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ANGULAR-CONTACT BALL BEARING OPTIMIZATION STUDY

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NOTICE

THIS TECHNICAL REPORT HAS BEEN REVIEWED AND IS APPROVED FOR PUBLICATION.

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ONGOING STUDIES HAVE INDICATED THAT ADVANCED ENGINES WILL BE REQUIRED TO OPERATE AT HIGHER TEMPERATURES AND INCREASED ROTOR SPEEDS. THE OBJECT OF THIS PROJECT WAS TO FIND THE OPTIMUM INTERNAL GEOMETRY FOR A HIGH SPEED, HYBRID THRUST BEARING, WHICH PRODUCES MINIMUM HEAT GENERATION. THE DEVELOPMENT OF ADVANCED BEARING DESIGNS WILL SIGNIFICANTLY INCREASE TURBINE ENGINE OPERATIONAL CAPABILITIES.

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1. Introduction

The continuing goal of improving aircraft propulsion system capability by increasing the turbine engine thrust-to- weight ratios has resulted in the need to develop advanced technology for engine mechanical systems. Ongoing studies have indicated that advanced engines will be required to operate at higher component temperatures and increased rotor speeds. In these applications, mechanical and lubrication system components, including bearings, seals, dampers, gears, and lubricants, will be required to perform beyond the current state of the art.

The objective of this project was to find the optimum internal geometry for a high speed, hybrid (silicon nitride / M50 steel) thrust bearing, which produces minimum heat generation. Rolling element bearings used in current production engines are operating at speeds up to 2.4 million DN, where DN is equal to the shaft diameter in millimeters times the shaft RPM, and temperatures around 400° F. However, the advanced engine designs being considered will require the bearings to operate in excess of 2.4 million DN and at temperatures the exceed the bearing material capability (hot hardness) and the liquid lubricant thermal stability. Therefore, the development of alternative bearing designs and lubrication techniques, such as the use of ceramic materials, high temperature composites, solid lubricants and improved bearing internal geometry and cage designs will be required.

2. Problem Description & Objective:

Angular-contact ball bearings, as shown in Figure 1, are used in aircraft turbine engines to support combined radial and thrust loads or very high thrust loads depending on the bearing contact angle magnitude. Figure 2 shows the ball-raceway contact configuration due to axial shift of the inner and outer rings when a thrust load is applied. The contact angle, α, changes with applied load and bearing speed. The bearing geometry, which includes internal radial clearance, ball diameter, ball complement and race curvatures, fully define the bearing response to the operating environment, the internal load distribution and the load/stress/deflection relationships. Consequently, these parameters are principal design variables. Race curvature defines the conformity between the ball and raceway. Figure 3 is an axial section showing this relationship. Raceway radius (r) must always be greater than the ball radius and is usually expressed as a percentage or decimal fraction of ball diameter, thus race curvature is given by $f = r / D \times 100$ (%). Race curvature, f, is normally between 51.5% and 54% for aircraft steel bearings.

Loads acting between the rolling elements and raceways in rolling bearings develop only small areas of contact between the mating members. Consequently, although the elemental loading may only be moderate, stresses induced on the surfaces of the rolling elements and raceways are usually large. It is not uncommon for rolling bearings to operate continuously with normal or Hertz contact stresses exceeding 200,000 psi compression on the rolling surfaces. Contact deformations caused by these contact stresses are generally of a low order of magnitude, for example 0.001 in or less in steel bearings. It has been considered that if a rolling bearing in service is properly lubricated, properly aligned, kept free of abrasives, moisture, corrosive reagents and properly loaded, then all causes of damage are eliminated except one, material fatigue. However, it has been shown that if the maximum Hertz contact stress is kept below 300,000 psi for steel bearings, then infinite life is possible if the above conditions are also satisfied.

Ideally, a ball in an angular-contact bearing operating at a contact angle α, can only have pure rolling motion with respect to one race. At the other race there will be rolling combined with spinning. The magnitude of spin depends on race curvatures, speed, contact angle and load. High spin-to-roll ratios increases bearing heat generation and may initiate local ball/race surface distress and heat damage.

