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RACEWAY CURVATURE EFFECT ANALYSIS AND OPTIMUM DESIGN ON BALL BEARING LIFE PERFORMANCE

Liangheng Qiu and Jim Parejko
Emerson Power Transmission
909 N. Lafayette Street, Valparaiso, IN 46383

ABSTRACT

The effect of raceway curvature on ball bearing contact stress and fatigue life is analyzed with both Hertzian theory and FEM. A numerical program and a 3-D FEM model are developed to calculate the contact stress and deformation at the bearing ball and raceway. The simulation results of the contact stress and deformation are discussed. The accuracy is evaluated by contrasting finite element results with analytical solutions from Hertzian theory. The effect of bearing race curvature on the contact maximum stress and area between ball and races is discussed. The results show that the race curvature is very sensitive factor to affect the bearing contact stress. The raceway curvature effect on the bearing thrust load capability and bearing running temperature is analyzed and discussed as well. For the validation, A. B. Jones' program has been used to calculate bearing life with different raceway curvatures.

Keywords: Ball Bearing, Raceway Curvature, Stress, Life.

INTRODUCTION

Bearings designed and made by different manufacturers generally have a same dynamic load rating calculated by the ISO standards if the bearings have the same values for their basic envelope dimensions like ball size, ball number, and pitch diameter etc. The standards specify the accepted methods for calculating dynamic load rating and fatigue life of a ball bearing and are primarily concerned with the bearing envelope geometry. The bearings that have a same dynamic load rating should have the same operational life and performance. But, an individual bearing designed and made by different manufacturers has demonstrated significant life difference from each other even under the same application conditions and environments. One reason is that the materials used for bearing and manufacturing processes are different with different manufacturers. The other is that the bearing internal geometry greatly affects the bearing life and performance. Thus the internal geometry of a bearing must be carefully chosen for better bearing life and performance.

The internal contact stresses between ball and rings in a ball bearing are primary factors to determine the bearing fatigue life so that they affect the bearing life and performance. While a ball bearing is running under a load, the contact area between each ball and the rings is relatively small and a moderate load can produce stresses of tens, even hundreds of thousands of pascal contact stress. An important factor among the bearing internal geometry that will influence the contact stresses between ball and rings is raceway curvatures of ball bearing that determine the shape of contact bodies. The contact type between ball and rings can be modeled as Hertzian contact and calculated by the formulae developed by Harris in his book [1]. The raceway curvatures of a ball bearing control the shape of the modeled Hertzian contact. The Hertzian contact stresses and stress distribution are sensitive to the shape of contact bodies. Therefore, carefully selecting raceway curvatures in the bearing design can optimize the contact stresses in the bearing and improve the bearing life dramatically.

Among the bearing manufacturers, the used raceway curvatures of ball bearing are different from one manufacturer to another manufacturer. The raceway curvatures of ball bearing are designed usually from the experience of a designer or manufacturer. There are different concerns for bearing performance by different manufacturers to design the raceway curvatures of ball bearings. Limited literatures like the article [2] have been published to describe the analyses and optimum design for raceway curvatures and bearing performance. In this paper, the analysis about the effect of raceway curvatures of ball bearing is described on the basis of the results of numerical and finite element calculation to bearing contact stresses.

Conventionally, the dynamic capacity and load rating of a ball bearing is handled by Lundberg and Palmgren's formulae [3]. The book of Harris' [1] has described detailed analysis and calculation about the contact stresses between ball and rings in a ball bearing with given the geometry and load. Harris' formulas as well as most other authors' analytical formulae have to use numerical computerized analysis tools to implement the calculation. One commercial computer program

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created by A. B. Jones [4] that can handle contact fatigue life calculation of ball bearings includes the effect of raceway curvatures on the bearing life.

The study in this paper develops a computer program with all numerical methods to calculate the contact stresses between ball and rings in a ball bearing. Based on numerical and finite element analysis results, the change of contact stresses in a ball bearing with the raceway curvature change as well as the effect of raceway curvatures on the bearing life and performance is discussed.

NOMENCLATURE

B: Total Raceway Curvature
D: Ball Diameter
D_p: Bearing Pitch Diameter
E: Modulus of Elasticity
$E(e)$: Complete elliptic integral of the second kind
Fr: Bearing Radial Load
$Jr(\varepsilon)$: Load Distribution Integral
$K(e)$: Complete elliptic integral of the first kind
P_d: Bearing Diametric Clearance
P_e: Bearing Free Endplay
$Qmax$: Maximum Ball Load
Z: Bearing Ball Number
a, b: Half elliptical axels
d_i: Inner Ring Raceway Diameter
d_o: Outer Ring Raceway Diameter
f_i: Inner Ring Raceway Curvature
f_o: Outer Ring Raceway Curvature
n: Load Deflection Exponent
r_i: Inner Ring Raceway Radius
r_o: Outer Ring Raceway Radius
p, p_0: Contact and Maximum Contact Pressure
α^0: Free Contact Angle
δ_r: Deformation
ψ_1: Azimuth angle

NUMERICAL ANALYSIS

Geometry Parameters of Ball Bearing

The analysis in this paper uses a groove ball bearing with a spherical OD outer race as the study target. This type of ball bearing is widely used in the mounted bearing industry. Except for the spherical OD outer race, all other geometry shape factors are typically the same as the cylindrical OD outer race bearing. The geometry parameters of the bearing are shown in Fig. 1.

