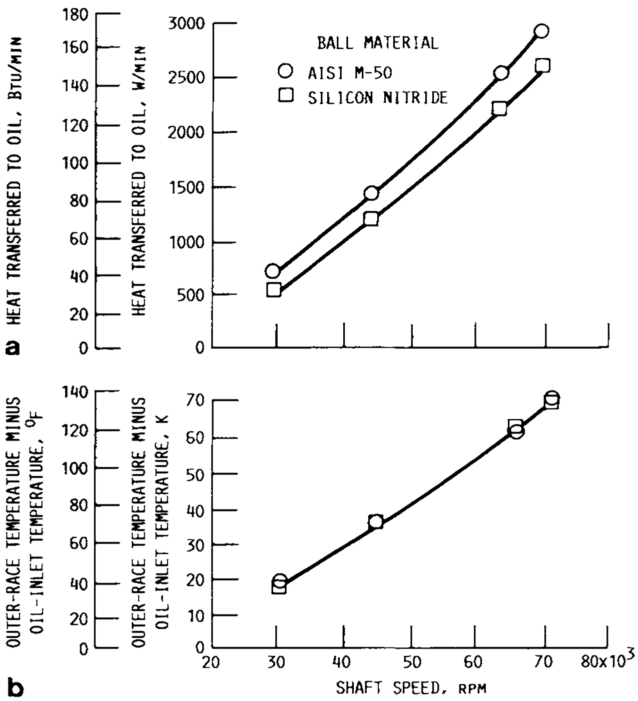

Fig. 7. Performance of 35 mm-bore angular-contact ball bearing as function of shaft speed. Thrust load, 1200 N (270 lb); oil flow rate, 1.1 kg/min (2.5 lb/min); oil inlet temperature, 399 K (150° F). (a) Power loss, (b) outer-race temperature. [16]

eration for the hybrid bearing was comparable to that of the bearing with the steel rollers. It can be concluded that the bearing power loss or heat generation is more a function of the individual bearing design and operation than whether steel or ceramic rolling elements are used within the bearing.

UNLUBRICATED BEARINGS

It has been proposed that unlubricated ceramic bearings offer an approach toward meeting operating requirements in excess of 578 K (600° F) where both conventional and nonconventional liquid lubricants are not capable of sustaining these higher temperatures or providing an elastohydrodynamic film. Tests were performed of full complement 17 mm bore silicon nitride cylindrical roller bearings operated at 644 K (700° F) (Bailey [19]). The test vehicle used to evaluate these bearings was a modified J402 Turbojet engine. The first test resulted in a catastrophic ceramic roller bearing failure after $11^1/_2$ min of operation. The second test ran for a total time of 2 hr and 3 min of which 54 min were run unlubricated. In the unlubricated condition, 30 min were run at shaft speeds between 39,000 and 39,600 rpm or in excess of 660,000 *DN*. However, residuary lubricant may have been present to sustain the bearing for the 54 min. A totally unlubricated endurance test was run with a René 41 gold plated cage. A catastrophic failure was encountered after 30 min of operation.

Solid film lubricants applied in a manner similar to that reported in Parker et al. [6] may be capable of sustaining full complement ceramic roller bearings at these higher temperatures for longer periods of time. However, extensive work is required to both prove and develop this concept for practical turbine engine applications.

BEARING MOUNTING

The use of full complement ceramic rolling element bearings present unique mounting problems. Referring to Table 2, the thermal expansions of the refractory materials are less than that of steel. As a result, where a ceramic bearing is mounted on a steel shaft, large hoop stresses can be induced in the bearing inner ring which can result in fracture of the ring. Hosang [18] proposes the use of a corrugated liner interposed between the journal and the bore of the inner ring. The corrugations run parallel to the bearing and journal axis. This is illustrated in Figure 8. In principle, the liner's diametral thermal expansion is less

