

Analysis of Angular Contact Ball Bearings Design

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Abstract: *This article is all about the design of angular contact ball bearing which is vital components in most of the machines or application with moving or rotating parts. In the bearing industry, the customers are always looking for fulfilment of all requirements from a single source to make the supplier more effective in supplying the products in time with consistent quality. In-house speedy development of needle roller bearings was one of the steps to fulfill the customer requirements. The design methodology was developed for single row and double row angular contact ball bearing called 1st generation wheel bearing. The mathematical relations were formulated for inner or outer races of angular contact ball bearing. This design methodology was incorporated into a computer program. I-DEAS master series design software was used for verification of the design methodology. With the manufacturer of angular contact ball bearing, it is possible to cater one of the most important requirements of the customer.*

Keywords: Angular contact ball bearing, Design methodology, In-house development, I-DEAS master series, Customer requirement.

1. Introduction

Rolling bearings are among the most important element in machine due to their precision, they are very sensitive elements. The designer must check all boundary condition most meticulously and select the proper bearing type. Many external influences must be taken into account before the optimum bearing can be chosen for a variety of possibilities. Currently, the bearings are designed by bearing manufacturer starting with application details as design inputs. The functional characteristics of bearing are defined by the designer to meet the requirements of the application the bearing designer must consider all the techno – economical aspects while establishing the system solution to the bearing user. An intense and active communication between the bearing manufacturer and the user is, therefore needed to optimally meet the requirement of technical function.

It has been studied that all angular-contact ball bearings have similar features regarding geometry, mechanism, and structure [1]. The stiffness of angular-contact ball bearings has a significant influence on the dynamics of a rotating shaft and the precision of the machine system [2], the rolling contact fatigue (RCF) and life of rolling elements can be determined by full-scale bearing endurance tests. These tests that are conducted in bearing life test rigs are expensive and time consuming [3]. Bearing designers would like to understand the impact of four variables namely Ball material density, Subsurface residual stress, Gradient in yield strength with depth Raceway surface hardness/yield strength that are thought to affect spall propagation[4].

Lin [5] analyzes the displacement and the coefficient of Stiffness for radial ball bearings with angular contact from the production of SKF. For the analysis John Harris's method based on the theory of Hertz-contact has been used. In order

to compare the results obtained by the mentioned method and Palmgren's empirical relations the finite element method has been used. Mullick [6] researches radial stiffness of radial bearings and ball bearings with angular contact using John Harris's method and the finite element method. For solving systems of nonlinear equations, using Newton Raphson's method, while in contact analysis uses finite elements method. The influence of sliding and gyroscopic moment is neglected. The results show that the relative displacement and stiffness rings of bearings depend on the combined loads and centrifugal force. Antoine et al. [7], [8] propose two new approximate, methods for determination the angle of contact on the outer and inner bearing depending on the preloaded and speed for special cases of elastic preloaded. Methods based on the theory of Hertz-contact. They start from assumptions that the force of preload constant and does not affect on the speed and change the angle of contact. In solving the system of equations, it is considered that a certain speed, for a preload comes only to the cancellation of the axial clearance, and that there is no axial deformation.

Sun M. K. at al. [9] investigated the relationship between contact deformation, clearance and change of stiffness bearing. The results obtained by analytical and experimental method suggest determining the elastic deformation of bearing elements must be taken into account axial and radial clearances in the bearing. Wei L. at al. [10] investigated the influence of preload, centrifugal force and gyroscopic moment on the bearing stiffness. They concluded that if the value of contact angle of ball and raceway exceed 8.9 °, the value of the radial bearing stiffness decreases with increasing speed. Experiments have shown that by contact angle 40 ° and revs of 15,000 rpm radial stiffness decreases more than 20%. They also noted that an increase in temperature affects the increase in preload of bearing and increase the natural frequency of oscillation.

In this paper an analysis is carried out for the development of single and double row angular contact ball bearing for the wheel hub, application to cater the entire market requirement in the automobile sector and increase the product mix.