The ball bearing raceway curvature is the ratio of the raceway radius to the ball diameter. In the following equations, race curvatures as well as several other factors are defined; this can be calculated with the shape parameters of a ball bearing.

Bearing Pitch Diameter:

$$D_p = \frac{d_i + d_o}{2} \tag{1}$$

Raceway Curvatures:

$$f_i = \frac{r_i}{D} \text{ and } f_o = \frac{r_o}{D} \tag{2}$$

Total Curvature:

$$B = f_i + f_o - 1 \tag{3}$$

Bearing Diametric Clearance:

$$P_d = d_o - d_i - 2D \tag{4}$$

Free Contact Angle:

$$\alpha^0 = \cos^{-1}(1 - \frac{P_d}{2BD}) \tag{5}$$

Bearing Free Endplay:

$$P_e = 2BD \cdot \sin(\alpha^0) \tag{6}$$

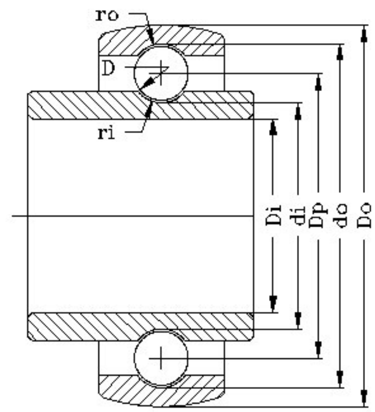

Fig. 1: Bearing Cross-Section Sketch

Contact Stress Analysis

In a ball bearing running under a load, only a small contact area between each ball and ring raceway is developed to transmit the load from one bearing ring to the other through the balls. Consequently, although the elemental loading may only be moderate, the contact stresses induced on the surfaces of ball and raceway are usually large. These contact stresses on the ball and raceway surfaces have a great impact on the bearing fatigue life and create destruction on the rolling contact surfaces between the ball and raceway in a ball bearing [1]. The raceway curvature generally is a significant factor in the ball bearing design. Carefully selecting bearing raceway curvatures so that the bearing has better contact stress distribution and has less contact stress is very important to improve the bearing life and performance.

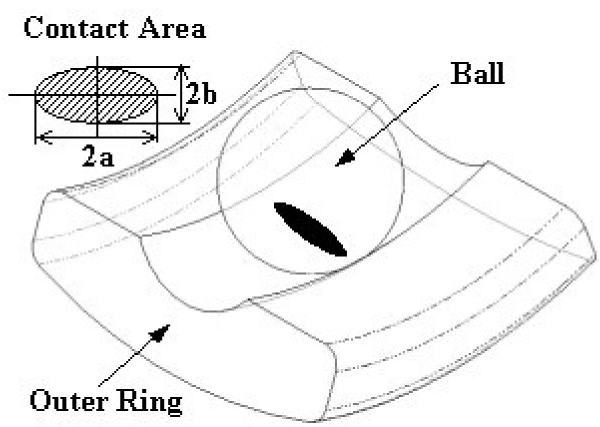

Fig. 2: Contact Area between Ball and Ring

As described in Harris' book [1], the contact of ball and ring in a bearing can be modeled as a Hertzian elastic contact. The contact area (shown in Fig. 2), which is an elliptical shape,

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is determined by the applied contact load and two contact bodies shape - raceway curvatures. In the results and discussion section, it will show that the contact area and contact stresses are sensitively changed with raceway curvatures. Harris, in his book, has described detailed analysis and developed formulae to calculate the contact stresses between ball and rings in a ball bearing as well as the ball load distribution and contact deformation. The book includes tabular and figure data for approximate calculations. However, for the study of the raceway curvature effect on the contact stresses, an accurate numerical tool is necessary to calculate the contact stresses in a ball bearing.

In this paper study, a computer program has been developed to calculate the contact stresses and deformations between ball and rings in a ball bearing by pure numerical methods. The basic equations and implementation used for the program is as follow.

1. Ball Maximum Solution

The following formulae derived by Harris [1] are used to solve the ball maximum load in a ball bearing. The formulae establish the relationship between the bearing applied radial load and bearing ring deformation. With given bearing geometry and load, the deformation δ_r can be solved by numerical methods.