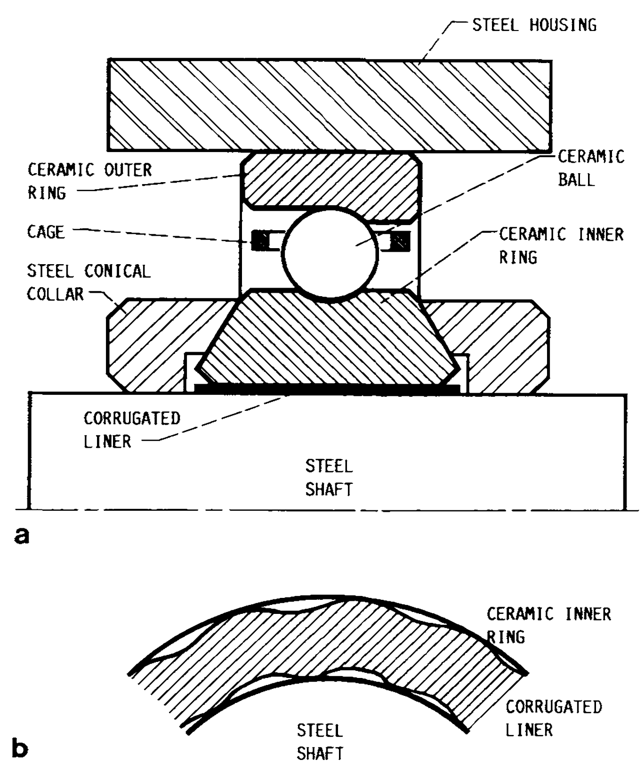

Fig. 8. Mounting arrangement of full complement ceramic bearing on steel shaft, using corrugated liner. (a) Mounting arrangement, (b) corrugated liner. [18]

than that of the shaft journal. The difference is accommodated by stretching of the liner in the circumferential direction. According to Hosang, this action also reduces the envelope outer diameter of the corrugations from that dictated by thermal expansion. The radial stiffness of the liner should also be as high as possible so as not to affect the stiffness of the bearing. An alternate design proposed by Hosang [18] is the use of the corrugated liner and conical retainers shown in Figure 8(a).

Bailey [19] reports the use of a collar for the inner ring and a spacer for the outer ring to accommodate differences in thermal expansion (Fig. 9).

Baumgartner et al. [12] use a clamping collar against the inner ring. The axial clamping force is maintained by the clamping collar with an angled face to match the face angle of the inner ring. As the shaft expands axially, the radial expansion forces the collar against the inner ring face, holding it in position.

The use of full complement ceramic bearings requires special mounting design considerations not currently used in turbomachinery. Consequently, these bearings cannot be substituted for steel bearings without extensive design modifications of the rotating shaft and housing.

MANUFACTURING AND PROCESSING

It has long been recognized that voids or surface defects in ceramic or cermet rolling elements can be the source of a subsurface or surface induced spall (Carter and Zaretsky [3]; Zaretsky and Anderson [4]; Parker et al. [6,25,26]; and Parker and Zaretsky [24]). As critical flaw sizes are reduced, it is probable that the values of dynamic capacity summarized in Table 1

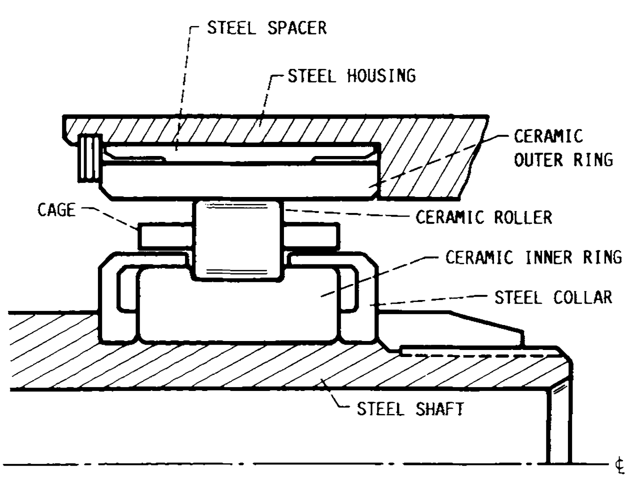

Fig. 9. Mounting arrangement of full complement ceramic roller bearing on steel shaft, using steel collar and spacer. [19]

and life can be increased. Thus, processing and manufacturing methods can be critical to long life functioning.

