Numerical simulation of the effect of angular misalignment on the dynamic behaviour of bearing

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Abstract. Bearing has been used extensively in numerous applications and their unplanned failure has a consequential effect on the smooth operation of the machinery. A slight misalignment in bearing has a detrimental effect on the smooth running of most machines. Hence, the paper leverages finite element technique to simulate the consequential effect of different degrees (0.1°, 0.2°, 0.3°, 0.4°, 0.5°) of misalignment on the dynamic behaviour of a cylindrical roller bearing subjected to typical operating conditions of an airflow root blower. The results of the study show that during operation, the temperature and Hertzian stress developed increased with an increase in the degree of misalignment and operating/rotating speed, and the maximum Hertzian stress was developed on the outer ring of the bearing in all the degrees of misalignments and operational speeds considered. Thus, making the outer ring of the bearing component, the most prone to failure during operation in the presence of misalignment.

Keywords: Bearing, misalignment, Hertzian stress, failure, friction

1 Introduction

Bearings are an essential component of all rotating machines and they function as a support to machines, and about 90 per cent of their failure during operation is catastrophic [1]. The radial bearings are often considered one of the most crucial components of rotating machinery in industrial applications. This is because they are designed to withstand high loads and are usually mounted on expensive and complex machines such as cars, airflow root blowers, etc [2]. Thus, proper selection, monitoring, and maintenance of bearings are paramount in order to avert possible component damage that could result from bearing failure during operation [3]. In practice, different types of bearings are in use, and they include thrust bearings, cylindrical roller bearings, spherical roller bearings, and tapered roller bearings [4]. Several designs and technologies have been adopted in the aspect of heat treatment of bearing

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materials, machining, bearing components contacts, and the use of different simulation software in a bid to improve the knowledge of preventing the failure of bearings [1].

In a roller bearing, the defects that occur are classified according to the bearing element/component damage like the inner raceway defect, outer raceway defect, ball defect or a combination of the defects [5]. Being a key part of rotating machinery, cylindrical roller bearings play a key role in the load-bearing ability and friction characteristics of airflow root blowers [6]. In service, undesired vibration resulting from variable stiffness, internal clearance, localized or distributed defects, or varying compliance contributes to the unplanned failure of bearings. The distributed defects experienced in bearings are often ascribed to manufacturing errors like inappropriate surface roughness, wrong installation, misaligned raceways, waviness and off-size rolling element [6-8]. On the other hand, surface fatigue failure like pits, spalls and cracks are presumed to be responsible for the localized failure in bearings [6, 7].

In the power generation industry, the frequent and unplanned failure of bearings due to misalignment, particularly in the airflow root blower has resulted in the production of low-quality demineralized water. This low-quality produced demineralised water has significantly affected the smooth running of the plant as a substantial amount of time and cost is spent during the replacement of the failed component. Hence, this study leverages the finite element (FE) technique (using Abaqus, a commercial finite element analysis software) in the determination of the dynamic behaviour of cylindrical balls roller bearing of an airflow root blower, fabricated from X20 martensitic stainless steel (X20 CrMoV12-1, DIN EN 10302 standard) and cage made from polyamide 66, nylon. The bearing of the airflow root blower is operated without and with different degrees of misalignment at different operating/rotating speeds in order to determine the effect of the operating conditions on the bearing.

2 Bearing loading and heat generation

In this section, the different expressions for the determination of the heat generated and the stress developed in a cylindrical roller bearing are presented. Also presented is the analysis of the motion of bearings subjected to radial loading.

2.1 Loading of bearings

If other factors such as operational speed, type of bearing, clearance between ball and cage, etc., and not the applied load are used to determine the size of a bearing system, the bearing should be loaded slightly with respect to its size and load-carrying capacity. If the load the bearing is subjected to is too light, other failure mechanisms aside from fatigue i.e raceway smearing, cage damage, or skidding tend to prevail. Hence, to maintain a satisfactory operating condition, it is important that roller bearings must be subjected to a minimum load at all times [9, 10]. For bearings with spherical balls, the minimum load, \( P_{m} \) must be equal to 0.01C while for bearings with cylindrical balls, the minimum load, \( P_{m} \) must be equal to 0.02C, where C represents the basic static load rating in kN [9].