$$F_r = ZK_n\left(\delta_r - \frac{1}{2}P_d\right)^n J_r(\varepsilon) \tag{7}$$

$$J_r(\varepsilon) = \frac{1}{2\pi}\int_{-\psi 1}^{+\psi 1}\left[1-\frac{1}{2\varepsilon}(1-\cos\psi)\right]^n \cos\psi d\psi \tag{8}$$

$$\varepsilon = \frac{1}{2}\left(1 - \frac{P_d}{2\delta_r}\right) \tag{9}$$

Where

Azimuth angle: $\psi_1 = \cos^{-1}\left(\frac{P_d}{2\delta_r}\right)$

Load-deflection factors:

$$K_n = \left[\frac{1}{(1/K_i)^{1/n} + (1/K_o)^{1/n}}\right]^n, \quad n=2/3$$

$$K_i = \frac{2^{n+1}/3 * \sum \rho_i^{\,1-n}\left(\delta^*\right)^{-n}}{\left[\frac{\left(1-\xi_b^2\right)}{E_b} + \frac{\left(1-\xi_i^2\right)}{E_i}\right]}$$

$$K_o = \frac{2^{n+1}/3 * \sum \rho_o^{\,1-n}\left(\delta^*\right)^{-n}}{\left[\frac{\left(1-\xi_b^2\right)}{E_b} + \frac{\left(1-\xi_o^2\right)}{E_o}\right]}$$

$\sum \rho_i$ and $\sum \rho_o$ is determined by raceway curvatures:

$$\sum \rho_i = \frac{1}{D}\left(4 - \frac{1}{f_i} + \frac{2\gamma}{1-\gamma}\right)$$

$$\sum \rho_o = \frac{1}{D}\left(4 - \frac{1}{f_o} - \frac{2\gamma}{1+\gamma}\right)$$

$$\gamma = \frac{D\cos(\alpha)}{D_p}$$

δ^* is dimensionless related to Hertzian contact shape.

With the given F_r and bearing geometry parameters, the equations (7) to (9) can be solved to obtain the deformation δ_r. The load distribution integral $J_r(\varepsilon)$ is calculated by numerical integral method.

When the deformation δ_r is solved, the maximum ball load Q_{max} is calculated by the equation:

$$Q_{\max} = K_n\left(\delta_r - \frac{1}{2}P_d\right)^n \tag{10}$$

2. Contact Stress Solution

Hertzian Contact Parameters for ball and ring contacts (Fig. 3) is:

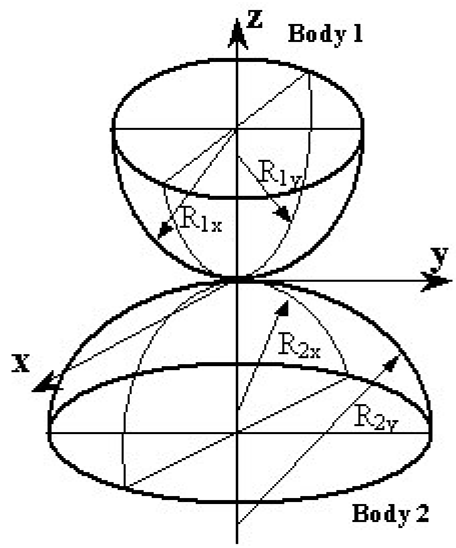

Fig. 3: Geometry of Elliptical Contact

$$R_x = \left(\frac{1}{R_{1x}} + \frac{1}{R_{2x}}\right)^{-1};\ R_y = \left(\frac{1}{R_{1y}} + \frac{1}{R_{2y}}\right)^{-1} \tag{11}$$

$$E = \left(\frac{1-\upsilon_1^2}{E_1} + \frac{1-\upsilon_2^2}{E_2}\right)^{-1} \tag{12}$$

For ball and inner ring contact, there are:

$$r_{1x} = r_{y1} = \frac{1}{2}D;\ r_{x2} = \frac{1}{2}(Dp - D);\ r_{y2} = -f_i D \tag{13}$$

For ball and outer ring contact, there are:

$$r_{x1} = r_{y1} = \frac{1}{2}D;\ r_{x2} = -\frac{1}{2}(Dp + D);\ r_{y2} = -f_o D \tag{14}$$

Hertzian Contact Formulae

$$p = p_0\left\{1 - (x/a)^2 - (y/b)^2\right\}^{1/2} \tag{15}$$

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$$\frac{R_x}{R_y} = \frac{(a/b)^2 E(e) - K(e)}{K(e) - E(e)} \tag{16}$$

$$(ab)^{3/2} = \left(\frac{3PR_e}{4E}\right)\frac{4}{\pi e^2}(a/b)^{-3/2} \cdot \left[\{(a/b)^2 E(e) - K(e)\}\{K(e) - E(e)\}\right]^{1/2} \tag{17}$$

Where $e = \left(1 - b^2 / a^2\right)^{1/2}$, $b < a$

With the equations (15) to (17), half elliptical axels a and b can be solved by numerical iteration for the given raceway curvatures and applied load.

