Fatigue Life Analysis of Thrust Ball Bearing Using ANSYS

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Abstract

This paper compares the total deformation of thrust ball bearing & contact stress b/w ball & raceways & its effect on fatigue life of thrust ball bearing. The 3-Dimensional Modeling has been done through modeling software Pro-e wildfire-5.0. The parts assembly is also done in Pro-e wildfire-5.0 & analysis has been done through ANSYS-14. An analytical method is good, less expensive and gives the best results. Analytical results give good agreement with the experimental data. The thrust ball bearings are subjected to various, thrust & dynamic loads, which simulated easily through Pro-E software & analysis because experimentally calculation is very complicated. The general theory used for calculating the Fatigue life of Bearing is basic life rating theory. The material taken for the Bearing is AISI8720H. In this study we have used various analysis codes and got a good result through these codes.

Keywords: Fatigue limit stress. Pro-E 5.0, ANSYS 14, ball to raceway contact, contact stress analysis. Thrust bearing, axial load.

I. Introduction

A type of bearing designed to reduce friction by carrying thrust or axial loads. Thrust bearings can be either plain or anti-friction bearings. The type of component supported determines the type of thrust bearing used. A thrust bearing is a particular type of rotary bearing. Like other bearings they permit rotation between parts, but they are designed to support a high axial load while doing this. Single direction thrust ball bearings consist of a shaft washer, a housing washer and a ball and cage thrust assembly. The bearings are separable so that mounting is simple as the washers and the ball and cage assembly can be mounted separately.

Fatigue life is the design basis for selecting and sizing Ball contact bearings. As the balls continuously make contact with new raceway surfaces, a given point on either the raceway or ball surface, is loaded and unloaded. This type of loading in any machine component is recognized to reduce the amount of load that may be applied before failure occurs. Cracks propagate as load is continuously reapplied. When several cracks propagate to a point where they intersect, a chunk of raceway surface can be expected to break off and a spall is formed. This is usually considered the point of fatigue failure.

II. General Theory for Bearing Life Calculation

The fatigue life of the thrust ball bearing can be estimated by “Lundberg & Palmgren theory”,

\[ L = \left( \frac{C}{P} \right) ^ n \]

- Where
- \( L_{10} \) = rated fatigue life with a statistical reliability of 90%
- \( P \) = bearing equivalent load
- \( C \) = basic radial dynamic load rating (Get from individual bearing selection charts)

There are generally two factors used in thrust ball bearing:

a. The bearing fatigue life Criterion. - Under ideal conditions the repeated stresses developed in the contact areas b/w the ball & raceways eventually can result in the fatigue of the material. In most applications the fatigue life is the maximum useful life of a Bearing.

b. Static loading criterion. - A static load is load acting on a non rotating bearing. The permissible static load is dependent upon the permissible magnitude of permanent deformation. Depending on requirements for smoothness of operation, friction, higher or lower static load limits may be tolerated.
Basic Life Rating

The Basic Life Rating (L10) is defined in specification JIS B1518 "Dynamic load ratings and rating life for ball bearings" as follows:

The Basic Life Rating is the life obtained with 90% reliability, when an individual bearing or an identical group of bearings are manufactured with common materials, common manufacturing processes and quality, and operate under the same conventional conditions. L10 Life is the accumulated rotation where 90% of survive without material flaking when they are operated under fixed conditions, of a population of bearings.

The calculation formula for the Basic Life Rating is the following.

\[
L_{10} = \left( \frac{C_r}{P_r} \right)^{3/2} \quad \text{L}_{10} \quad \text{: Basic Life Rating in millions of revolutions} \\
C_r \quad \text{: Basic Dynamic Load Rating} \\
P_r \quad \text{: Equivalent Dynamic Radial Load Factor}
\]

There is a relationship between the Basic Life Rating (revolutions) and Basic Life (time).

\[
L_{10} = \left( \frac{10^6}{60^{n-h}} \right) \times \left( \frac{C_r}{P_r} \right)^{3/2} \quad \text{n} \quad \text{: Rotation Speed (min}^{-1}) \\
h \quad \text{: Time (hours)}
\]

Dynamic Equivalent Radial Load Factor (Pr)

The Dynamic Equivalent Radial Load Factor is defined as "the direction and magnitude to the bearing, which is able to obtain the same life under the actual load and rotation conditions".

From the calculation formula and the table below, the axial and the radial loads are replaced by the Dynamic Equivalent Radial Load Factor (Pr).

\[
Pr = XFr + YFa
\]

X and Y are taken from the table below.