2.2 Motion analysis of bearings

When a roller bearing is subjected to radial loading, the inner ring will rotate at an angular velocity, \( \omega_i \) thus, driving the cage and the balls at an angular velocity, \( \omega_c \) around the rotation
axis and, the balls of the bearing revolve along their own axis at an angular velocity \( \omega_r \) as shown in Fig. 1.

![Fig. 1. Motion and bearing load analysis diagram [11]](image)

In spherical roller bearings, skid between the balls and the raceway is negligible, this type of bearing is often used for heavy load and low-speed applications, and the roller-raceway motion relationship can be assumed to be pure rolling [11, 12]. Thus, the average linear velocities of contact of the roller-inner raceway and the roller-outer raceway can be expressed as follows [11]

\[
U_i = 0.5d_m \left[(1 - \gamma)(\omega_i - \omega_c) + \left(\frac{D}{d_m}\right)\omega_r\right] \\
U_o = 0.5d_m \left[(1 + \gamma)\omega_c + \left(\frac{D}{d_m}\right)\omega_r\right]
\]

2.3 Heat generated due to contact between the cage and the inner ring

During the operation of a roller bearing, the cage rides on the inner ring surface such that the developed friction force between the cage and the inner ring surface \( F_{CS} \) can be computed using Petroff’s law [13],

\[
F_{CS} = \frac{2\pi \varphi b C r^2 (\omega_i - \omega_c)}{\delta_c}
\]

and the heat generated due to friction between the cage and the surface of the inner ring, \( H_{CS} \) can be computed as

\[
H_{CS} = 0.5d_m F_{CS}(\omega_i - \omega_c)
\]

where \( F_{CS} \) is the force (kN), between the inner ring and surface \( \varphi \) is lubricant oil dynamic viscosity, \( \omega_i \) is the angular velocity (rad/s) of the inner ring \( \omega_c \) is the angular velocity (rad/s) of the cage, \( \delta_c \) is the clearance (m), between the roller and inner ring, \( r \) is the radius (m), of component \( d_m \) is the pitch diameter (m), \( b \) is the width of contact (m), \( \alpha \) is the nominal contact angle, and \( \gamma = \frac{D}{d_m} \cos \alpha \).

2.4 Contact heat generation between roller-cage pocket

The frictional contact force, \( F_p \) between the roller-cage pocket, is given as [14]

\[
F_p = \frac{2\pi \varphi b \delta \omega_r}{\delta_p}
\]

And the heat generated, \( H_p \) in Z roller-cage pocket contact is given as

\[
H_p = ZD F_p \omega_r
\]
where \( l_p \) is the axial length (m) of the cage pocket, \( \omega_r \) is the angular velocity (rad/s), of the roller rotating around its own axis, \( \delta_p \) is the clearance (m) between roller and cage \( Z \) is the number of rollers/roller-cage pockets and \( b_p \) is the width of the elliptic point contact (m).

### 2.5 Heat generated by roller churning

In the presence of lubricant and air, rollers rotating around a shaft experienced drag force \( F_d \), which results in the generation of heat. The experienced drag force [12] is expressed as follows:

\[
F_d = \frac{1}{8} C_d \rho D l_e (d_m \omega_c)^2
\]

and the total heat generated \( H_d \), due to the drag force and \( Z \) roller churning is given as:

\[
H_d = 0.5ZF_d d_m \omega_c
\]

where \( D \) is roller diameter (m), \( C_d \) is the roller drag force coefficient, and \( \rho \) is the mass density of the roller (kg/m³)