Maximum Hertzian contact stress and deformation are calculated by the equations:

$$p_0 = \frac{3P}{2\pi ab}; \quad \delta = \frac{3P}{2\pi abE} bK(e) \tag{18}$$

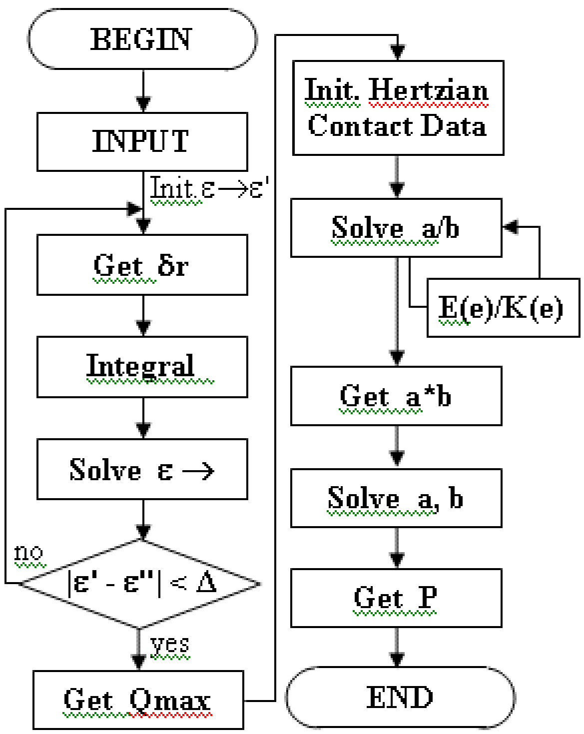

Fig. 4: Program Flow Chart

Figure 4 is a flow chart of the program showing the implementation for the bearing contact stress calculation. By inputting bearing geometry parameters and application parameters, the program will output the bearing ball maximum load, contact elliptical axis length, contact stress, and deformation.

Finite Element Analysis

To more accurately obtain the contact stresses in a ball bearing, a finite element method is developed for the analysis in the study. The method provides the capability to obtain the bearing ball load distribution and the bearing ball maximum load under a given applied load and calculate the contact stress distribution and stresses between ball and ring at the area having the maximum ball load. The method provides a tool to determine the effects of raceway curvatures on the contact stresses of the bearings more accurately for the analysis. It is a useful tool to assist optimized analysis in the bearing design. The finite element method uses two steps to calculate the contact stresses in a bearing: first calculating bearing ball load distribution and obtaining maximum ball load; second calculating the contact stresses at the contact area having the maximum ball load.

1. Bearing Ball Load Distribution

A finite element model to calculate the bearing ball load distribution has been developed by the author in the reference [5]. The current paper uses the finite element model provided by the author in the reference [5] and focuses on the contact stress model development. This finite element model is carried out within ANSYS/Mechanical. The model uses a structural solid element to mesh the rings of bearing and uses a nonlinear spring element to simulate a ball contacting the inner and outer races in a bearing. Such the model can simulate the deflection between the ball and rings and quickly obtains the ball load distribution in a bearing. It avoids a large time-consuming iterative procedure when using a contact element to simulate the contact between the ball and rings.

Table 1 shows the comparison results of maximum ball load calculated respectively by the numerical program described in the previous section, the finite element method, and A.B. Jones' Program. All dimensions of bearings are same except for diametric clearances. The diametric clearances are 0.0051, 0.0102, 0.0152, 0.0203 mm respectively for Case #1, Case #2, Case #3, and Case #4.

Table 1: Bearing Ball Max. Loads (units: N)

Case #	Program	FEA	A. B. Jones
1	2639.7	2663.5	2659.6
2	2685.1	2721.1	2721.4
3	2730.1	2776.9	2770.6
4	2773.2	2824.9	2815.9

2. Contact Stresses Distribution

In order to see in detail more accurate contact stress distribution in a bearing, a finite element model is developed to calculate the contact stresses between ball and rings where the ball has maximum ball load in the bearing. The model is developed with ANSYS/Mechanical. The model uses partial of inner and outer rings and the ball having maximum ball load in the bearing. Such the small size model of a bearing can have fine meshes to obtain more accurate contact stresses between ball and rings under the maximum ball load. The calculated contact stresses include the maximum contact stress in a bearing since the model has the ball having maximum load.

Figure 5 shows the finite element model with mesh carried out in ANSYS/Mechanical. ANSYS CONTACT174 and TARGET170 elements [6] are used to simulate the contact between ball and rings. ANSYS SOLID45 element is used for 3-D structure. Figure 6 shows FEA model applied load and constraints. The maximum ball load calculated by the above model with nonlinear spring elements is used as given input for the model.

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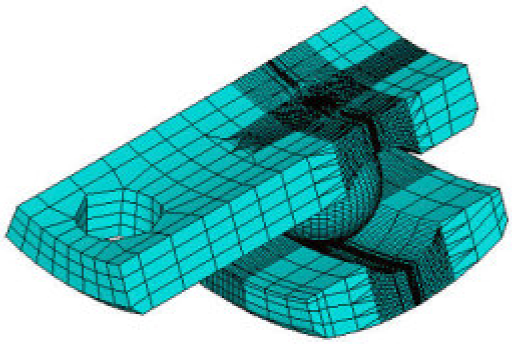

