Thermal Elastohydrodynamic Lubrication Analysis of High-Speed and Light-Load Rolling Bearing with Double Rings Rotation

Junning Li*, Ka HAN, Qian WANG, Wuge CHEN, Jiafan XUE

Abstract: In order to reveal the lubrication performance of the rolling bearing under the condition of double rings s rotation, a novel rolling bearing contact lubrication analysis model for double rings rotation with an improved load distribution is developed. The effects of different parameters, such as the load, slip and speed, on the distribution of the temperature, pressure and film thickness of the rolling bearing are studied. The results show that the change rate of speed in double rings homonymous rotation is greater than that for single ring rotation and double rings reverse rotation on the inner and outer rings. Thus, the temperature change is higher for the former rotation than the latter two rotations. The oil film thickness for double rings reverse rotation is less than that for the other two types of rotations. The pressure distribution for the three types of rotations is more complex than the behaviour of the contact velocity and film. For variable speed, the temperature of the reverse rotation increases with the inner ring speed, which decreases as the outer ring speed increases. The temperature increases with the inner ring speed, which increases with the outer ring speed. The temperature of the double rings homonymous rotation or the reverse rotation increases with the load and slip. This manuscript can serve as a reference for research on the life and reliability of helicopter engine bearings under various rotation conditions.

Keywords: double rings rotation; high speed; light load; lubrication performance; rolling bearing; slip

1 INTRODUCTION

A rolling bearing is one of the most widely used components in rotating machinery, whose running state directly affects the accuracy, reliability and service life of the whole machine [1]. In helicopter, to improve the energy efficiency and compactness of the engine structures, a dual-rotor system has been widely employed. Frictional heating in high-speed bearings has been one of the primary factors in the failure of the main shaft bearing in helicopter engine [2-3]. Modern aero-engine spindle shafts mostly adopt a double-rotor structure, which forms a coupling between the low-pressure rotor and the high-pressure rotor, and the bearings connecting the low-high pressure rotor are the intermediate bearings [4]. In normal operation, the inner and outer rings of the inter-shaft bearing run at the same time. The engine structure makes it extremely difficult to lubricate the inter-shaft bearing. Therefore, it is of great significance to analyse the lubrication of the inter-shaft bearing. An integral calculation method for the frictional heat generation of rolling bearings is presented by Palmgren [5], which is suitable for low-speed, light-load and good lubrication conditions. Gao [6] proposes a new theoretical method to investigate the thermal behaviours of the inter-shaft bearing considering the nonlinear dynamic characteristics of a dual-rotor system by combining heat transfer and nonlinear dynamics. Wang [7-8] analyses the dynamic characteristics of a dual-rotor system with an inter-shaft bearing and performed a theoretical analysis and an experimental study on the vibrational response characteristics of a dual-rotor with imbalance-misalignment coupling faults. In order to reduce cross-excitation between the low pressure and the high-pressure shafts of two spool rotor, Gupta [9] proposes some designs of inter-shaft bearing with squeeze film. Jiang [10] considers the variable speed of dual rotors, as well as the weak signal, and uses linear prediction, spectral kurtosis, and order tracking in the rotational speed difference domain to develop a fault feature extraction scheme for the weak fault signals of inter-shaft bearings. Tian [11] accurately describes the dynamic features of inter-shaft bearings with localized defects in operation. Guskov [12] uses a rather fine numerical model to predict experimentally observed phenomena for a dual rotor test rig. The thermal elastohydrodynamic lubrication theory can be used to analyse the minimum oil film thickness, pressure distribution and local oil film temperature distribution in the key contact pair of bearings by including an energy equation and upper and lower boundary temperature conditions based on isothermal elastic lubrication. The lubricating oil film can separate the rolling raceway from the contact raceway. The viscosity of the lubricant causes the oil film to be subjected to viscous shear forces and the extrusion pressure of the fluid, thereby generating heat and increasing the temperature of the oil film. As a result, heat is transferred from the high-temperature environment to the bearing, increasing the temperature. R. Kumar [13] reviews the experimental methods that have been applied to date for measuring oil film thickness and pressure. The lubricating oil is regarded by Doki-Thonon [14-15] as a non-Newtonian fluid, and thermal elastohydrodynamic lubrication is applied to the thermal analysis of a rolling bearing. Huang [16] performs a comparative analysis of the line contact multigrid method and the Newton iteration method. The results show that the multigrid method is convergent and fast. Popinceau [17] correlates the fatigue life of rolling bearings using the trends indicated by the EHL theory and tests a large number of identical ball bearings over a range of rolling speeds, using various mineral oils and greases as lubricants. Corrections to the classical equation for the prediction of rolling bearing life are proposed.