The design of a ball bearing is an iterative process. The goal is to maximize the rolling contact fatigue life while ensuring that the bearing will not experience other distress modes which result in premature bearing failure. This is done by designing the bearing such that pertinent bearing performance parameters are constrained to fall into ranges that previous experience has shown will produce robust bearing designs. One such parameter is bearing frictional heat generation due to ball spinning. Friction of any magnitude represents an energy loss and causes a retardation of motion. Hence, friction due to spinning in rolling bearings is witnessed as a temperature increase and may be measured as a retarding torque or moment which is given for a single bearing contact as follows:

$$Ms := \frac{3 \cdot \mu \cdot Q \cdot a \cdot \varepsilon}{8}$$

where, μ -coefficient of friction (0.01-0.03 for oil lubricated bearings)

Q -normal load (lbs)

a -semimajor axis of deformed contact ellipse caused by Hertz contact stress

ε -complete elliptic integral of second kind

Bearing race curvature has been identified as a design characteristic having a strong influence on a number of design parameters including frictional heat generation. A loose curvature (i.e., ≈ 54%) results in higher Hertz contact stress which lowers rolling contact fatigue life. However, a loose curvature also provides lower heat generation due to the smaller contact ellipse area. Conversely, a tight curvature (i.e., ≈ 51%) lowers Hertz stress, increasing analytical predicted life while increasing heat generation and wear. A successful bearing design requires that these two opposing effects be balanced.

Bearing material is another design consideration which can have a strong influence on bearing performance and life. The use of light weight ceramics, such as silicon nitride, has the potential for increasing rolling contact fatigue life at high bearing speeds. Silicon nitride is 60% lighter than steel and thus, when used as a ball material, can reduce the centrifugal ball loads on the stationary outer ring. However, the higher Young's modulus of silicon nitride tends to negate this benefit through higher Hertz stresses. Successful hybrid bearing designs employing silicon nitride balls and steel rings requires parametric trades which capitalize on the benefits of the lighter balls and minimizes the penalty of the higher Hertz stresses associated with their use. Tighter race curvatures are a key element in this process.

The objective of this project is to find an optimum inner race curvature, f_i , value for a 206 standard sized angular-contact hybrid bearing (silicon nitride balls/steel raceways) ,which minimizes heat generation due to spinning, while meeting a maximum Hertz contact stress level of 300,000 psi.

3. Task Description:

Task (1): The design features of a standard 206 size angular-contact bearing were compiled. The following is a list of the primary design characteristics:

* Inner diameter (bore) 1.1811 inch

* Outer diameter 2.4409 inch

* Width 0.6299 inch

* Radial clearance 0.0021/0.0028 inch

* Pitch diameter 1.818 inch

* Inner ring curvature 52.5 %

* Outer ring curvature 51.5%

* Ball complement 13

* Ball diameter 0.375 inch

* Normal contact angle 25

* Material M50 tool steel

Task (2): The inner ring curvature was selected as the primary design variable for this study. The moment due to spinning was minimized by varying inner ring curvature and ball diameter. Besides the geometric constraints (i.e. bore, diameter, width) the primary inequality constraint was to stay at or below a maximum Hertz contact stress level of 300,000 psi. The following is a short list of bearing operating conditions which remained constant:

* Bearing speed 50,000 RPM (1.5 MDN)

* Bearing thrust load 1000 lb.

* Bearing radial load 0

Task (3): The following objective or primary design function was selected.

$$Ms := \frac{3 \cdot \mu \cdot Q \cdot a \cdot \varepsilon}{8}$$

The function was minimized by a multivariate exhaustive search via optimization of the inner race curvature, f_i and ball diameter, d. A maximum Hertz stress inequality constraint of 300,000 psi was imposed on the design.

