Study on lubrication performance of heavy duty bearing of shearer based on fluid-solid coupling

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Abstract. The rolling bearing was an important part of the mining machine. Once the damage and maintenance was difficult, it will seriously affect the working efficiency of the mining machine. Based on the reliability analysis project of a coal mining machine, the lubrication performance of the output bearing of rocker arm was studied. Of thin seam coal winning machine developed by the project team get force curve of coal winning machine load simulator, based on the theory of Hertz contact mechanics study of cylindrical roller bearing contact state of low speed and heavy loading, on the basis of the contact behavior simulation, add the bearing oil film, through the establishment of bearing hydrodynamic lubrication theory and fluid-structure coupled model of oil film were analyzed, and the oil film thickness is obtained by analyzing the results of processing, the oil film pressure, film velocity and changes of the bearing parts contact. The fluid-solid coupling method is applied to the contact analysis of low-speed heavy-duty bearing, which provides a new method to study the contact problem of coal mining machine bearing.

1. PREFACE
The bearing, especially the heavy-duty bearing at the output end, is a key component of the shearer system. Its reliability is crucial to shearer’s efficiency. The heavy-duty bearing at the output end of the shearer bears a complex nonlinear load in working, and generates a large amount of heat, so its failure is difficult to handle and will seriously reduce the shearer’s working efficiency, thus affecting the working mining system as a whole. Therefore, it is of great practical significance to study the lubrication performance of the heavy-duty bearing at the output end. In recent years, scholars at home and abroad have been studying bearing lubrication. In 2008, Lin Tengjiao and Rong Qi conducted numerical analysis of bearing contact characteristics based on the thermo-mechanical coupling model. The analysis process is based on the established coupled model of the fluid heat flow in the inner cavity of rolling bearing. At the same time, the ANSYS-FLOTRAN module is used to systematically process the coupled flow field and temperature field. Division of the mesh is changed so as to observe its effect on bearing contact. V. Gerdun, T. Sedmak scholars, taking rolling bearings of railway carriage axle box as an example, introduced the mechanism of bearing fatigue and lubrication failure, thereby putting forward the multi-interface contact behavior and its requirements for contact performance. Liu Xiaoling, using Eyring shear fluid to establish the model of thermal shock rheologic elastohydrodynamic lubrication of ball bearings and multi-grid method and time-varying, rheological and temperature field analysis techniques, studied the effects of component temperature on the dynamic turbulent lubrication performance of dynamic oil film and thermal failure phenomenon in extreme conditions [1]. Lin Weibin analyzed the impact of lubrication and cleanliness on bearing life in
order to improve the life of rolling bearings and ensure the stable operation of machines \cite{2}. Peng
Chaolin analyzed the interaction mechanism between various failures and lubrication through the
study of the failure modes of rolling bearings; analyzed the failure mechanism of rolling bearings from
the bearing itself and grease; explained the influence of lubrication on the life of rolling bearings \cite{3}. 
However, many scholars studied the bearing lubrication performance based on ball bearings or simple
load-bearing conditions. This paper, by using virtual prototype technology to extract the dynamic
bearing load of the shearer, studies its lubrication performance, thus offering new methods and data
support to the research on heavy-duty bearings.

2. THEORETICAL BACKGROUND

2.1 Hertz Contact Theory of Cylindrical Roller Bearings

The kind of contact between cylindrical roller and the raceway is line contact. The length of the two
contacts must be equal in ideal line contacts, as shown in Figure 1.

Fig.1 Line contact half elliptic cylinder compressive stress distribution

In actual contact, $K$ is close to infinity. Stress distribution in the contact zone becomes a semi-
elliptical cylinder as shown in Figure 1. In this case:

$$ \sigma_{\text{max}} = \frac{2Q}{\pi d} $$  \hspace{1cm} (1.1)

$$ \sigma = \frac{2Q}{\pi d b} \left[ 1 - \left( \frac{y}{b} \right)^2 \right]^{1/2} $$  \hspace{1cm} (1.2)

$$ b = \left[ \frac{4Q}{\pi d^2 \sigma} \left( \frac{1 - \xi^2}{E_1} + \frac{1 - \xi^2}{E_2} \right) \right]^{1/2} $$  \hspace{1cm} (1.3)

$Q$ is the normal force between the rolling element and the raceway; $\sigma$ is the normal stress; $\sigma_{\text{max}}$ is
the maximum normal stress; $b$ is the half width of the contact surface; $\xi$ is Poisson's ratio; $E$ is elastic
modulus.