### 2.6 Heat generation rate in roller-raceway contact

In order to obtain the total heat generated \( H_{RR} \) due to roller-raceway contact, the product of the expression for the contact due to roller slicing linear velocity and the roller-raceway lubricating shear stress can be integrated as expressed by Ma et al. [11, 15] to give the following final expression:

Since the components of a spherical roller bearing are the outer ring, inner, and cage; the total heat generated in the bearing can be expressed as follow:

\[
H_{Tot} = H_d + H_{CS} + H_P + H_{RR}
\]

### 2.7 Developed contact stress

When two cylindrical balls with parallel axis, and radii \( R_A \) and \( R_B \) are in contact as shown in Fig. 2, the developed contact pressure and Hertzian stress can be computed as follows [16, 17]:

![Fig. 2. Two cylindrical surfaces in contact [10, 17]](image-url)
The reduced radius \( R' \), and reduced Young’s modulus \( E' \), of the two surfaces in contact are computed as follows:

\[
\frac{1}{R'} = \frac{1}{R_A} + \frac{1}{R_B} \tag{10}
\]

\[
\frac{1}{E'} = \frac{1}{2} \left[ \frac{1-v_A^2}{E_A} + \frac{1-v_B^2}{E_B} \right] \tag{11}
\]

Also, the width of the contact area is determined using the following expression

\[
b = \sqrt{\frac{4WR'}{\pi E'}} \tag{12}
\]

Hence, the developed maximum pressure due to the contact of the two surfaces during operation is computed using the following formula

\[
P_{\text{Max}} = \frac{W}{\pi bl} \tag{13}
\]

and the developed Hertzian stress due to the contact between \( Z \) cylindrical surfaces is calculated as follows:

\[
\sigma = 0.304P_{\text{Max}} \times Z \tag{14}
\]

\( E_A \) and \( E_B \) are Young’s moduli (Pa), of materials of cylinders A and B, respectively, \( v_A \) and \( v_B \) are the Poisson’s ratios of materials of cylinders A and B, \( R_A \) and \( R_B \) are the outer radii (m) of cylinders A and B in contact \( R' \) is the reduced radius (m), \( E' \) is the reduced Young’s modulus (Pa), \( l \) is the length of contact (height of cylindrical roller) (m), \( b \) is the width of contact (m), and \( W \) is the normal applied load (N).

### 3 Development of FE model

The model of cylindrical bearing was developed using Abaqus (a commercial finite element analysis software). The developed model is made of one inner, outer ring, and pin-type cage; and 13 cylindrical balls. The cage is made from polyamide (nylon) 66 while the balls, inner and outer rings are made from X20 martensitic stainless steel. Fig. 3 shows the parts and assembly model of the bearing system while Tables 1 and 2 show the dimensions of the components of the bearing and their mechanical and thermal properties. The choice of polyamide 66 as a material for the cage in a bearing is attributed to its good sliding properties that lead to friction reduction and allow for high speed during operations. In addition, a significant reduction in the risk emanating from seizure and form secondary damage has been reported when polyamide material was used as a cage in bearings. Compared to bearings whose cages are made of metal, polyamide cage-incorporated bearings with limited lubrication can operate for a longer period before failure [18].

![Fig. 3. Components of a bearing and assembly model](image)
Table 1. Dimensions of the bearing components [10]

<table>
<thead>
<tr>
<th>Components</th>
<th>Internal diameter (mm)</th>
<th>External diameter (mm)</th>
<th>Height (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inner ring</td>
<td>50.0</td>
<td>65.0</td>
<td>24.0</td>
</tr>
<tr>
<td>Outer ring</td>
<td>91.4</td>
<td>110.0</td>
<td>27.0</td>
</tr>
<tr>
<td>Cage</td>
<td>71.5</td>
<td>87.5</td>
<td>24.0</td>
</tr>
<tr>
<td>Cylindrical ball</td>
<td>-</td>
<td>14.5</td>
<td>16.0</td>
</tr>
</tbody>
</table>