In addition to the studies discussed above, researchers have achieved remarkable advances in the field of rolling bearings; however, most of these theories did not include distribution laws for the temperature rise, pressure and oil film thickness in the critical contact area under different rotation conditions for double rings rotation in helicopter engine bearing. Slip has rarely been included in analysing the temperature rise distribution in condition of double rings rotation. Therefore, in this manuscript, we compared and analysed the lubrication performance of key contact pairs of rolling bearings under different loads, slip rates, rotation speeds and rotation states by introducing different motion parameters and using the thermal
elastohydrodynamic calculation program for different double rings rotation states.

2 MATHEMATICAL MODEL
2.1 Kinematics Analysis Model

To determine the rotational speed of a rolling element and a cage considering load conditions, the following assumptions need to be adopted before deducing the simple kinematics of the rolling bearing: 1) The bearing element is a rigid body, and the influence of contact deformation is neglected; 2) The effect of radial clearance is neglected; 3) The rolling element purely on the ring raceway without sliding; and 4) The action of the oil film is neglected [18].

The case of a cylindrical roller bearing is used as an example to derive the angular velocity of the bearing cage, the rotational angular velocity of the rolling element and the average velocity. The simple motion relationship of a rolling bearing is shown in Fig. 1. The clockwise rotation direction is considered to be positive, and the counterclockwise rotation direction is considered to be negative.

1) The theoretical angular velocity of the cage
The inner raceway B has a velocity of:

\[ v_B = \frac{\pi n_i R_i}{30} \]  

(1)

Consider point A to be the instantaneous center of velocity and that the velocity of the roller point B is:

\[ v_B = 2\omega_c R_3 \]  

(2)

In the formula given below, \( R_1 \) is the inner ring raceway radius (m), \( R_3 \) is the roller radius (m), \( n_i \) is the inner ring speed (r/min), and \( (1) \) is the angular velocity of the roller relative to point A (rad/s).

\[ \omega_c = \frac{\pi n_i R_1}{60R_3} \]  

(3)

The point speed \( v_{O1} \) of the roller center \( O1 \) is:

\[ v_{O1} = \omega_c R_3 = \omega_c (R_1 + R_3) \]  

(4)

\[ \omega_c = \frac{\pi n_i R_1}{60(R_1 + R_3)} = \frac{\pi n_i}{60} (1 - \frac{D_r}{D_m}) \]  

(5)

In the formula, \( \omega_c \) is the angular velocity of the cage (rad/s), \( D_r \) is the diameter of the roller (m), and \( D_m \) is the diameter of the bearing pitch circle (m).

2) The theoretical rotational angular velocity of a rolling element
When the center of the roll is fixed and the speed of the roller \( n_r \) is zero, the retainer speed relative to that of the inner ring \( n_{ci} \) is:

\[ n_{ci} = n_c - n_i \]  

(6)

3) The average speed at the contact point between the roller and the ring raceway
The average velocity of the contact point can be expressed as:

\[ \omega_c = \frac{n_r D_m \left[ 1 - \left( \frac{D_r}{D_m} \right)^2 \right]}{60D_r} \]  

(7)

Similarly, under pure rolling conditions, the rotational speed of the cage and the rolling element under various rotating conditions of the two rings in rolling bearing can be obtained, as shown in Tab. 1.