Fr = Radial load (N or kgf)
Fa = Axial load (N or kgf)

<table>
<thead>
<tr>
<th>Axial Load Ratio</th>
<th>( \frac{F_a}{F_r} \leq \varepsilon )</th>
<th>( \frac{F_a}{F_r} &gt; \varepsilon )</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \frac{F_a}{iZD_w} )</td>
<td>( X )</td>
<td>( Y )</td>
</tr>
<tr>
<td>0.172</td>
<td>{ 0.0175 }</td>
<td>{ }</td>
</tr>
<tr>
<td>0.345</td>
<td>{ 0.0352 }</td>
<td>{ }</td>
</tr>
<tr>
<td>0.689</td>
<td>{ 0.0703 }</td>
<td>{ }</td>
</tr>
<tr>
<td>1.03</td>
<td>{ 0.105 }</td>
<td>{ 0 }</td>
</tr>
<tr>
<td>1.38</td>
<td>{ 0.143 }</td>
<td>{ }</td>
</tr>
<tr>
<td>2.07</td>
<td>{ 0.211 }</td>
<td>{ }</td>
</tr>
<tr>
<td>3.45</td>
<td>{ 0.352 }</td>
<td>{ }</td>
</tr>
<tr>
<td>5.17</td>
<td>{ 0.527 }</td>
<td>{ }</td>
</tr>
<tr>
<td>6.89</td>
<td>{ 0.703 }</td>
<td>{ }</td>
</tr>
</tbody>
</table>

\( i \) : No. of rows

\( Z \) : No. of balls

\( D_w \) : Ball Diameter (mm)

The values for X and Y that are not in the above table shall be calculated by linear interpolation.
III. Selection of Bearing and Modelling of the Bearing

Computational 3 dimensional simulation & analysis indicates the numerical solution of differential governing equations of thrust ball bearings, with the help of computers. This technique has a wide range of engineering applications. In the field of dynamic research this technique has become increasingly important and it is prominent for studying bearings & other materials. As the name implies, the thrust bearings are used either to absorb axial shafts load or to position shafts. The single row radial ball bearing is probably the most widely used ball bearing and is employed in many modified forms. It is also known as deep groove type. This bearing is capable of taking pure thrust load.

**Thrust ball bearings** - It composed of ball bearings supported in a ring, can be used in low thrust applications where there is little axial load.

![Fig-1 Thrust Ball Bearing](image)

**Bearing Race**

The rolling-elements of a rolling-element bearing ride on races. The large race that goes into a bore is called the outer race, and the small race that the shaft rides in is called the inner race.

**Design of thrust ball bearing**

In the case of ball bearings, the bearing has inner and outer races and a set of balls. Each race is a ring with a groove where the balls rest. The groove is usually shaped so the ball is a slightly loose fit in the groove. The race also dents slightly where each ball presses on it. Thus, the contact between ball and race is of finite size and has finite pressure. Note also that the deformed ball and race do not roll entirely smoothly because different parts of the ball are moving at different speeds as it rolls. Thus, there are opposing forces and sliding motions at each ball/race contact.

![Fig-2 Block Diagram of Thrust Bearing](image)

IV. Simulation & Analysis with 3 Dimensional Software

Computational 3 dim simulation & analysis indicates the numerical solution of differential governing equations of thrust ball bearings, with the help of computers. Computational 3 dim simulation & analysis also provides the convenience of being able to switch off specific terms of governing equations. This permits the testing of theoretical models and, inverting the connection, suggesting new paths for theoretical explorations. Computational 3 dim simulation & analysis provides five major advantages compared with various experimental, thrust & dynamic loads:

a. Lead time in design and development is significantly reduced.
b. It can simulate flow conditions not reproducible in experimental model test.
c. It provides more detailed and comprehensive information.
d. It is increasingly more cost-effective than real time testing.

V. Dimensions

1. The ball cage is a formed carbon steel design.
2. Note that one washed is ID piloted and the other is OD piloted.

**TABLE-1**

<table>
<thead>
<tr>
<th>Attributes</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bearing Type</td>
<td>with grooved raceway(s)</td>
</tr>
<tr>
<td>Brand model</td>
<td>SKF 351006 A</td>
</tr>
<tr>
<td>category</td>
<td>Thrust ball bearing</td>
</tr>
<tr>
<td>Bore Dia (d)</td>
<td>10.0000</td>
</tr>
<tr>
<td>Outer Dia (D)</td>
<td>24.0000</td>
</tr>
</tbody>
</table>

Attributes Values
Radius (min) (rs) 0.300
Static Load Rating (Cor) 14000
Dynamic Load Rating (Cr) 10000
Max Speed (Grease) \(x1000\) RPM 7
Max Speed (Oil) \(x1000\) RPM 10
Height (H) 9.0000
Outer Dia. Pilot (D1) 24.0000
Bore Dia. Clearance (d1) 11.0000
Weight (g) 22.00
Material 52100 chrome steel, or equivalent.