Table 2. Material properties of the bearing components [19-22]

<table>
<thead>
<tr>
<th>Material properties</th>
<th>Polyamide (Cage)</th>
<th>X20 steel</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density (kg/m3)</td>
<td>1 310</td>
<td>7 800</td>
</tr>
<tr>
<td>Poisson’s ratio</td>
<td>0.41</td>
<td>0.28</td>
</tr>
<tr>
<td>Module of elasticity (GPa)</td>
<td>2.95</td>
<td>200.00</td>
</tr>
<tr>
<td>Thermal expansion (× 10⁻⁶K⁻¹)</td>
<td>66.00</td>
<td>10.00</td>
</tr>
<tr>
<td>Specific heat capacity (J/kgK)</td>
<td>1 680</td>
<td>46.00</td>
</tr>
<tr>
<td>Thermal conductivity (W/m²K)</td>
<td>0.25</td>
<td>28.00</td>
</tr>
</tbody>
</table>

The coupling interaction was created between the cage and the balls, the inner ring and the balls, and the outer ring and the balls, and a general coefficient of friction of 0.002, was applied to the entire assembly in the interaction step. For the analysis of the bearing, a dynamic explicit step was used. The components of the bearing assembly were bounded mechanistically such that the balls, inner ring, and cage are allowed to undergo displacement and rotation, while the outer ring is fixed as depicted in Fig. 4. Also, the inner ring was subjected to a body load of 160 kN, while the cylindrical balls and inner ring were subjected to an angular velocity of 157 and 300 rad/s respectively without and with different degrees of misalignment (0.1°, 0.2°, 0.3°, 0.4°, 0.5°). The entire bearing assembly was assigned a temperature of 25 °C in the predefined field, as this represents the value of room temperature. The operating conditions used in this study represent the typical operating condition of a cylindrical roller bearing in the airflow root blower of a power generation plant.

Fig. 4. Applied mechanical boundary conditions to bearing

Fig. 5. Mesh convergence plot
The developed bearing assembly has a total of 83 230 C3D8T (linear hexahedral elements of type) and a total of 105 285 nodes, consisting of 5 208 elements from the inner ring, 41 600 elements from the 13 cylindrical balls, 9 734 elements from the outer ring and 26 688 elements from the cage. A mesh size or global seed of 2 mm was assigned to the outer and inner ring while cylindrical balls and cage were assigned a 1 mm mesh size. The used mesh sizes were arrived at after conducting mesh convergence studies [19, 23-29]. Using these mesh sizes, suitable results with reasonable computational time were obtained. Depicted in Fig. 5 is the mesh convergence study graph conducted on the bearing assembly with the outer ring being the component of focus, while Fig. 6 shows the assembly and part mesh of the bearing.

4 Results and discussion

The contour plot for the temperature distribution profile in the roller bearing with an angular misalignment of 0.1° and without misalignment under operating/rotating speeds of 157 rad/s (1 500 rpm) and 300 rad/s (2 850 rpm) is shown in Fig. 7. From the temperature distribution plot, it was observed that the higher the operating/rotating speed, the higher the temperature developed in the bearing assembly. Based on this, the roller bearing operated at a rotational speed of 300 rad/s developed a higher temperature with a value of 118.2 °C, while the bearing operated at a rotational speed of 157 rad/s developed a lower temperature with a value of 49.03 °C when the bearing was misaligned at an angle of 0.1°. Table 3 shows the maximum temperature developed for the 5 different case scenarios of bearing misalignments (0.1°, 0.2°, 0.3°, 0.4°, 0.5°) for the two different rotating speeds considered From the table, it was discovered that the developed temperature increases with an increase in the degree of misalignment and rotational speed.