Previously, frictional thermal analysis of a rolling bearing is mainly carried out when the outer ring is held fixed and the inner ring is rotated. In this manuscript, the theory of single ring rotation and analytical methods from the reference [19] are used to develop a frictional thermal analysis model of a rolling bearing with a slip factor for two rings rotating simultaneously. Using the case of a NU210 roller bearing as an example, the temperature rise curves of each layer on the contact surface between the roller and inner ring are calculated using the program when double rings rotate simultaneously.
Table 1: Rotational speed of the cage and the rolling element under various rotating conditions

<table>
<thead>
<tr>
<th>Rotating conditions</th>
<th>Schematic diagram</th>
<th>Cage speed ( n_c / \text{r/min} )</th>
<th>Roll element speed ( n_r / \text{r/min} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inner ring rotation</td>
<td><img src="image1" alt="Image" /></td>
<td>( n_i K_1 )</td>
<td>(- n_i \times \frac{2 D_m K_1 K_2}{D_r} )</td>
</tr>
<tr>
<td>Double rings homonymous rotation</td>
<td><img src="image2" alt="Image" /></td>
<td>( n_i K_1 + n_s K_2 )</td>
<td>((n_i - n_s) \times \frac{2 D_m K_1 K_2}{D_r} )</td>
</tr>
<tr>
<td>Double rings reverse rotation</td>
<td><img src="image3" alt="Image" /></td>
<td>( n_i K_1 - n_s K_2 )</td>
<td>(- (n_i - n_s) \times \frac{2 D_m K_1 K_2}{D_r} )</td>
</tr>
</tbody>
</table>

The load distribution is calculated using the modified load calculation method, and the resulting value for the maximum contact load is used as an input into the thermal elastohydrodynamic calculation program as a "contact load" to calculate the temperature and the oil film pressure distribution.

### 2.2 Thermal Elastohydrodynamic Lubrication Model

1) Governing equation for temperature

The energy equation [20] for a line contact thermal elastohydrodynamic lubrication problem may be written as follows:

\[
\rho c_p u \frac{\partial T}{\partial x} + \frac{T}{\rho c_T} \left( \frac{\partial p}{\partial x} \right) - k \frac{\partial^2 T}{\partial z^2} = \eta \left( \frac{\partial u}{\partial z} \right)^2
\]  

(12)


2) Reynolds equation

\[
\frac{d}{dx} \left( \frac{\rho h^3 dp}{\eta dx} \right) = 12U \frac{d(\rho h)}{dx}
\]  

(14)

3) Film thickness equation

\[
h(x) = h_0 + \frac{x^2}{2R} - \frac{2}{\pi E} \int_{x_0}^{x} p(s) \ln(s-x) \, ds + c
\]  

(15)

After dimensional normalization, the above equation becomes:

\[
H(X) = H_0 + \frac{X^2}{2} - \frac{1}{\pi} \int_{x_0}^{x} \ln |X-X'| \, p(X') \, dX'
\]  

(16)

\( H_0 \) is an infinite number of unknowns whose value is obtained in the solution process.

4) Viscosity-temperature equation

\[
\eta = \eta_0 \exp \left\{ (\ln \eta_0 + 9.67)[(1+p/p_0)^{\frac{T-138}{T_0-138}} - 1] \right\}
\]  

(17)

5) Pressure-temperature equation

\[
\rho = \rho_0 \left( 1 + \frac{0.6 \rho}{1+1.7 \rho} + D(T-T_0) \right)
\]  

(18)

6) Load equation

\[
\int_{x_{in}}^{x_{out}} \rho(x) dx = W
\]  

(19)

Let the load \( W \) applied per unit length be a constant, and dimensional normalizing of the load Eq. (19) yields:

\[
\int_{x_{in}}^{x_{out}} P(X) dX = \frac{\pi}{2}
\]  

(20)
2.3 Average Velocities of Double Rings and Single Ring

The basic parameters for the roller bearing are shown in Tab. 2. The initial value of the dimensional normalization coordinates is taken to be $X_{in} = -4$, and the final value of the dimensional normalization coordinates is taken to be $X_{out} = 1.4$.