2. Types of Angular Contact Ball Bearings

The angular contact ball bearing is able to carry combined radial and axial loads or pure axial load. In this type of bearings the loads are transmitted from one raceway to another at a certain contact angle. The ISO has given the nomenclature of various dimensions of single and double row angular contact ball bearing as per IS: 2399-1988/ISO5593-1984, which is used by bearing manufactures worldwide. The various terms are used for single and double row angular contact ball bearing by bearing manufactures as shown in Fig.1

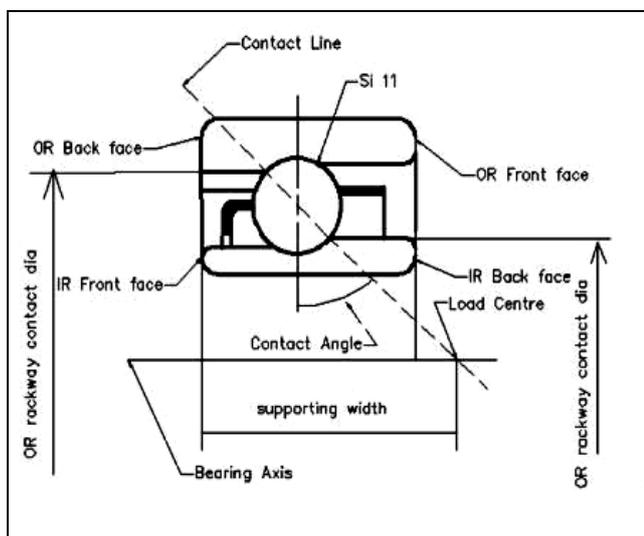


Figure 1: Sketch showing different terms used in Angular Contact Ball Bearing

2.1 Single Row Angular Contact Ball Bearings

Single row angular contact ball bearing support thrust load in one direction only, consequently they have to be mounted in pairs opposed to each other. The nominal contact angles of the bearing are 15, 25, 35, and 40 degree. The greater contact angles are used for application of heavy axial loads. In the case of high accuracy or at high speed it is necessary to select a bearing with a contact angle of 15 degrees. However the contact angle selection depends on the direction of applied load.

2.2 Double Row Angular Contact Ball Bearings

Double row angular contact ball bearings function similar to single row angular contact ball bearings mounted back to back. They can accommodate heavy radial and axial load in both directions. Double row angular contact ball bearings are various types like standard bearing, filing slot bearing, split inner or outer race bearing. Filing slot and split inner race is provided in design to accommodate more number of balls to enhance the load carrying capacity of bearing. The maximum limiting speeds of such bearings are reduced. In

this case of double row angular contact ball bearings with filing slot, the bearing should be mounted in such a way that larger thrust loads are carried by the raceway without filing slot.

3. Radial and Axial Clearance

The amount of alternating radial movement of free bearing rings are defined as radial clearance and the amount of alternating axial movement of free bearing rings are defined as axial clearance as shown in Fig 2. The term internal clearance refers to a state where no force is applied to the bearings and the rolling element is in unload state.

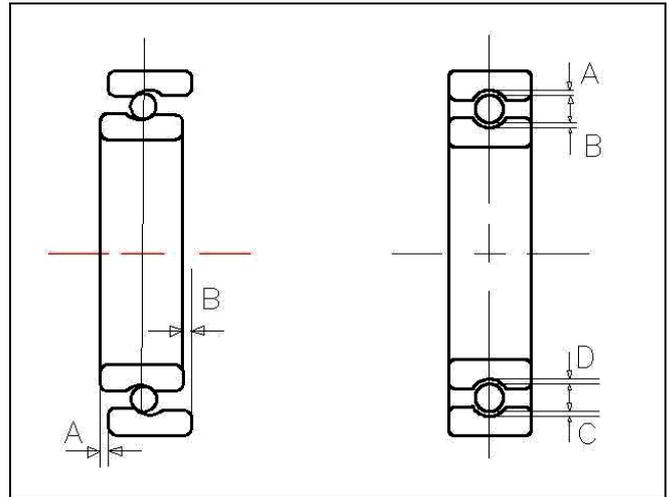


Figure 2: Axial Clearance (A+B) and Radial Clearance (A+B+C+D)