In recent years, a relatively large effort has been devoted to the processing and manufacture of silicon nitride bearings (Bhushan and Sibley [28]; Dalal et al. [33]; Baumgartner and Cowley [34]; and Baumgartner and Wheidon [35]). The most commonly used processing method is hot processing. In hot processing the powder is sized, blended with hot processing aids and pressed in graphite dies using temperatures in the 1973 to 2173 K (3091 to 3451° F) range and pressures above 14 MPa. The most common sintering aid is the addition of 1% to 2% MgO (Bhushan and Sibley [28]).

Other processes include blending of silicon nitride powder with a binder and then cold pressing to near-net shape preforms. The cold pressed parts are subsequently sintered at high temperature without application of the high pressures present in hot processing [28]. Another processing method is hot isostatic pressing or partial sintering and then hot isostatic pressing [28]. Cold processing silicon and reaction sintering in hot, high pressure nitrogen, called reaction bonding, is still another method.

Silicon nitride is hot pressed into billets in the form of plates and then diamond machined. The material can be hot pressed directly into blanks of the required shape for bearing rings, balls, and rollers using suitable multiple cavity graphite molds (Bhushan and Sibley [28]).

The rolling element fatigue life of silicon nitride was found to be strongly influenced by finishing procedures (Bhushan and Sibley [28]; Baumgartner and Cowley [34]; and Baumgartner and Wheidon [35]). As with steels, improved rolling element fatigue life was obtained with better surface finishes [34,35]. The machining of silicon nitride for bearing application begins with ultrasonic machining, followed by diamond grinding, and then by lapping and polishing [28]. Surface preparation should insure that coarse grit grinding damage is removed during final finish [35].

It was recommended by Baumgartner and Cowley [34] that, for producing silicon nitride rollers with a straight roller geometry, diamond grinding and honing be used. The use of a formed silicon carbide wheel after initial diamond wheel grinding was recommended for shaping crowned roller geometries [34].

SUMMARY

For three decades research has been performed on the use of nonmetallic and refractory materials as rolling element materials for use in gas turbine engines. Materials and bearing design methods have continuously improved over the years. Materials such as alumina, silicon carbide, titanium carbide, silicon nitride, and a crystallized glass ceramic have been investigated by NASA in the past. Rolling element endurance tests and analysis of full complement bearings were performed. The following results were obtained:

1. Silicon nitride material produces the longest life of the materials studied. However, the dynamic capacity of a full complement silicon nitride bearing will be only 5% to 12% that of an all-steel bearing of similar geometry.
2. The use of bearings having ceramic rolling elements and steel races can result in lives less than full complement steel bearings where the elastic modulus of the ceramic is greater than steel as in the case of most ceramics.
3. Bearing power loss or heat generation is more a function of the individual bearing design and operation than whether steel or ceramic rolling elements are used within the bearing.
4. The lives of ceramic rolling elements are an inverse function of temperature. It is suggested, based upon endurance tests with alumina to 1366 K (2000° F), that life is inversely proportional to temperature to the 1.8 power.
5. Unlubricated tests of a full complement silicon nitride bearing at 644 K (700° F) resulted in catastrophic failure after 30 min, suggesting the need for lubrication at elevated temperatures.
6. Special design and mounting requirements are needed to accommodate a full complement ceramic bearing into turbomachinery applications. Optimum designs have yet to be developed.