(B) Material Properties
Material used for thrust ball bearing is AISI 52100 alloy steel whose properties are same for in ANSYS 14.

Bulk modulus (typical for steel)= 140 GPa,
Shear modulus (typical for steel)= 80 GPa,
Young’s Modulus E = Elastic modulus =190-210 GPa,
Poisson’s Ratio \(\nu\) = 0.27-0.30,
Tensile Strength = 3445 MPa

(C) Boundary Conditions
During analysis three boundary conditions is considered which are shown:
1. Fixed support at the Outer Dia. Pilot (D1).
2. Standard gravitational is also taken for analysis.
3. Force acted at the Outer Dia (D) which is calculated from Bearing life calculation Theory.

Force = 2500 N, 5000N & 7500 N

VI. Boundary Conditions, Material Properties and Meshing (FEM Modelling)

(A) Meshing
During analysis default meshing method is used in ANSYS 14 for bearing and fine sizing type is shown in figure. Breaking the part into many, small pieces (a fine mesh) will give more accurate results, but will use up more time and memory.

(VII. Results and Comparison Table
We have found the contact deformation in different loads & compared the result with analytical method & software result. Results obtained at various forces which are same on bearing which is acted on the top of the bearing is discuss below, these forces got from basic life rating theory for thrust bearing:

Force = 2500 N, 5000n & 7500N has been compared with the software & get the Deformation & Equivalent (von-Mises) Stress in the bearings are shown in figure 6(a), 6(b).

\[ P=2500N \text{ for total deformation} \]
Fig-6 Total deformation

P=2500N for Equivalent (Von-Mises) Stress

Fig-7 Equivalent (Von-Mises) Stress

P=5000N for total deformation

Fig-8 Total deformation

P=5000N for Equivalent (Von-Mises) Stress

Fig-9 Equivalent (Von-Mises) Stress

P=7500N for total deformation

Fig-10 Total deformation

P=7500N for Equivalent (Von-Mises) Stress

Fig-11 Equivalent (Von-Mises) Stress
Table 2 shows the comparison of Total deformation b/w Analytical result & Experimental Result in thrust ball bearing at 3 different loads. We have got the Max. Equiv. Von-Mises Stress

<table>
<thead>
<tr>
<th>S.No</th>
<th>Force (N)</th>
<th>Deformation in Thrust bearing (Analytical) µm</th>
<th>Difference µm</th>
<th>Max. Equiv. Von-Mises Stress in (Pa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.</td>
<td>2500</td>
<td>0.5163</td>
<td>0.74</td>
<td>0.7827</td>
</tr>
<tr>
<td>2.</td>
<td>5000</td>
<td>1.7726</td>
<td>1.18</td>
<td>1.1742</td>
</tr>
<tr>
<td>3.</td>
<td>7500</td>
<td>1.7726</td>
<td>1.55</td>
<td>1.4364</td>
</tr>
</tbody>
</table>

VIII. Result
Comparison of the simulation results with the previous research work result shows good qualitative agreement, both in shape & magnitude of wear profile. The analytical result is equivalent the simulation result. The total deformation of bearing is compared for 3 different loads and the difference between result is varies from 0.01 µm to 0.5 µm. It has also been shown that the changes due to wear causes high pressure peaks at the contact area, which explains that bearings fail at the contact area under working conditions.

IX. Concison
This research shows the stress developed & deformation in Thrust Bearing is less as compared to Experimental result in same force conditions, material properties, thickness and boundary condition. In this paper The Basic Life Rating theory is used to calculate the load, but more accurate result is obtained experimentally & has been compared with the Computational 3 dim simulation & analysis which gives exact force on the Balls & raceways contact. It helps to calculate the exact stress developed & Deformation occurs on Contact b/w ball & bearing raceways in Finite Element Analysis.

X. Acknowledgement
This work has been performed under the supervision of Professor Upendra Kumar Joshi read the manuscript & made the valuable comments. I would also like to thanks HOD mechanical Engg Department for giving me permission to use the software Lab of Department.

XI. References
[9] Effects of inclusion size and stress ratio on fatigue strength for high-strength steels with fish-eye mode failure Chengqi Sun, Zhengqiang Lei, Jijia Xie, Youshi Hong, International 


Friction torque in grease lubricated thrust ball bearings Tiago Coussseau a,n, BeatrizGrac-a a, ArmandoCampos b,1, JorgeSeabra, Tribology International 44 523–531, ELSEVIER (2011)


A finite element study of short cracks with different inclusion types under rolling contact fatigue load, Arne Melander, PII: S0142-1123, 00045-X, ELSEVIER (1996)

STANDARD HANDBOOK OF MACHINE DESIGN, Joseph E. Shigley.