Since a radial ball bearing is generally designed to have a diametrical clearance in no-load state, the bearing also experiences an axial play. Removal of the axial freedom causes the ball raceway contact to assume an oblique angle with the radial plane, hence a contact angle different from zero will occur. For Angular Contact Ball Bearings (ACBB) especially designed to operate under thrust load, the clearance built into the unloaded bearing along with the raceway groove curvature, determines the bearing free contact angle. Fig. 3 shows the geometry of radial ball bearings with removed axial play.

3.1 Definition of Axial Clearance as Per ISO:

The arithmetical mean of axial distances through which one of the rings or washers may be displaced from one axial position to the opposite extreme positions, without being subjected to any external load is axial clearance. The mean value includes displacements with the rings or washers in different angular positions relative to each other and with the set of rolling elements in different angular positions in relation with the ring or washer.

3.2 Relation Between Radial and Axial Clearance

The diametrical clearance of radial ball bearings is given by

$$Pd = tdo - tdi - 2 \times db$$

Where,

Pd = Diametrical clearance

tdo = outer race track diameter
tdi = inner race track diameter
db = ball diameter

Distance between raceway groove curvatures in is given by
 $A = t_{ro} + t_{ri} - db$

Osculation ratio of outer and inner race is given by

$$f_i = t_{ri}/db \text{ and } f_o = t_{ro}/db$$

$$\text{Hence, } A = (f_i + f_o - 1) \times db = B \times db$$

Where,

fo = osculation ratio of outer race

fi = osculation ratio of inner race

tro = Track curvature of outer race

tri = Track curvature of inner race

$$B = f_i + f_o - 1$$

Bearing free contact angle is given by

$$\alpha = \cos^{-1}(1 - Pd/2.A)$$

Relative axial movement of inner rings with respect to outer rings under zero loads is given by:

$$P_e = 2.A.\sin \alpha$$

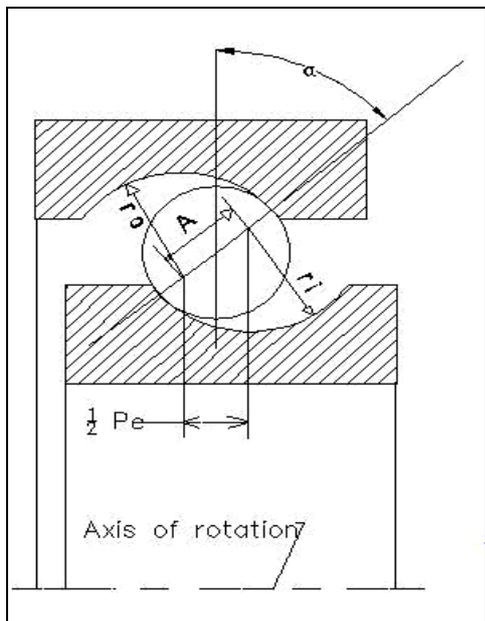


Figure 3: Radial Ball Bearing under Axial Movement

3.3 Axial Clearance in Angular Contact Ball Bearing and Measurement

By design single and double row angular contact ball and taper roller bearings have no predetermined clearance. The clearance is zero when the inner rings, rolling elements and outer ring is in contact with respect no load. With respect to the reference position, preload may be introduced during fitting. The axial clearance is shown in Fig. 5 is measured by the clearance measuring equipment, by the axial limiting positions of the inner rings while holding the outer ring stationary.

3.4 Angular Contact Ball Bearing Under Thrust Load

Axial deflection of angular contact ball bearings increase by applying the axial load is shown Fig.5 If Fa axial force applied on inner race, this results in an increase in the contact angle and axial shift to α_1 and δ_a respectively. The following formulation is generally used in calculation of shoulder diameter of ball bearings. By application of engineering, the amount of axial and radial load which an angular contact ball bearing could take can be computed.