ACKNOWLEDGMENTS

The author wishes to acknowledge the technical contributions of Richard J. Parker and Salvatore J. Grisaffe who, with other colleagues at the NASA Lewis Research Center, collaborated with him in conducting over the years the NASA nonmetallic bearing research reported herein. Also, the author wishes to thank Mr. Grisaffe for his technical recommendations and comments which have been incorporated throughout this paper.

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Discussion

Y.P. Chiu

Mr. Zaretsky is to be congratulated for his extensive review on research in ceramic rolling element components, in which he has been involved for more than two decades at the NASA Lewis Research Center. While this discusser agrees with most of the statements in the paper about various aspects of ceramic rolling elements, he is somewhat surprised to find in author's Table 1 that very low load capacity is given to full ceramic bearings, notably silicon nitride. This is contrary to early conclusions by Bhushan and Sibley [28].

Recognizing the author's keen interest to generate simple factors for use by engineers, this discusser wishes to elaborate on several points regarding the possible limitations on the use of Table 1 as listed below:

1. Presently, there is lack of endurance test data for

all ceramic bearings. Such test data may be desired to assess the load capacity of rolling element bearings.

2. Most of the author's tests were performed prior to 1975, which is about the time Scott et al. [9,10] published their experimental results on silicon nitride rolling elements. The author has correctly pointed out in the paper that the fatigue life of ceramic rolling elements depends on the size of voids and surface defects, but with improved manufacturing methods in recent years, much longer fatigue life should be expected. Based on the author's own argument, it is clear that Table 1 is applicable to silicon nitride rolling elements manufactured in the early seventies, rather than the late eighties.
3. The results reported on the five-ball tests by the author contradict the (RCF) roll-disk test rig data for silicon nitride by Baumgartner [12] shown in Figure 2. The latter's test results yield a life about eight times that of the M-50 steel rod at the same maximum stress level. Since the author's test and Baumgartner's test were conducted at about the same time and using very similar material, the reason why they latter test yields significantly greater relative life than the author's test is a question of interest.

Although experiment shows silicon nitride ball (or rod) fails by spalling as with steel rolling elements, there is reason to believe that the spalling mechanism in ceramics is different from that of steel. In general, ceramics is a brittle material weak in tension but strong in compression. Although the process of fracture (or spalling) in ceramic rolling contacts is not well understood, the existence of tensile stress along the edge of the contact area and its effect on Hertzian fracture has been analyzed (Lawn [37], Johnson [38]). Morrison et al. [39] reported that fatigue spalls on ceramic balls in the hybrid bearings are originated at the Hertzian cracks. The load life exponent of the hybrid bearing is about the mean of the theoretical value of 3 for steel ball bearings and the author's value of approximately 5.4 from the five-ball test rig. Valori [40] suggests that surface initiated cracks causing fatigue spalling will not occur below a critical load or Hertz stress level.

In the five-ball test rig, the nominal area of contact is circular with the maximum tensile stress approximately 17% of the maximum contact stress. This drops to about 10% for an elliptical contact of axes ratio equal to 10. This smaller tensile stress will enable a greater load capacity in a ball (or roller) to race contact. Another possible cause for the conflicting data in five-ball and RCF rigs using ceramic specimens is the existence of spin in the five-ball tester, which can generate tensile stress inside the contact and initiate cracks or spalls.

Recent tests conducted in Japan by Komeya and Kotani [41] using three $^{3}/_{8}$ in. steel balls and a silicon nitride disk show at least two times the fatigue life of the same test rig with a steel disk under the same load (400 kgf).

Finally, the author has reported conflicting results between frictional loss in ceramic hybrid bearings tested by the authors and that by Reddecliff and Valori [16]. The author attributed his finding of higher torque to the higher sliding traction in the steel–ceramic contact than in the steel–steel contact, which has been observed in early ball–disk rolling/sliding experiment (Delal et al. [42]). However, for high speed ball bearings, the use of lightweight ceramic balls tends to decrease the inner ring contact angle (or less spin) and shorten contact ellipse (lower spin moment); both of these factors can contribute to lower friction loss than in the case where steel balls are used.