Fig. 5: FEA Contact Model and Mesh

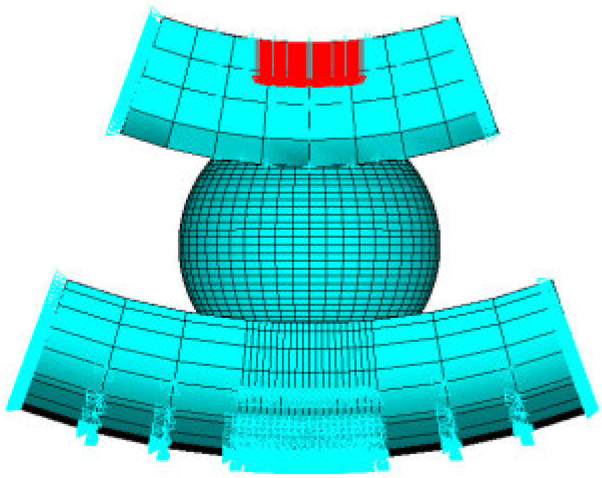

Fig. 6: FEA Load and Constraints

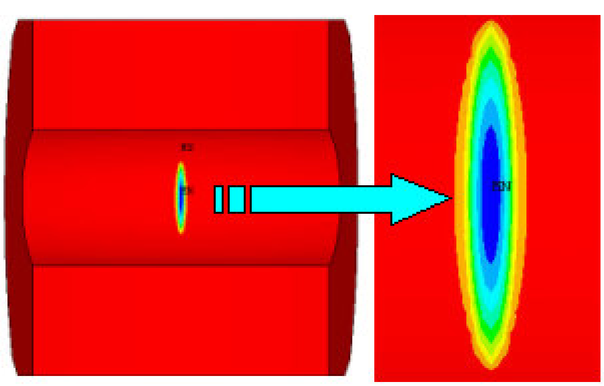

Fig. 7: FEA Contact Stress Contour

Figure 7 is one contact stress contour from FEA results. The contour shows an elliptical contact area between ball and outer ring. The maximum contact stress is retrieved from the nodes attached to the contact elements. Figure 8 shows how to get the lengths of contact elliptical axels.

Table 2 shows the comparison results of maximum contact stresses by the numerical program described in the previous section, the finite element method, and A.B. Jones' Program. Table 3 shows the comparison results of the axial lengths of elliptical contact area. All dimensions of bearings are same except for raceway curvatures. The raceway curvatures (f_i/f_o) are 0.51/0.51, 0.515/0.51, 0.51/0.515, 0.515/0.515 respectively for Case #1, Case #2, Case #3, and Case #4.

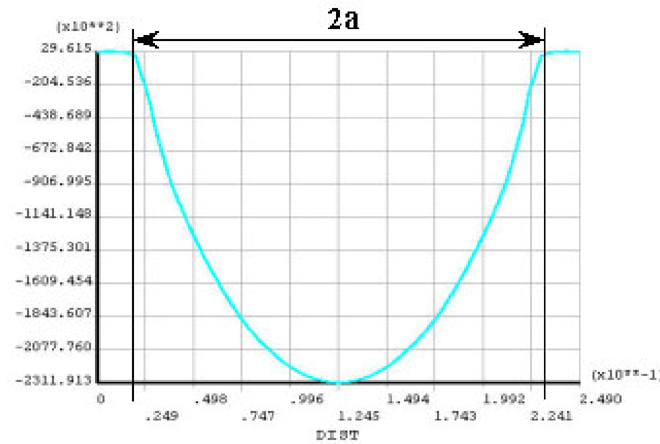

Fig. 8: FEA Surface Stress Distribution

Table 2: Bearing Maximum Contact Stresses (units: GPa)

Case #	Program		FEA		A. B. Jones	
	Inner	Outer	Inner	Outer	Inner	Outer
1	2.361	1.944	2.605	1.949	2.377	1.957
2	2.561	1.959	2.788	1.949	2.576	1.957
3	2.361	2.097	2.605	2.109	2.377	2.126
4	2.561	2.113	2.788	2.109	2.575	2.125

Table 3-(a): Bearing Contact Elliptical Axels (units: mm)

Case #	Program	FEA	A. B. Jones
	Inner (a/b)	Inner (a/b)	Inner (a/b)
1	2.778/0.198	2.352/0.241	2.735/0.195
2	2.324/0.210	2.123/0.256	2.336/0.210
3	2.727/0.193	2.357/0.236	2.735/0.195
4	2.324/0.210	2.123/0.256	2.334/0.210

Table 3-(b): Bearing Contact Elliptical Axels (units: mm)

Case #	Program	FEA	A. B. Jones
	Outer (a/b)	Outer (a/b)	Outer (a/b)
1	2.702/0.246	2.357/0.314	2.661/0.243
2	2.641/0.241	2.357/0.314	2.661/0.243
3	2.263/0.264	2.169/0.337	2.265/0.264
4	2.252/0.261	2.169/0.337	2.265/0.264

RESULTS AND DISCUSSION

All results in the following discussion are calculated with the bearing geometry data and material properties given in Table 4. The raceway curvatures are changed when the different analyses and comparisons require. A radial load 5400 N and speed 1000 rpm are used when a load and speed are applied for the calculations.