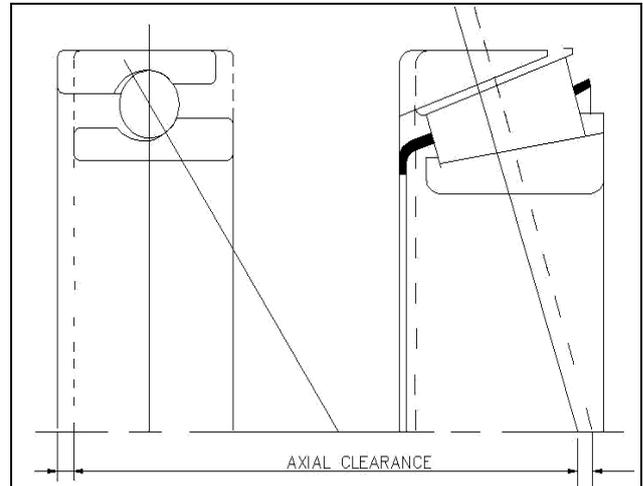


Figure 4: Axial Clearance in Angular Contact Ball Bearing

Distance between track centres is given by

$$A = B \times db$$

$$= (f_i + f_o - 1) \times db$$

Nominal contact angles given by

$$\alpha = \cos^{-1}(1 - pd / 2.A)$$

Where,

pd = diametrical clearance of angular contact ball bearings.

Nominal deflection along with the contact is given by

$$\delta_n = A \times (\cos \alpha / \cos \alpha_1 - 1)$$

With the help of Newton – Rapson method or trial and the error reducing the difference error to zero for contact angle α_1 , the following is arrived at:

$$F_a / (nb.K.db) = \sin \alpha (\cos \alpha / \cos \alpha_1 - 1)$$

Where,

nb = Number of balls

K = Axial deflection constant

α_1 = Contact angle after axial loading

db = Ball diameter

Axial deflection under axial load Fa

$$\delta_a = A \times \sin (\alpha_1 - \alpha) / \cos \alpha$$

This equation can also be used to the variation under axial load, preload and the external load and for checking the variation of axial clearances under different measuring loads. Finally, the operating clearance can be computed for best operating performance and the maximum life angular contact ball bearing. The solution of the above questions has been incorporated into the programmed as it requires digital computer for iteration.

4. Development of Design Methodology

Angular contact ball bearings are designed to support axial and radial load. These bearings are designed and manufactured with a set of contact angles. The design methodology of inner race and outer race for single row Angular contact ball bearing is developed and applied to double row angular contact ball bearings. The design of inner and outer race is carried out after optimization of angular contact ball bearing. This design methodology requires validation for manufacturing and testing

5. Optimization of Angular Contact Ball Bearings

Optimization of Angular contact ball bearing can be carried out by in-house developed software Obel 1.0. The program optimizes number of balls, ball diameter and pitch circle diameter, contact angle and number of rows. Based on these optimized parameters, the dynamic and static capacity is computed as per ISO 281-1990 and ISO 76-1987 respectively.

The basic dynamic load rating, C_r , for radial and angular contact ball bearings is given by

$$C_r = \text{bm.fc.} (i \cdot \cos \alpha)^{0.7} Z^{2/3} D_w^{1.8}$$

For $D_w \leq 25.4 \text{ mm}$

$$C_r = \text{bm.fc.} (i \cdot \cos \alpha)^{0.7} Z^{2/3} D_w^{1.4}$$

For $D_w > 25.4 \text{ mm}$

The basic static load rating, C_{or} , is given by

$$C_{or} = f_o \cdot i \cdot z \cdot D_w^2 \cdot \cos \alpha$$

Where

- bm = material and manufacture factor
- fc = factor depends on geometry of the bearing various components.
- f_o = factor depends on geometry of the bearing components and on the applicable stress level.

By carrying out the optimization of Angular contact ball bearings, the following optimized parameters are obtained:

Number of ball Angular contact ball bearings can accommodate (z).

Ball diameter, (D_w).