Lundberg and Sjovall proposed contact deformation under line contact conditions:

$$ \sigma = \frac{2Q(1 - \xi^2)}{\pi El} \ln \left[ \frac{\pi El^2}{Q(1 - \xi^2)(1 + r)} \right] $$  \hspace{1cm} (1.4)

This formula applies to rational line contact. In practice, the roller has a convexity. Therefore,
based on the experimental results of loaded roller with a convexity in the raceway, Palmgren proposed
the following formula for contact deformation:

$$ \sigma = 3.84 \times 10^{-5} \frac{Q^{0.9}}{l^{0.8}} $$  \hspace{1cm} (1.5)
However, Hertz made an explanation in his study of the contact area: When contacting, a small contact area must form to replace the line contact, thereby decomposing the load and avoiding excessive local stress.

2.2 Fluid Dynamics Lubrication Theory
To simplify the study of bearing movement, many calculations are carried out under the assumption that rigidity equals viscosity. But in actual operation, the friction pair is in a good fluid lubrication state. It is worth noting that under high load, both the elastic deformation of the friction pair and the increase in the viscosity of the lubricating oil have a great influence on lubrication. Fluid dynamics lubrication theory is the basis for studying lubricant membranes, while Reynolds equation is the basis of fluid dynamics lubrication theory\[^{4-6}\].

Under the condition of steady, isothermal, and incompressible Newtonian flow, Reynolds equation is:

\[
\frac{\partial}{\partial x} \left( h^3 \frac{\partial p}{\partial x} \right) = 6u_h \frac{\partial h}{\partial x} + 12 \frac{\partial h}{\partial t} \quad (1.6)
\]

where \( u_h \) is the fluid dynamic viscosity; \( h \) is the oil film thickness; \( P \) is the oil film pressure; \( u_h = u_1 + u_2 \) while \( u_1 \) and \( u_2 \) are the tangential velocity of the two contact faces respectively.

When the load is stable, oil film does not change with time, i.e. \( \frac{\partial h}{\partial t} = 0 \), the equation at this point is:

\[
\frac{\partial}{\partial x} \left( h^3 \frac{\partial p}{\partial x} \right) = 6u_h \frac{\partial h}{\partial x} \quad (1.7)
\]

3. Establishing Bearing Fluid-solid Coupling Model

3.1 Meshing
The three-dimensional model of the rolling bearing model is built using por/e. ABAQUS provides a variety of ways for meshing. What this paper applies is hexahedral structured meshing, which can handle more complicated models with more contact such as rolling bearings. An 8-node hexahedral de-integration unit (C3D8R) is used for meshing, as shown in Figure 2\[^{7-8}\].

Fig.2 Mesh partition

Figure 2 shows clearly the shape similar to rolling element interface on the Eulerian mesh surface. This deliberate meshing has two purposes: to make Eulerian mesh and roller shape adapt to each other and to render the initial material designation more convenient. The final nodes totaled 33560, units 23704, and Euler units (EC3D8R) 9072, of which 14632 are first-order hexahedral units (C3D8R).
3.2 Boundary Conditions
The outer ring of the bearing is fixed on the bearing housing so it does not rotate with the inner ring. For clearer observation of the boundary condition setting process, the Euler part in the model is hidden. Since the lubricating oil represented by the Euler body is difficult to flow out of the space between the outer wall of the inner ring and the inner wall of the outer ring due to the limitation of the various parts of bearing, the flow velocity of the Euler body on the two annular sides thereof should be limited. The specific boundary conditions are applied as shown in Table 1 and Figure 3.

| Inner ring | displacement /m | Outer appearance | displacement /m | circular side |
|------------|-----------------|-----------------|-----------------|--------------|
| UY         | 0               | UX              | 0               | VY           |
| RX         | 0               | UY              | 0               |              |
| RY         | 0               | UZ              | 0               |              |
| RX         | 0               |                  |                 |              |
| RY         | 0               |                  |                 |              |
| RZ         | 0               |                  |                 |              |

