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

Distance between raceway grooves curvatures in is given by
A = tro + tri – db

Osculation ratio of outer and inner race is given by
Fi = tri/db and fo = tro/db
Hence, A = (fi +fo – 1) x db = B x db
Where,
fo = osculation ratio of outer race
fi = osculation ratio of outer race
tro = Track curvature of outer race
tri = Track curvature of inner race
B = fi + fo - 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:

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

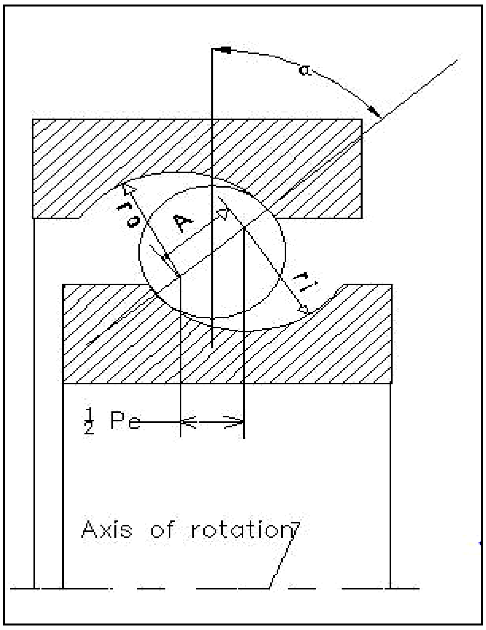

Figure 3: Radial Ball Bearing under Axial Movement

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 introduce 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 angel and axial shift to α1 and δa 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.

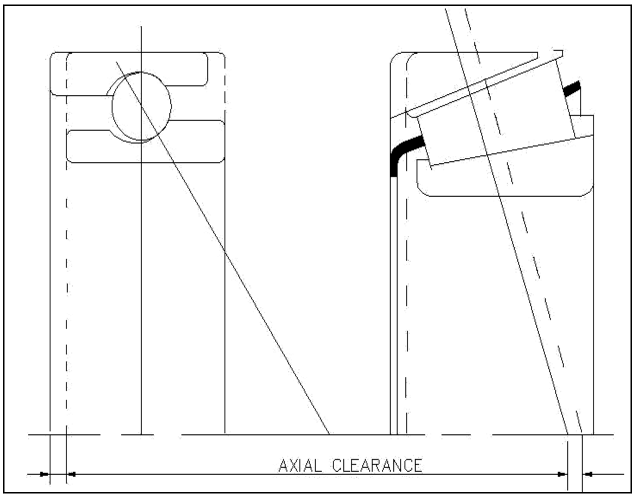

Figure.4: Axial Clearance in Angular Contact Ball Bearing

Distance between track centres is given by

$$A = B \times db$$

$= (fi + fo – 1) \times db$

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 α1, the following is arrived at:

$$Fa / (nb.K.db) = \sin \alpha (\cos \alpha / \cos \alpha 1 – 1)$$

Where,
nb = Number of balls
K = Axial deflection constant
α1 = 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, Cr, for radial and angular contact ball bearings is given by

$$Cr = bm.fc.\ (i.\ Cos\ \alpha)^{0.7}\ Z^{2/3}\ Dw^{1.8}$$
For $Dw \leq 25.4$ mm
$$Cr = bm.fc.\ (i.\ Cos\ \alpha)^{0.7}\ Z^{2/3}\ Dw^{1.4}$$
For $Dw > 25.4$ mm

The basic static load rating, Cor. Is given by
$Cor = fo.\ i.\ z\ .\ Dw^2.\ \cos\alpha$
Where
bm = material and manufacture factor
fc = factor depends on geometry of the bearing various components.
fo = factor depends on geometry of the bearing components and on the applicable stress level.

By carting 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, (Dw).
Circle diameter, (Dp).
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.

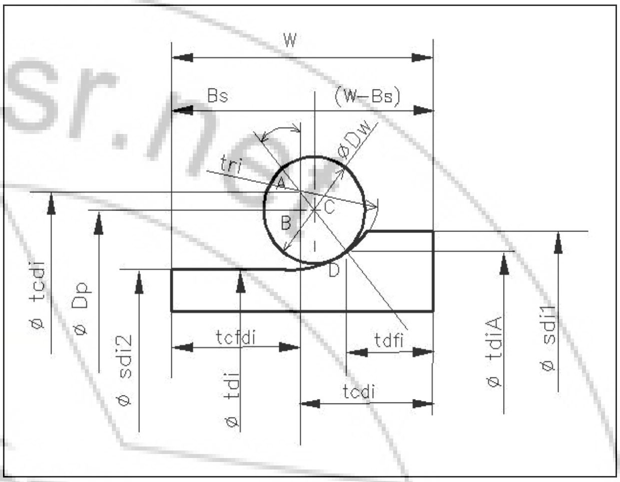

Figure.5: Optimized Design Parameters of Inner Race

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

* tcdo : Track centre diameter
* tdo : Track diameter
* tdoA : Raceway contact diameter
* tcbdo : Track centre from back face
* tcfdo : Track centre from front face
* tdfo : 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.

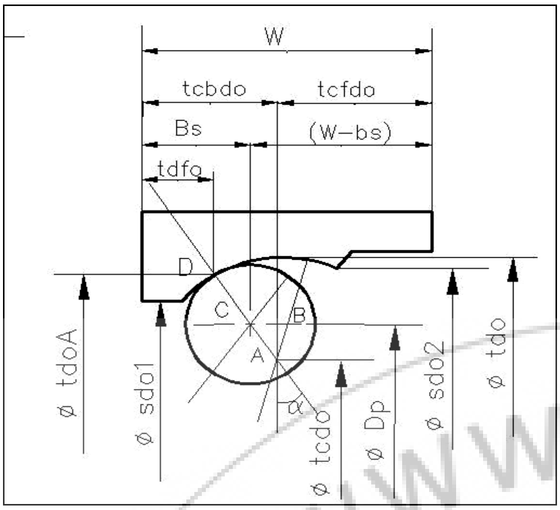

Figure.6: Optimized Design Parameters of Outer Race

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:

$L \alpha HV^{6\text{-}6\text{-}7}$

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.

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Author Profile

Shashi Ranjan Dutta has completed his M.Tech from NIT Jamshedpur and B.Tech from AIME (Institute of Engineers). He has work experience of 14 years in TATA Steel and 14 years in teaching. He is presently working as HOD in Mechanical Engineering Department, in SIEM Siliguri, West Bengal, India.

Sunil Kumar Singh has completed his M. Tech from Kalyani Government Engineering College and B.Tech from RVSCET. He is presently working as Assistant Professor in Mechanical Engineering Department, in SIEM Siliguri, West Bengal, India.

Rakesh Ranjan has completed his M.Tech from IIT Kharagpur and B.Tech from Institute of Mechanical Engineers. He is presently working as Assistant Professor in Mechanical Engineering Department, in SMIT Sikkim, India.