Using the operating condition parameters given in Tab. 3, the film thickness is divided into five layers, and the number of nodes is 258. Using the modified load calculation method, the maximum load for a single roller under a radial load of 2000 N is 1053.34 N, so the maximum unit load for the rolling element is 81026.15.

<table>
<thead>
<tr>
<th>Roller number</th>
<th>Roller diameter / mm</th>
<th>Roller length / mm</th>
<th>Inner ring diameter / mm</th>
<th>Outer ring diameter / mm</th>
<th>Pitch diameter / mm</th>
<th>Bearing width / mm</th>
<th>Cage diameter / mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>16</td>
<td>13</td>
<td>13</td>
<td>50</td>
<td>90</td>
<td>70</td>
<td>20</td>
<td>75</td>
</tr>
</tbody>
</table>

Table 2 Basic parameters of a NU210 cylindrical roller bearing

<table>
<thead>
<tr>
<th>Dynamic viscosity of lubricating oil / Pa·s</th>
<th>Comprehensive modulus of elasticity / Pa</th>
<th>Slip factor</th>
<th>Radial load / N</th>
<th>Inner ring speed / r/min</th>
<th>Outer ring speed / r/min</th>
<th>Initial temperature / °C</th>
<th>Inlet oil temperature / °C</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.015</td>
<td>2.25E11</td>
<td>0.2</td>
<td>2000</td>
<td>12000</td>
<td>10000</td>
<td>40</td>
<td>40</td>
</tr>
</tbody>
</table>

Table 3 Operating condition parameters of a NU210 cylindrical roller bearing

Combining with Tab. 1 and Eq. (11), the average velocity of the contact area for the double rings homonymous rotation, double rings reverse rotation and single ring rotation can be calculated and obtained.

2.4 Numerical Scheme

The method for calculating the load distribution is modified to obtain the maximum unit load in the contact area, and the average velocity in the contact area is obtained by kinematics analysis.

The slip rate and the lubricant dynamic viscosity are taken into account to calculate the film thickness, density and viscosity. A new generation of equations is then substituted into the Reynolds equation for the stress distribution using the pressure value from the previous iteration, and the energy equation is then solved for the temperature distribution, using the newly generated temperature and viscosity. A new iteration is performed to obtain the pressure again and this process is iterated until there is relative error convergence between the last two iterated pressure values. Finally, the following results are obtained for the bearing zone: the inner ring raceway roller contact pressure distribution, the film thickness distribution and the temperature distribution. A detailed flowchart of numerical analysis is shown in Fig. 2.

3 RESULTS AND DISCUSSION

In the following figures, the parameters on both abscissas are dimensionless. Fig. 3 and Fig. 4 indicate the distribution of the temperature rise in each layer of contact line thermal elastohydrodynamic flow. It can be seen that the temperature distribution does not change much for $X \leq -1$. When $X > -1$, the temperature of each layer first increases then decreases, and then reaches a plateau for $X > 1$. As the oil film thickness decreases, the pressure increases sharply. Under the action of the shear extrusion force, the oil film thickness increases sharply, resulting in a sharp rise in temperature.
Temperature rise distribution in each layer for reverse rotation

The temperature of the outlet area is higher than that of the inlet area because the thickness of the oil film in the outlet area is thinner than that in the inlet area. Thus, the fluid in the outlet area is subjected to a greater shear force than that in the inlet area, resulting in higher frictional heat. Second, the lubricant in the inlet area is cold oil, whereas the lubricant in the outlet area is hot oil.