Table 4: Bearing Geometry and Material Properties (mm)

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Bearing Geometry Parameters	Material Properties
Ball Diameter $D = 11$ Inner Ring Bore Dia. $D_i = 37$ Outer Ring Outer Dia. $D_o = 72$ Inner Ring Raceway Dia. $d_i = 43$ Outer Ring Raceway Dia.: $d_o = 65$ Bearing Dia. Clearance $P_d = 0.005$ Inner Ring Width = 43 Outer Ring Width = 24 Inner Ring Land Dia. = 47 Outer Ring Land Dia. = 60	Modulus of Elasticity E = 2.0E+11 Pa Poisson' Ratio ν = 0.3

Effect of Raceway Curvatures on Contact Stresses

The raceway curvatures determine the contact shape of the ball and rings and greatly influence the contact stresses and contact stress distribution between the ball and rings in a bearing. The maximum contact stress between the ball and rings changes greatly even though there is a small change of the raceway curvatures. Figure 9 shows the change of the maximum contact stress between the ball and inner ring, calculated by the program developed in this paper, when the raceway curvature in the inner ring varies and the raceway curvature in the outer ring is fixed. The results shows that only 0.005 raceway curvature change can bring up a 7~8% maximum contact stress difference. In term of the S-N diagram of material fatigue strength [7], the fatigue life of the contact ball and rings will be greatly reduced due to the contact stress increase.

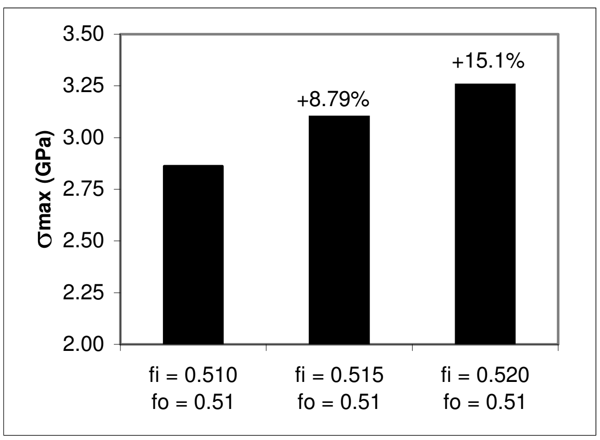


Fig. 9: Effect of Curvature on Maximum Contact Stress

Figure 10 shows the inner ring and outer ring maximum contact stresses with different curvature combinations for inner and outer rings, calculated by the finite element analysis. The raceway curvatures for each case are: Case #1: $f_i = 0.51$ and $f_o = 0.51$; Case #2: $f_i = 0.515$ and $f_o = 0.51$; Case #3: $f_i = 0.51$ and $f_o = 0.515$; Case #4: $f_i = 0.515$ and $f_o = 0.515$. The results confirm that the maximum contact stress is changed significantly with the race curvature change. The results also show that in a bearing the values of maximum contact stress in the inner ring and outer ring are big different. It indicates that carefully selecting raceway curvatures for inner and outer rings to balance the inner and outer ring maximum contact stresses can reduce the maximum contact stress for the bearing and improve the bearing life.

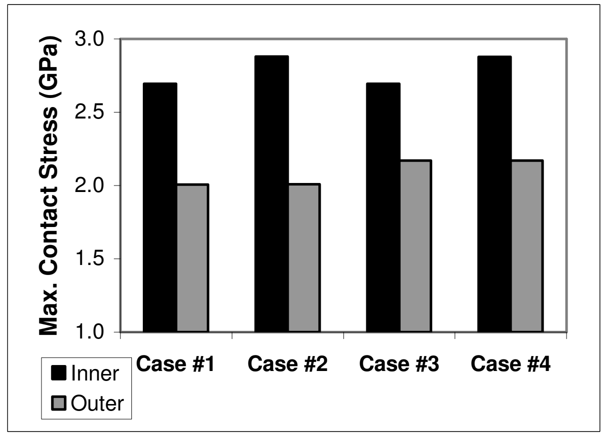
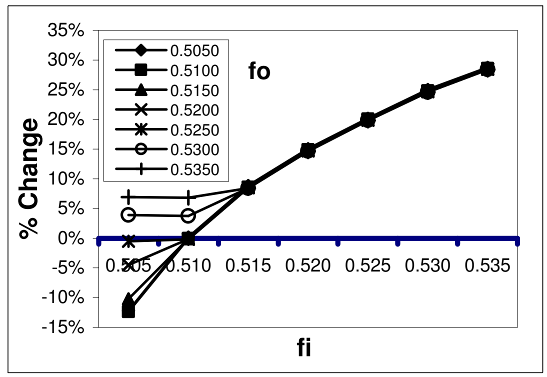
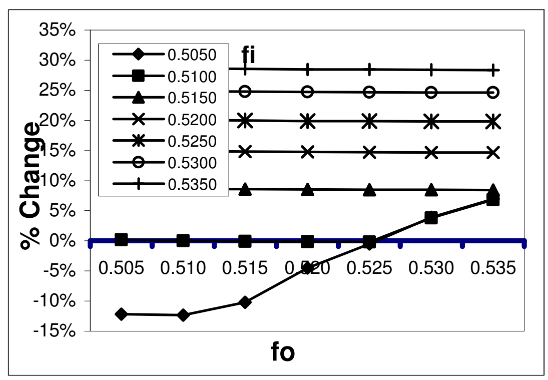