The distribution of the temperature during service across the components of the bearing showed that the tip of the cylindrical balls experiences the highest temperature due to its constant contact with both the inner ring and outer ring during operation, while the outer ring experiences the least thermal effect due to its fast heat loss to its surrounding. On the meshed contour plot, the blue regions represent parts of the model that experience the least temperature, the green regions represent the parts that experience an intermediate temperature
and the red region experience the highest temperature. It is worth noting that the contact of the cylindrical balls with the inner and outer ring, and the characteristic high operating speed lead to friction, which consequently results in the rise in temperature, thermal expansion and stiffness of the bearing during operation [30].

![Temperature Distribution Profile Contour Plot and Results of Bearing with No Misalignment and with 0.1° Misalignment for the Different Rotational Speeds](image)

**Fig. 7.** Temperature distribution profile contour plot and results of bearing with no misalignment, and with 0.1° misalignment for the different rotational speeds

<table>
<thead>
<tr>
<th>Angular of misalignment (°)</th>
<th>Temperature (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.1</td>
<td>49.03</td>
</tr>
<tr>
<td>0.2</td>
<td>175.3</td>
</tr>
<tr>
<td>0.3</td>
<td>176.0</td>
</tr>
<tr>
<td>0.4</td>
<td>176.6</td>
</tr>
<tr>
<td>0.5</td>
<td>231.1</td>
</tr>
</tbody>
</table>

**Table 3.** Results of the maximum temperature developed in the bearing for the different angular misalignment and rotational speeds

<table>
<thead>
<tr>
<th>Angular of misalignment (°)</th>
<th>Temperature (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.1</td>
<td>128.8</td>
</tr>
<tr>
<td>0.2</td>
<td>532.8</td>
</tr>
<tr>
<td>0.3</td>
<td>534.4</td>
</tr>
<tr>
<td>0.4</td>
<td>536.2</td>
</tr>
<tr>
<td>0.5</td>
<td>741.2</td>
</tr>
</tbody>
</table>

Depicted in **Fig. 8** are the contour plots of the contact stress distribution profile developed on the bearing and its components without misalignment and with an angular misalignment of 0.1° for the two considered different rotational speeds. From **Fig. 8**, it was observed that when the bearing was subjected to angular misalignment of 0.1°, the maximum contact/Hertzian stress with a value of 896.6 MPa was developed in the analysis subjected to a rotational speed of 300 rad/s while the least stress with a value of 680.8 MPa was developed when the rotating speed of the bearing was 157 rad/s. On the meshed contour plot, the blue, green, yellow and red regions show in increasing order how the stress increase across the components of the bearing. Based on these colour notations, the blue colour denotes the regions of the component that developed the least Hertzian stress while the red colour represents the regions of the component that develop the highest Hertzian stress. **Table 4** shows the maximum Hertzian stress developed in the bearing for the various considered angular misalignments and rotational speeds. For all investigated rotational speeds and angular misalignments, the maximum Hertzian/contact stress was developed at the outer ring (at the point of possible contact between the edge of the cylindrical ball and the outer ring).
For the entire assembly, the cage made of polyamide material experienced the least contact stress due to its low interference with other components of the bearing during operation.

Owing to the thermo-mechanical contact stress developed in the roller bearing assembly with misalignment, the outer ring is the most prone to thermo-mechanical failure in service due to the higher stress developed on the component.

![Contact/Hertzian stress distribution contour plots and results of bearing with no misalignment, and with 0.1° misalignment for the different rotational speeds](image)

**Fig. 8.** Contact/Hertzian stress distribution contour plots and results of bearing with no misalignment, and with 0.1° misalignment for the different rotational speeds

<table>
<thead>
<tr>
<th>Angular of misalignment (°)</th>
<th>Hertzian stress (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>157 rad/s</td>
</tr>
<tr>
<td>0.1</td>
<td>680.2</td>
</tr>
<tr>
<td>0.2</td>
<td>641.8</td>
</tr>
<tr>
<td>0.3</td>
<td>678.0</td>
</tr>
<tr>
<td>0.4</td>
<td>656.7</td>
</tr>
<tr>
<td>0.5</td>
<td>709.0</td>
</tr>
</tbody>
</table>