It is of interest to point out a recent observation by Aramaki et al. [43] on testing silicon nitride hybrid ball bearings of two types, namely, 95 mm bore, 40 deg contact angle, 1000 N axial load, lubricated by grease, and 65 mm bore, 17 deg, 75 N load lubricated with an oil and air mixture. This test shows greater (30% to 50%) reduction in power loss in hybrids than with steel balls when the bearing speed is greater than 3000 rpm.

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Discussion

T.E. Tallian

1. The author rightfully stresses the significance of the high stress levels resulting, for a given load, from the high elastic modulus of the ceramics. It is not uncommon to see comparisons of the fatigue life of rolling contacts between steel and ceramics, based on identical stress, even though, for any practical purpose, the comparison must be based on identical load.
   However, the disadvantage in stress arising from the high modulus of ceramics is severe only when both the rings and rolling elements are made of ceramic. Hybrid bearings (ceramic rolling elements only) suffer much less extra stress from the high modulus of the ceramic. Since hybrids are so much easier to use than full ceramics bearings, they are the design of choice for high speed applications, leaving full ceramic designs for high temperature or dry lubricated applications where no other solution serves.
2. While hybrid bearings running in conventional lubricants have undergone sufficient evaluation to be seriously considered where they present a life advantage, the same cannot yet be said about full ceramic bearings in dry lubricant. More than a decade of research has been devoted to this configuration, yet no consistently funded product development is reported by any group which had control over fabrication and application of the component, including the lubricant, and was faced with a continuing major engineering need that only a full ceramic bearing would solve. As a result, only pilot quantities of dry lubricated ceramic bearings have been made or installed. Neither the ceramic material nor its finishing are fully evolved. As the paper states, we do not even have proven solutions for mounting ceramic rings. All this is the case, since a way has so far always been found to design around the need for a truly high temperature (over 800° F) rolling bearing, (or, for that matter, a long-lived bearing, which must run in a cryogenic liquid).

It is instructive to contrast this situation with the development of aircraft gas turbine mainshaft bearings. The evolution of their material, finishing and mounting techniques and lubrication methods to the present state has taken about 20 years, during which such bearings were made and used in large numbers. All improvements were incremental and driven by copiously funded major engineering centers at aircraft engine manufacturers and military users. The motivating force was that turbine engine longevity was largely bearing limited and no design other than rolling bearings worked. If a similar situation were to arise for full ceramic bearings, then we would eventually see them realize their great potential and become a practical machine component.

Author's Response

Erwin V. Zaretsky

The author would like to thank Dr. Y.P. Chiu and Mr. T.E. Tallian for their respective discussions. Both men have made significant contributions to rolling element bearing technology over the years and their discussions further add to the author's paper.

Dr. Chiu states that "the results of the author's Table 1 show a very low (relative) load (dynamic) capacity for full ceramic bearings, notably silicon nitride . . . contrary to early conclusions by Bhushan and Sibley [28]." Bhushan and Sibley did not compare the dynamic load capacities of bearing steel and silicon nitride. What they did in their paper was to compare the experimental lives reported in the literature for silicon nitride rolling element test specimens or hybrid bearings with silicon nitride rolling elements with the predicted lives of steel bearings or test elements using the Lundberg-Palmgren theory [21–23] without life adjustment factors (Bamberger et al.

[30]). If Bhushan and Sibley had put life adjustment factors into their predictive lives for the steel bearing and test elements, they would have found that the predicted lives would have exceeded the experimental lives obtained with the silicon nitride in nearly all the tests reported. Had Bhushan and Sibley calculated the dynamic load capacities for the tests reported, they would have found that in all the tests the dynamic load capacity of the silicon nitride would have been less than that of bearing steel. Bhushan and Sibley did not report on other ceramic materials in their paper.