Task (4): Development of an optimization algorithm was completed using Mathcad. The algorithm, which identifies the variables, constants, and the equations needed to minimize the objective function, is included in the appendix. An exhaustive search approach of limited array size was chosen so that some generalizations about the optimum design point could be made.

4. Results

Mathcad was used to conduct the multivariate optimization. Mathcad was chosen for several reasons: First, some of the intrinsic functions are powerful. Secondly, this program allows you to write equations out symbolically. This is an important parameter for group activities (i.e., you don't have to be the programmer to understand the logic). Finally, and most importantly the authors had a working knowledge of the system. The bearing optimization code is basically a straightforward flow down program. The user inputs the equality constraints and the fundamental equations are calculated in order to obtain a measure of the inequality constraint and the objective function. If the inequality constraint or hertz stress is exceeded then the respective equality constraints are disregarded. This process is continued for every equality constraint in the multivariate solution space. The solution space was limited to an 11x21 array where ball diameter was varied from 0.375 to 0.475 by 0.01 and the inner race way curvature was varied from 0.505 to 0.525 by 0.002. The array was limited in size because certain intrinsic Mathcad functions couldn't handle range variables, but the size was large enough to make generalizations. In fact after becoming familiar with the affects of changing certain design variables the authors could have honed in on the minimum without completing the array. The array was completed so that an objective party could make decisions based on the data.

There were three intrinsic functions that made the optimization possible. Pspline was used to create a cubic spline polynomial fit to a set of three curves ($F\rho i$ vs a^* , $F\rho i$

vs b* and B vs K). The interp function was used to interpolate values from these curves. The find function was used to iterate on an embedded variable (the contact angle). The find function proved to be the main draw back of using Mathcad because this function will not accept a range variable. Therefore, manual insertion of each variable in the solution array was necessary and proved to be tedious. However, this allowed the operator to check the exceedance criteria of the inequality constraint. If the constraint was exceeded, then the remaining inner race curvatures could be neglected. Due to the fact that hertz stress increases with radial clearance for a fixed geometry, the size of the exhaustive search array, 231 calculations (see Table 1 and Fig 4), was reduced to 143 calculations (NOTE: an exceedance was documented as zero spinning moment for graphing purposes). So, with a little insight the size of the solution array was reduced and for real design purposes this number could be significantly reduced or increased depending on the users' familiarity and confidence.

Table 1. shows the Ms values obtained for the analysis. The data are also shown in Figure 4 as a 3-D representation of the resulting spinning moment values as a function of both inner race way curvature and ball diameter. The Hertz contact stress inequality constraint is clearly shown as the " cliff " which falls off to zero Ms values. As described earlier, these zero values were chosen arbitrarily for graphing purposes.

It is interesting to note that Ms varies parabolically with fi and linearly with ball diameter, D. As expected, Ms decreases with increasing race curvature and decreasing ball diameter. Both result in a smaller contact ellipse size and hence a higher Hertz contact stress. For the fi and D values analyzed, the race curvature value and ball diameter that give the minimum Ms value occurred at fi = 0.51073 and D = 0.275. These values were found to lie on the stress inequality constraint of 300,000 psi. The resulting minimum Ms value obtained was 0.02271 in-lb. This value represents the minimum moment due to spinning for each ball-raceway contact within the bearing. For a comparison, the current M-50/ M-50 bearing has geometry of D = 0.375 and fi = 0.525, which results in a spinning moment of 0.02902 and Hertz stress = 289,753 psi.