Fig. 10: Maximum Contact Stresses vs. Curvatures



(a)



(b)

Fig. 11: Bearing Max. Contact Stress vs. Curvatures

Figure 11 shows the bearing maximum contact stress in a bearing with different inner and outer raceway curvature selections. The bearing maximum contact stress is defined as the largest maximum contact stress of all components in a bearing, that is, it will take the largest contact stress value of the balls, inner ring, and outer. In Fig. 11 (a), one derives that the bearing maximum contact stress increases with the inner raceway curvature increase when f_i is greater than 0.51 no matter whether f_o changes or it's not. Figure 11 (b) gives the same conclusion derived from Fig. 11 (b) and shows that when $f_i < 0.51$ and $f_o < 0.525$ the bearing maximum contact stress is less than one at $f_i = 0.51$ and $f_o = 0.51$.

Bearing Life vs. Race Curvatures

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The results shown in Fig. 12 and Fig. 13 are calculated with A. B. Jones' Program [4]. The program includes the raceway curvatures of a bearing as input and its output shows the effect of raceway curvatures on bearing life. Figure 12 shows the maximum mean Hertzian contact stress between the ball and rings. The results conclude that the raceway curvature selection of $f_i = 0.51/f_o = 0.51$ or $f_i = 0.51/\ f_o = 0.515$ has less Hertzian contact stresses than ones of the selection of $= 0.515/f_o = 0.51$ or $f_i = 0.515/\ f_o = 0.515$. The mean Hertzian contact stresses increase 8% when f_i changes 0.005 from 0.51 to 0.515, which is matching to the conclusion from Fig. 9.

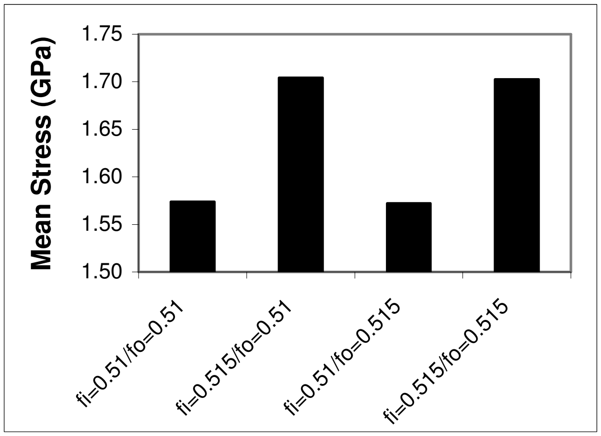

Fig. 12: Hertzian Contact Stresses Change vs. Curvatures

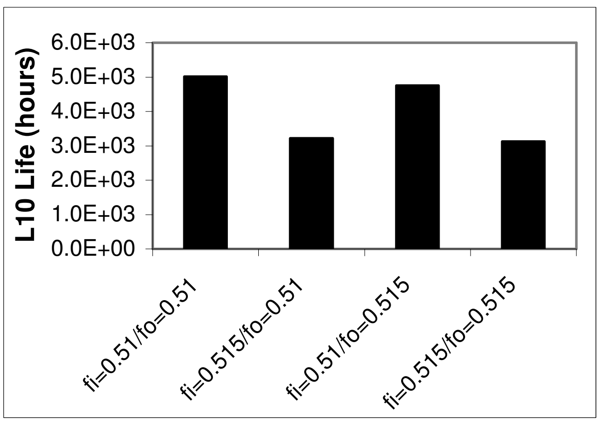

Fig. 13: L10 Life Changes vs. Curvatures

Figure 13 shows the calculated L10 life values of the bearings with the same inner and outer raceway curvatures combinations. The L10 life for the bearing with $f_i = 0.515/\ f_o = 0.51$ is 36% less than the L10 life for the bearing with $f_i = 0.51/\ f_o = 0.51$ only due to 0.005 change of f_i. It indicates again the raceway curvatures in a bearing are important factor in the bearing design and must be carefully selected.

Curvature Selection in Bearing Design

In the bearing design, if only considering fatigue life from only the design point of view, it should select a combination of inner and outer raceway curvatures which will have a smallest bearing maximum contact stress. Figure 14 shows the maximum contact stresses of inner ring, outer ring, and bearing respectively. When $f_o < 0.525$, the bearing maximum contact stress is controlled by outer ring, and when $f_o > 0.525$ the bearing maximum contact stress is controlled by inner ring. The smaller the raceway curvature of control ring is, the less the maximum contact stress of bearing is.