The displacement (x-y axis) contour plot for the bearing without misalignment and with angular misalignment of 0.1°, subjected to two rotational speeds (157 and 300 rad/s) is shown in **Fig. 9.** Just like the contour plot notation in the stress and temperature distribution, the
regions of the component with the blue colour notation experience the least displacement, followed by the light blue, green, yellow and red. In general, the regions on the plot with blue notation experience the least displacement while the regions with red notation experience the highest displacement. From the plots in Fig. 9, it was observed that the displacement of the bearing components increases with an increase in the operating speed such that the maximum displacement in the bearing with 0.1° misalignment with a value of 9.488 μm was obtained when the rotational speed is 300 rad/s while the least displacement was with a value of 2.551 μm was obtained when the bearing’s rotational speed was 157 rad/s. For rotational speeds 157 and 300 rad/s, Table 5 summarizes the displacement experienced in the bearing when subjected to various angular misalignments. From the table, it was observed that the displacements in the components of the bearing increase with an increase in the degree of misalignment and operating/rotating speed. Furthermore, the maximum displacement was obtained in the inner ring of the bearing assembly while the cage (the component of the bearing with the least interference during the operating process) had the least displacement.

Fig. 9. Displacement contour plots and results of bearing with no misalignment, and with 0.1° misalignment for the different rotational speeds
<table>
<thead>
<tr>
<th>Angular misalignment (°)</th>
<th>Displacement (µm)</th>
<th>157 rad/s</th>
<th>300 rad/s</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.1</td>
<td>2.551</td>
<td>4.870</td>
<td></td>
</tr>
<tr>
<td>0.2</td>
<td>2.552</td>
<td>4.874</td>
<td></td>
</tr>
<tr>
<td>0.3</td>
<td>2.554</td>
<td>4.877</td>
<td></td>
</tr>
<tr>
<td>0.4</td>
<td>2.556</td>
<td>4.880</td>
<td></td>
</tr>
<tr>
<td>0.5</td>
<td>2.557</td>
<td>4.883</td>
<td></td>
</tr>
</tbody>
</table>

### 4.1 Analytical validation

In order to give credence to the developed model, it is paramount to validate experimentally or analytically the simulated results. The simulated Hertzian stress developed by the bearing when operated at rotational speeds of 157 and 300 rad/s, was analytically validated using Equations 10-14. The outcome of the validation shows that there is a good agreement between the analytical and simulated results since the percentage deviation is within the acceptable range (10 %). Table 6 shows the comparison and percentage deviation between the simulated and analytically computed Hertzian stress developed in the bearing during operation.

<table>
<thead>
<tr>
<th>Operational speed (rad/s)</th>
<th>Analytically computed Hertzian stress (MPa)</th>
<th>Simulated Hertzian stress (MPa)</th>
<th>Deviation (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>157</td>
<td>654.9</td>
<td>680.8</td>
<td>3.96</td>
</tr>
<tr>
<td>300</td>
<td>863.9</td>
<td>891.6</td>
<td>3.20</td>
</tr>
</tbody>
</table>

### 5 Conclusion

The dynamic behaviour of a cylindrical roller bearing subjected to two rotational speeds similar to those used in the airflow root blower of power generation plants, and five angular degrees of misalignment were simulated using Abaqus software. The outcome of the simulation showed that the developed temperature, Hertzian stress and displacement (x-y axis) of the components of the bearing increase with an increase in the operational speed and degree of angular misalignment. Furthermore, the maximum developed Hertzian stress in the bearing for the two operational/rotational speeds and five various degrees of angular misalignments was located on the outer ring of the bearing, thus making the outer ring the most prone component to failure during operation. Finally, for the two considered operating speeds, the obtained results from FE analysis were in good agreement with analytical calculations.

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