Multivariate Angular Contact Ball Bearing Design Exhaustive Search Array

Table 1. Ball Diameter vs Inner Race Curvature vs Spinning Friction x10^{-2}

	0.505	0.507	0.509	0.511	0.513	0.515	0.517	0.519	0.521	0.523
0.275	2.696	2.472	2.352	0	0	0	0	0	0	0
0.285	2.781	2.55	2.428	2.334	0	0	0	0	0	0
0.295	2.866	2.628	2.503	2.407	0	0	0	0	0	0
0.305	2.952	2.706	2.579	2.481	0	0	0	0	0	0
0.315	3.037	2.784	2.654	2.554	2.479	0	0	0	0	0
0.325	3.123	2.863	2.73	2.628	2.551	0	0	0	0	0
0.335	3.209	2.941	2.806	2.702	2.624	0	0	0	0	0
0.345	3.295	3.02	2.881	2.775	2.696	2.637	0	0	0	0
0.355	3.382	3.099	2.957	2.849	2.768	2.708	0	0	0	0
0.365	3.468	3.177	3.033	2.923	2.841	2.779	0	0	0	0
0.375	3.555	3.256	3.109	2.998	2.913	2.85	2.805	0	0	0
0.385	3.642	3.335	3.185	3.072	2.986	2.922	2.875	0	0	0
0.395	3.728	3.414	3.261	3.146	3.058	2.993	2.946	2.913	0	0
0.405	3.815	3.494	3.337	3.22	3.131	3.064	3.016	2.982	0	0
0.415	3.903	3.573	3.413	3.295	3.204	3.136	3.087	3.052	0	0
0.425	3.99	3.652	3.489	3.369	3.277	3.208	3.157	3.122	3.097	0
0.435	4.077	3.732	3.565	3.444	3.35	3.279	3.228	3.192	3.167	0
0.445	4.165	3.812	3.641	3.518	3.423	3.351	3.299	3.262	3.236	0
0.455	4.253	3.891	3.717	3.593	3.496	3.423	3.369	3.332	3.306	3.286
0.465	4.341	3.971	3.793	3.668	3.569	3.495	3.44	3.402	3.3705	3.356
0.475	4.429	4.051	3.869	3.742	3.642	3.567	3.511	3.472	3.445	3.425

* zero spinning friction represents an exceedance of the inequality constraint (max hertz stress = 300,000 psi)

** Neglecting interval error the actual minimum for a fixed diameter occurs at D = 0.275, fi = 0.51073, Ms = 2.271

*** Baseline M50-M50 D = 0.375, fi = 0.525, max hertz stress = 289,753 psi , Ms = 2.902

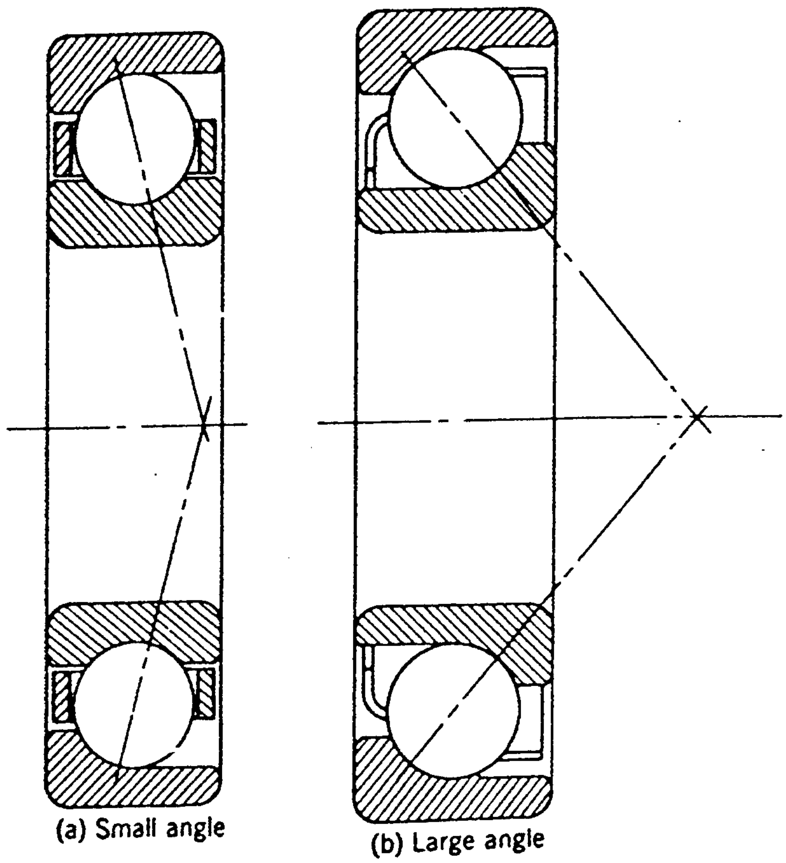

Fig. 1) Angular-contact ball bearings.