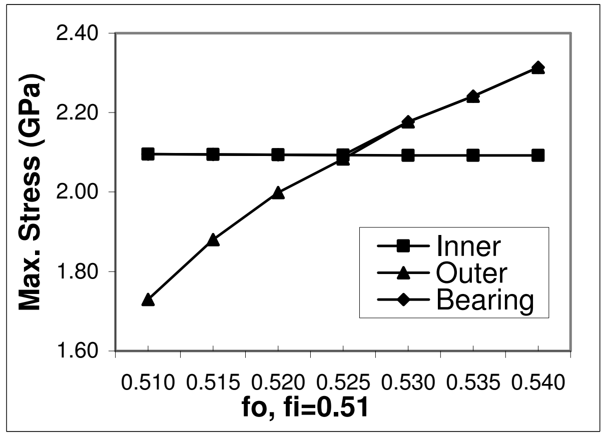

Fig. 14: Max. Contact Stresses vs. Curvatures

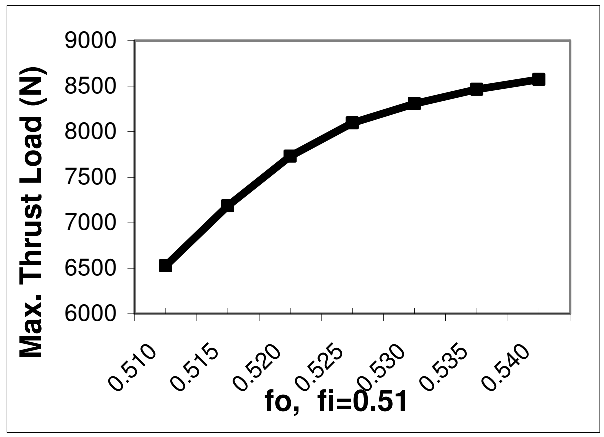

Fig. 15: Thrust Load Ratings vs. Curvatures

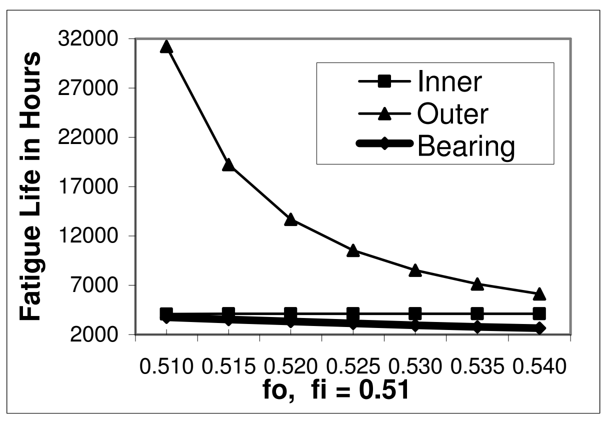

Fig. 16: Fatigue Life vs. Race Curvatures

However, when selecting bearing raceway curvatures in the design, the effect of raceway curvatures on bearing other features also must be considered. For instance, the bearing running temperature and thrust load rating are affected by raceway curvatures. With a small raceway curvature, the frictional heat in a bearing generated in the race contact area may be not as easily transferred so that the bearing would have high running temperature. One of other facts is that the bearing thrust load capability is greatly affected by raceway curvatures.

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Figure 15 shows the bearing maximum thrust load rating changes with different raceway curvature selections. In general, the bearing maximum thrust load rating will increase when the raceway curvatures increase.

Figure 16 shows the fatigue lives for inner ring, outer ring, and bearing respectively with different raceway curvature combinations calculated by A. B. Jones' program. The inner ring and outer ring fatigue lives are quite a big difference. With the bearing parameters given for the calculations, the bearing fatigue life is controlled by the inner ring which has shorter fatigue life than the outer ring. The outer ring raceway curvature change has a minor effect on the bearing fatigue life. It tells us that the outer ring raceway curvature can be a little larger than one of the inner ring for better bearing performance like to better heat transfer and thrust load capacity.

CONCLUSION

With the analysis of the contact stresses in a ball bearing, it demonstrates that the bearing raceway curvatures greatly affect the bearing internal contact stresses so to affect the bearing life and performance. Raceway curvature in a ball bearing is an important factor that should be considered and carefully selected in the bearing design.

The inner ring and outer ring generally do not have the same extent effect on the bearing fatigue life simultaneously. The bearing fatigue life is controlled by either inner ring or outer ring but not by both inner and outer rings. Inner raceway curvature and outer raceway curvature can be designed by optimization to minimize the bearing maximum contact stress between ball and rings at the time to outperform in other bearing features. The raceway curvature is a significant factor for a bearing's optimum design on the bearing life performance.

Besides the impact on bearing contact stresses, raceway curvatures greatly affect the bearing thrust load capacity. When using a small raceway curvature to reduce the contact stress between ball and ring, one should not ignore the influence of a small raceway curvature reducing the bearing thrust load capacity. The raceway curvature selection should be considered comprehensively for best bearing performance.

In this paper, the effect of raceway curvatures in a bearing is studied in detail only on the contact stresses between the ball and rings. For further optimum design for the bearing design, it should consider internal clearance, lubrication condition as well as raceway curvature together for best bearing performance.

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