Dr. Chiu is correct in pointing out that the data reported for the silicon nitride in the author's Table 1 was generated in the 1970s. However, the author could not find in the reported literature any other data except that of Morrison et al. [39]. While the author would expect improvement to have been achieved in the 1980s in the performance of silicon nitride and the other ceramic materials reported in the table, it would be reasonable to expect that these improvements, if they exist, would have been reported in the technical literature.

With regard to the comparison of the Baumgartner data [12] with AISI M-50 (Fig. 2), the author reviewed data reported by Bamberger and Clark [44] which were also generated in the rolling-contact (RC) fatigue tester. The data generated by Baumgartner et al. [12] was typical but on the low side of CVM AISI M-50 bearing steel fatigue data obtained with the RC tester. Likewise, the data for CVM AISI M-50 reported in the author's Figure 1 is consistent with similar data obtained in the NASA five-ball fatigue tester (Zaretsky et al. [45]). This would suggest differences in either manufacturing processes or material quality between the batches of hot pressed silicon nitride tested by Baumgartner et al. [12] and Parker et al. [24]. Both batches of material came from a single supplier.

Dr. Chiu states that "the load life exponent of the hybrid bearing is about the mean of the theoretical value of 3 for steel ball bearings and the author's value of approximately 5.4 (for silicon nitride) from the five-ball test rig." In order to ensure that there is no misunderstanding of the author's data, the stress life exponent n given in Table 1 would be for a full complement ceramic bearing and not for a hybrid bearing. For the silicon nitride material, the value for n of 16.1 was independently determined from Parker et al. [13,24] and Baumgartner et al. [12] in the five-ball rig and the RC rig, respectively. For the hybrid bearing calculations of Table 3, the author assumed ceramic rolling elements and steel races. The author further assumed that the ceramic rolling elements would not fail. Hence, it was assumed that the life of the bearing was solely dependent on the failure of the steel races. Since the stress life exponent n of bearing steel is 9, the load life exponent for a steel bearing would be 3. However, should there be a combination of steel race failures and ceramic rolling element failures, the apparent load life relation of the hybrid bearing would fall between 3 and 5.4. If only the silicon nitride rolling elements failed, then the load life exponent would be around 5.4. This is illustrated by the work of Morrison et al. [39].

Morrison et al. tested four groups of hybrid 45 mm bore angular-contact ball bearings having double-vacuum melted (VIM-VAR) AISI M-50 steel races and silicon nitride balls. There were seven balls in each bearing. The bearings were tested at four thrust loads. These were 4.45, 5.00, 6.45, and 9.56 kN (1000, 1125, 1450, and 2150 lb). These loads produced inner-race maximum Hertz stresses of 1.95×10^9, 2×10^9, 2.17×10^9, and 2.44×10^9 N/m^2 (281, 290, 315, and 354 ksi), respectively. The failure index, or the number of bearings failed out of those tested for each load, were 5 out of 10, 4 out of 20, 11 out of 20, and 5 out of 10, respectively. All of the failures for each thrust load were spalling of a silicon nitride ball. There were no failures of the steel raceways. The experimental lives obtained by Morrison et al. are given in Table 4 together with the theoretical lives calculated by the author for full complement AISI M-50 bearings having the same dimensions. The life and dynamic capacity of the hybrid bearing was less than the theoretical life and capacity of full complement AISI M-50 steel ball bearings.

Morrison et al. [39] reported that the load life exponent for the hybrid bearings was 4.29 with 95% confidence limits of 3.16 and 5.42. Based upon a load life exponent of 4.29, the stress life exponent n for these bearings is approximately 13, which is solely a function of the failure of the silicon nitride balls. From the experimental data, the actual stress life exponent was approximately 14. These values are not significantly less than those obtained in the five-ball and RC rigs previously discussed.

The author has read the paper by Komeya and Kotani [41]. They concluded that "silicon nitride retains a rolling life equivalent to or better than that of a conventional bearing steel." Unfortunately, the information reported in the paper is not sufficient to determine with reasonable certainty whether the steel and ceramic specimens were run under the same load or the same stress.