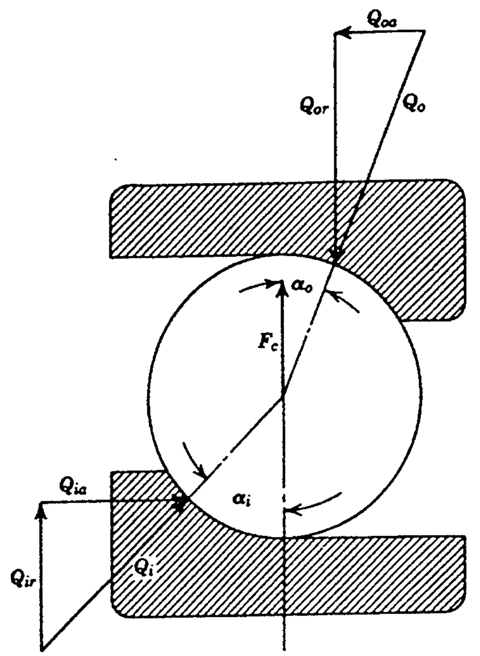

Fig. 2 Ball under thrust load and centrifugal load.

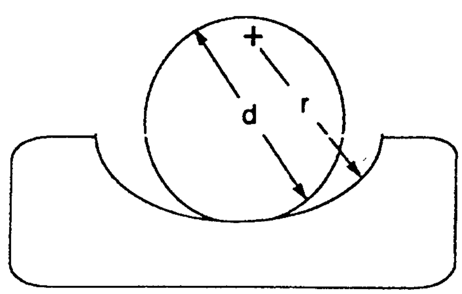

Fig. 3 Race Curvature Defines Conformity Between Ball and Raceway

5. Appendix 1
Mathcad Program

Angular-Contact Ball Bearing Optimization Program

$$Frhoi := \begin{bmatrix} 0.0 \\ 0.1075 \\ 0.3204 \\ 0.4795 \\ 0.5916 \\ 0.6716 \\ 0.7332 \\ 0.7948 \\ 0.83495 \\ 0.87366 \\ 0.90999 \\ 0.93657 \\ 0.95738 \\ 0.97290 \\ 0.983797 \\ 0.990902 \\ 0.995112 \\ 0.9973 \\ 0.9981847 \\ 0.9989156 \\ 0.9994785 \\ 0.9998527 \end{bmatrix} \quad astar := \begin{bmatrix} 1 \\ 1.076 \\ 1.2623 \\ 1.4556 \\ 1.644 \\ 1.8258 \\ 2.011 \\ 2.265 \\ 2.494 \\ 2.8 \\ 3.233 \\ 3.738 \\ 4.395 \\ 5.267 \\ 6.448 \\ 8.062 \\ 10.222 \\ 12.789 \\ 14.839 \\ 17.974 \\ 23.55 \\ 37.38 \end{bmatrix} \quad bstar := \begin{bmatrix} 1 \\ 0.9318 \\ 0.8114 \\ 0.7278 \\ 0.6687 \\ 0.6245 \\ 0.5881 \\ 0.548 \\ 0.5186 \\ 0.4863 \\ 0.4499 \\ 0.4166 \\ 0.383 \\ 0.349 \\ 0.315 \\ 0.2815 \\ 0.2497 \\ 0.2232 \\ 0.2072 \\ 0.18822 \\ 0.16442 \\ 0.1305 \end{bmatrix}$$

vs1 := pspline(Frhoi, astar)   vs2 := pspline(Frhoi, bstar)

$$B1 := \begin{bmatrix} 0.0 \\ 0.02 \\ 0.04 \\ 0.06 \\ 0.08 \\ 0.1 \\ 0.12 \\ 0.14 \\ 0.16 \\ 0.18 \\ 0.2 \end{bmatrix} \quad K1 := \begin{bmatrix} 0 \\ 50000 \\ 110000 \\ 175000 \\ 250000 \\ 330000 \\ 415000 \\ 490000 \\ 575000 \\ 675000 \\ 760000 \end{bmatrix}$$

vs3 := pspline(B1, K1)