Circle diameter, (D_p).

Contact angle (α).

Number of row (i).

5.1 Calculation Method for Inner Race

From known optimized parameters as supplied in customer's drawings, and Fig 5. The geometrical relation is established for the inner race. The mathematical relations are formulated for following design parameter s for inner race of single and double row angular contact ball bearings:

- * tcdi : Track centre diameter.
- * tdi : Track diameter.
- * tdiA : Raceway contact diameter.
- * Tcdbi : Track centre from back race.
- * Tcfdi : Track centre from front race.

- * Tdfi : Track diameter contact point from back face.
- * Shoulder diameter and sill diameter , sdi1 and sdi2,

The calculation of shoulder diameter of back end face requires thrust load and this can be obtained by application engineering as already discussed. However, some empirical relation has been established for sill diameter, Sdi2 and shoulder diameter sdi1. The design of sill is important for assembly of bearings.

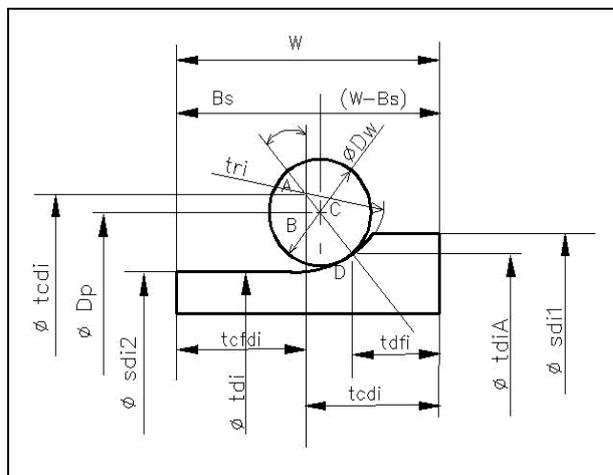


Figure.5: Optimized Design Parameters of Inner Race

5.2 Calculation Method for Outer Race

Similarly from Fig. 6 the mathematical relations are established for the following design parameter for the outer race, with the optimized parameters described above

- * tcd_o : Track centre diameter
- * tdo : Track diameter
- * tdoA : Raceway contact diameter
- * tcbdo : Track centre from back face
- * tcfdo : Track centre from front face
- * tdf_o : Track diameter contact point front face
- * Shoulder diameter and sill diameter , sdo 1 and sdo 2,

Calculation of shoulder diameter of back and face requires thrust loads which can be obtained by application engineering described earlier. However, some empirical relation has been established for sill diameter, Sdo 2 and shoulder diameter sdo 1. The design of sill is important for assembly of bearings.

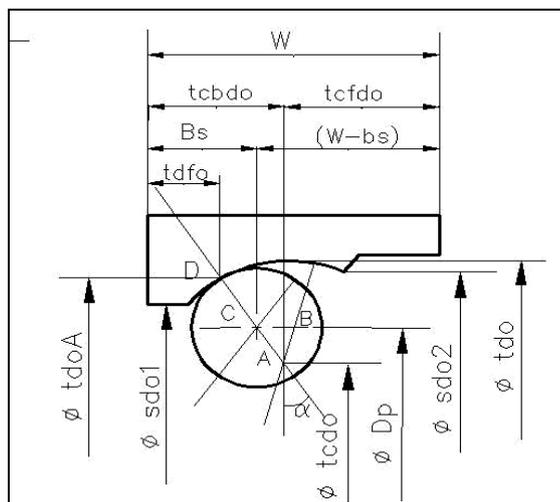


Figure.6: Optimized Design Parameters of Outer Race

6. Selection of Materials

The material and its quality play a predominant role for long survival of bearing. Earlier, the basic electric arc furnace was primarily used for production of AISI 52100 high-chromium-high-carbon steel for hardened ball and roller bearings. The steel making process was around 35 ppm with substantial amount of inclusions.