Fig. 4 and Fig. 5 show the distribution of the film thickness and pressure for reverse rotation. The film thickness distribution decreases as the contact pressure increases, and the minimum film thickness appears between $X = -1~1$. The oil film shape for the rest of the contact area is roughly parallel.

3.1 Lubrication Characteristics under Different Rotation Conditions

Fig. 7 and Fig. 8 correspond to a slip rate of 0.2. The distributions of the temperature rise and the film thickness for the middle layer are compared in the same direction for single ring rotation and homonymous and reverse rotation for the double rings. The relative velocity of the double rings homonymous rotation is larger than that for the single ring rotation, thus, the temperature rise is higher and the oil film is thicker for the former rotation than for the latter rotation. The relative velocity of the double rings reverse rotation is lower than that of the single ring rotation, thus, the temperature rise is lower and the oil film is thinner for the former rotation compared to the latter rotation.

Fig. 7 and Fig. 8 show the distribution of the film thickness and pressure for homonymous rotation. The pressure distribution is relatively complex under this operating condition because the relatively large rotational speed at the contact point brings in more oil, which increases the pressure in the inlet area, followed by a section where the pressure grows in a stable manner. After the accumulation of the oil film briefly increases the pressure at the outlet, the pressure distribution decreases sharply due to the quick release speed.
3.2 Temperature Rise Distribution under Different Slip Factors

Fig. 10 and Fig. 11 compare the temperature rise distributions for two rotations with slip factors of $s = 0.1, 0.3, 0.5$ and 0.7. The radial load is $F_r = 2000$ N. The inner ring speed $n = 10000$ r/min. The oil temperature of the oil film increases with the increase in the different slip factors, and the temperature rise for the double rings rotation is higher than that for the reverse rotation.

3.3 Temperature Rise Distribution under Different Rotational Speeds

In Fig. 12 and Fig. 13, the inner ring speed of the rolling bearing is 12000 r/min, and the outer ring speed is $-10000$ r/min. The change in the temperature rise is shown for the condition where the outer and inner rings have invariant speeds for double rings homonymous and reverse rotation. For the case of double rings reverse rotation, the temperature increases with the inner ring speed and decreases as the outer ring speed increases. For double rings homonymous rotation, the temperature rise increases with the inner ring speed and with the outer ring speed.

3.4 Temperature Rise Distribution under Different Loads

Fig. 14 and Fig. 15 compare the temperature rise distributions for a slip factor $s = 0.2$, radial loads of $F_r = 1000$ N, 2000 N, and 3000 N, and an inner ring speed $n = 10000$ r/min. The temperature of the oil film increases with the load, and the temperature rise for double rings homonymous rotation is higher than that for reverse rotation.
Figure 15: Temperature rise distributions for homonymous rotation under different loads

3.5 Model Verification

In reference [21], it is found that the shape of the pressure distribution curve changes obviously with the bearing speed at the contact point under rotation and gradually deviates from a Hertzian distribution. For small velocities, a secondary pressure peak almost disappears, and the position of the maximum pressure gradually moves towards the entrance area. The oil film thickness also increases and the ratio of the necking portion to the total contact width increases with the velocity. This phenomenon is basically in agreement with the trends for the oil film and pressure distribution that are reported in this paper, which verifies the rationality of the model.

4 CONCLUSIONS

This manuscript has focused on the contact lubrication performance of high-speed and light-load rolling bearings under different rotation conditions for double rings. The results show that the change in the contact velocity between the roller and the inner ring under homonymous rotation of the inner and outer rings is higher than that for reverse rotation, and the change in the temperature rise is more obvious for the former case than for the latter case. The oil film thickness is less than that of double rings homonymous rotation and single ring rotation. The temperature rise for double rings homonymous rotation or reverse rotation exhibits an upward trend with increasing loads and slip rates.

The research has significance for the reliability analysis of the double rings rotating rolling bearings and can serve as a reference for studying bearing-rotor system reliability and stability under the unique conditions of double-elements rotation, such as those that occur in helicopter engines.

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5 REFERENCES


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