Curves fitted by cubic spline polynomial
$F\rho i$ vs a*
$F\rho i$ vs b*
B vs K

$D := 0.275$ ***optimum diameter*** - EQUALITY CONSTRAINT -

$pd := 0.0023$ diametral clearance

$fi := 0.51073$ ***optimum inner race way curvature*** - EQUALITY CONSTRAINT -

$fo := 0.515$ outer race way curvature

$B := fo + fi - 1$ total curvature

$angi := acos\left(1 - \frac{pd}{2 \cdot B \cdot D}\right)$ initial contact angle

$angi = 0.57815$

$Fa := 1000$ thrust load

$k := interp(vs3, B1, K1, B)$ interpolated load deflection factor

$k = 66460.81082$

$angf := 0.35$ initial guess for final contact angle

$z := 13$ number of balls

given

$\frac{Fa}{z \cdot D^2 \cdot k \cdot \sin(angf)} = \left(\frac{\cos(angi)}{\cos(angf)} - 1\right)^{1.5}$ Equation that has the final contact angle embedded on the LHS and the RHS

$cang := find(angf)$ find function iterates based on initial value to solve for the final contact angle

$cang = 0.68733$ final cotact angle

$Q := \frac{Fa}{z \cdot \sin(cang)}$ ball load

$Q = 121.23907$

$dm := 1.818$ bearing pitch diameter

$\gamma := D \cdot \frac{\cos(cang)}{dm}$

$sum\rho i := \frac{1}{D} \cdot \left(4 - \frac{1}{fi} + \frac{2 \cdot \gamma}{1 - \gamma}\right)$ curvature sum

$sum\rho i = 8.38843$

$$Fρi := \frac{\frac{1}{fi} + \frac{2\cdot\gamma}{1-\gamma}}{4 - \frac{1}{fi} + \frac{2\cdot\gamma}{1-\gamma}}$$ cuvature difference

$Fρi = 0.96357$

$E := 5.4\cdot 10^{-8}$ material parameter relating race material to ball material

$ast := interp(vs1, Frhoi, astar, Fρi)$

$ast = 4.68528$ interpolated dimensionless semimajor axis of ellipse

$bst := interp(vs2, Frhoi, bstar, Fρi)$

$bst = 0.37076$ interpolated dimensionless semiminor axis of ellipse

$$a := ast\cdot\left(\frac{3\cdot Q\cdot E}{2\cdot sumρi}\right)^{\frac{1}{3}}$$

$a = 0.04938$ semimajor axis of contact ellipse

$$b := bst\cdot\left(\frac{3\cdot Q\cdot E}{2\cdot sumρi}\right)^{\frac{1}{3}}$$

$b = 0.00391$ semiminor axis of contact ellipse

$$\sigma max := \frac{3\cdot Q}{2\cdot\pi\cdot a\cdot b}$$

$\sigma max = 3\cdot 10^{5}$ *** hertz stress - INEQUALITY CONSTRAINT - satisfied for optimum condition ***

$$\varepsilon := \frac{bst^{2}\cdot\pi\cdot ast}{2}$$

$\varepsilon = 1.01169$ complete elliptic integral

$\mu := .01$ coefficient of friction

$$Ms := \frac{3\cdot\mu\cdot Q\cdot a\cdot\varepsilon}{8}$$ spinning moment - OBJECTIVE FUNCTION -

$Ms = 0.02271$ *** minimum value of objective function ***