The work of Aramaki et al. [43] comparing hybrid bearings with full complement steel bearings reinforces the author's conclusion that differences in power losses are a function of the individual bearing design. For the same design the operating contact angle on the inner race for the full complement steel bearing can be higher heat generation due to increased spinning relative to the same bearing using lighter weight silicon nitride balls. However, the geometry of a full

Table 4. Comparison of Experimental Lives Obtained with Hybrid 7209-Size Angular-Contact Ball Bearing Having Silicon Nitride Balls with Theoretical Lives of Full Complement AISI M-50 Angular-Contact Ball Bearings (Contact angle, 27 deg; speed, 9700 rpm, lubricant, MIL-L-23699, oil-in temperature, 311 K [100° F] [39])

Thrust Load, N (lb)	Inner-Race Stress, N/m^2 (ksi)	L_{10} Life, Millions of Inner-Race Revolutions: Experimental Hybrid[a]	Predicted AISI M-50: CVM[b]	Predicted AISI M-50: VIM-VAR[c]	Relative Hybrid Life to AISI M-50: CVM	Relative Hybrid Life to AISI M-50: VIM-VAR	Relative Hybrid Dynamic Capacity to AISI M-50[d]: CVM	Relative Hybrid Dynamic Capacity to AISI M-50[d]: VIM-VAR
44500 (1000)	1.95×10^9 (281)	404	4698	9396	0.09	0.05	0.24	0.19
50000 (1125)	2.00×10^9 (290)	244	3358	6716	0.07	0.04	0.24	0.19
64500 (1450)	2.17×10^9 (315)	82.0	1641	3282	0.05	0.03	0.24	0.19
95600 (2150)	2.44×10^9 (354)	15.1	548	1096	0.03	0.02	0.23	0.18

[a]Silicon nitride ball failures only, no steel race failures.
[b]Life adjustment factors: material and processing 6; lubricant, 2.9 [30].
[c]Life adjustment factors: material and processing, 12 [46]; lubricant, 2.9 [30].
[d]Load life exponent: hybrid bearing, 4.29; steel bearing, 3.

complement steel bearing can be optimized to reduce heat generation. Differences in Coulomb friction properties between steel and silicon nitride should not have any effect on rolling element bearing power losses under reasonable elastohydrodynamic lubrication conditions.

The author agrees with Dr. Chiu's statement that "there is a lack of endurance test data for all (full complement) ceramic bearings." However, manufacturers of these bearings have been claiming without the benefit of data that these bearings "will last 5 to 100 times longer than high performance steel bearings in standard operating environments and infinitely longer in hostile operating environments." These same manufacturers further claim that "ceramic bearings run 100% faster and 30% cooler than steel bearings." Considering that steel bearings have been run to speeds of 3 million *DN* (*DN* equals bearing bore in millimeters multiplied by bearing speed in revolutions/minute) at lives equivalent to those obtained at lower speeds (Bamberger et al. [46]), it is difficult to imagine the basis for the performance claims attributed to the full complement ceramic bearings.

Ceramic and hybrid bearings have found application in severe chemical and industrial environments where conventional steel bearings are adversely affected by the environment. Hybrid bearings have also found limited application in high speed machine tool spindles and unmanned missile applications. However, full complement ceramic rolling element bearings cannot be retrofitted into an existing design meant for a full complement steel bearing without redesign of the application. Further, full complement ceramic rolling element bearings and hybrid bearings, with today's technology for a given envelope size and load, will neither produce longer fatigue lives than an equivalent steel bearing nor necessarily run faster. The problems faced by ceramic bearings are the same as those for other ceramic structures. These are:

1. Fracture toughness
2. Batch-to-batch quality assurance
3. Nondestructive inspection methods
4. Manufacturing technology
5. Design methods and optimization
6. Environmental interaction

As these problems are solved, then, as Mr. Tallian states, "we would eventually see them (full complement ceramic bearings) realize their great potential and become a practical machine element."

ADDITIONAL REFERENCES

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