During the 1970s, the acid open hearth furnace was introduced, which improved the cleanliness i.e. oxygen contents reduced to 20 ppm and micro-inclusions reduced to over a period of time to virtually nil. In early eighties, the oxygen contents reduce to 15 – 10 ppm. Currently the steelmaking process is so effective that oxygen level can be maintained at around 7 ppm. The reduction of oxygen contents can lead to a tenfold improvement in future life. For commercial bearings application SAE 52100 and equivalent material is used with oxygen content less than 15 ppm. The customer has specified SAE 52100 grade used by the bearing manufacturer worldwide in wheel bearing application. The same material has been selected for development of the design. Hardness plays an important role in fatigue life. If the hardness of a bearing component is lower than HRC 60, its contact fatigue life will decrease sharply. For hardness less than HRC 60, the following relation between fatigue life and hardness HV is used:

$$L \propto HV^{6-6.7}$$

It has been established that hardness of rolling elements (balls) should be 1 to 2 HRC higher than races for better fatigue life. Accordingly, the hardness of races and balls are established.

7. Verification of Design Methodology

The design methodology has been verified with the I-DEAS design modeling software. The optimized parameter i.e. number of balls, ball diameter, pitch circle diameter, contact angle and number of rows. These data are taken as input to the program for design of angular contact ball bearing. It was found the result obtained from the programmed of design methodology were in agreement with a simple 2D design model sketch. The 2D design model tools were considered as

they need less time in drawing, and are simple to understand with assembly with inner, outer and balls. The most important factors for consideration are geometric constraints of the sketch which do not allow the user for extra dimensioning.

8. Conclusions

This is the first kind of development in bearings which called 1st generation wheel bearing i.e. double row angular contact bearing with split inner rings. With this development, it will be possible to cater important requirement of key customers. Bearing division is also in the process of development 2nd and 3rd generation wheel bearings. Although the design methodology will be same as discussed here, the designer will require a little different technology.

References

- [1] Liu, J., Shao, Y., Lin, C-M., "Vibration analysis of ball bearings with a localized defect applying piecewise response function", *Mechanism and Machine Theory*, Vol. 56 (2012) 156–169.
- [2] Kanga Y., Huang C., Lin C-M., Shen P., Changa Y., "Stiffness determination of angular-contact ball bearings by using neural network", *Tribology International*, Vol. 39 (2006) 461–469.
- [3] Mosleh, M., Bradshaw, K., "Role of component configuration in evaluation of accelerated rolling contact fatigue of ball bearings", *Wear*, Vol. 271 (2011) 2681–2686.
- [4] Nathan, A. B., Nagaraj, K., Forster N., Svendsen, V., "Critical stresses and strains at the spall edge of a case hardened bearing due to ball impact", *International Journal of Fatigue*, Vol. 47 (2013) 268–278.
- [5] Lin, C-M., "Analysis for the Stiffness of Ball Bearings, Master's thesis", Chung Yuan Christian University, Department of Mechanical Engineering (2002).
- [6] Mullick, A., "An Investigation in Stiffness of Rolling Contact", Ph.D dissertation, Chung Yuan Christian University, Department of Mechanical Engineering (1990).
- [7] Antoine, J-F., Abba, G., Molinari, A., "A New Proposal for Explicit Angle Calculation in Angular Contact Ball Bearing", *Journal of Mechanical Design, Transactions of the ASME*, Vol. 128 (2006) 468-478.
- [8] Antoine, J-F., Visa, C., Sauvey, C., "Approximate Analytical Model for Hertzian Elliptical Contact Problems", *Journal of Tribology, Transactions of the ASME*, Vol. 128 (2006) 660-664.
- [9] Sun, M. K., Kang, J. L., Sun, K. L., "Effect of bearing support structure on the high-speed spindle bearing compliance", *International Journal of Machine Tools & Manufacture*, Vol. 42 (2002) 365–373.
- [10] Wei, L., Jay, F. T., Kamman, J., "An integrated thermo-mechanical-dynamic model to characterize motorized machine tool spindles during very high speed rotation", *International Journal of Machine Tools & Manufacture*, Vol. 43 (2003) 1035–1050.

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