Fig. 3 Process of setting boundary conditions

3.3 Load
According to the real situation of the research object and research needs, the load of force and displacement load conditions of the model are set in this study. Concentrated force is applied to the reference point corresponding to the inner ring. This setting uniformly applies the concentrated force load to the inner surface of the inner ring of the bearing so as to avoid excessive local force caused by direct application of the concentrated force. Figure 4 shows load application, where the longer yellow arrow whose starting point is the origin of the coordinate represents the direction of the gravity load, while the shorter yellow arrow whose starting point is the reference point (RP) represents the direction of the concentrated force with the value of 17009 N.
3.4 Designating Initial Materials

The Euler body is used to represent the distribution of volume fraction and the state of motion of the fluid, including the lubricating oil. However, part of the space in the Euler body is initially occupied by the cylindrical roller and the cage, so it is specified that the area occupied by the cylindrical roller at the initial moment of calculation has no fluid material. This initial condition is specified in this article using the Predefined Field function. The material is designated as shown in Figure 5.

3.5 Model Verification

Simulate the established fluid-solid coupling model of the bearing; the distribution of pressure and thickness are shown in Figure 6.
When the rotational speed of the bearing is 8.714 rad/s, the distribution of oil film pressure and oil film thickness during lubrication is shown in Figure 6 and Figure 7.

1) The oil film pressure distribution curve is similar to the Hertzian pressure distribution curve, but is separated from it in the vicinity of the exit zone, and a slight deformation occurs. That is what we call the secondary pressure peak.

2) At the inlet of the bearing lubricating oil, the thickness of the oil film drops sharply due to heavy bearing load; when the oil film pressure distribution is abruptly separated from the corresponding position of the Hertz curve, the oil film thickness begins to decrease, and the corresponding oil film becomes thinner. In the exit area, the thickness of the oil film increases sharply.

Through the basic trend of thickness and pressure of oil film of bearing lubrication as shown in the figure, the simulation results are reasonable and correspond to the theory.

4. Analysis of Lubrication Performance of Bearing Contact Interface

4.1 Analysis of Contact Changes of Oil Film Thickness
Stimulate and analyze the thickness of contact oil film between the oil film and the inner and outer rings of the bearing under the working condition of the rotation speed of 8.714 rad/s. Results are shown in Figure 8.
Figure 8(a) shows the distribution of the lubricating oil film in the bearing. Red represents the lubricating oil, green the transition state, and blue the solid area. Although the applied external load is constant, each rolling element is subjected to different contact loads as the bearing operates. The rolling element with smaller contact load and that with larger contact load are selected to obtain the oil film thickness cloud map. As shown in Figures 8(b) and (c), on the rolling element with a smaller contact load, the minimum oil film thickness on the outer ring raceway is significantly larger than that of the inner ring. It is the same with the rolling element with larger contact load. Therefore, although the load on the inner and outer ring races is unevenly distributed, different rolling elements show the same trend of change in terms of the oil film thickness between the inner and outer races, that is, the oil film of outer ring contact race is thicker than that of the inner ring contact raceway. From the sectional view of the oil film thickness in Figure 8(d), it can be seen more clearly that the oil film thickness is distributed in a rectangular shape, and the rectangular area of the oil film thickness between the rolling element and the inner ring is smaller than that between the outer ring and the outer ring.

4.2 Oil Film Pressure Contact Change Analysis
Stimulate and analyze oil film pressure of the oil film and the inner and outer rings of the bearing under the working condition of the rotation speed of 8.714 rad/s. Results are shown in Figure 9.

(a) Overall oil film pressure

(b) Contact load less scroll of oil film pressure portion images

(c) Contact load larger oil film pressure roller
Figure 9 shows the distribution of oil film pressure. It can be seen that the oil film pressure at the inlet of the contact zone is increased. From the partial zoom-in view, it can be seen that the oil film pressure at the inner raceway contact is greater than that of the outer raceway. The sectional view shows that the oil film pressure is reduced in the contact area. This law of oil film pressure distribution occurs because a large amount of lubricating oil accumulates at the entrance of the rolling element and the inner and outer ring contact areas, which increases the oil film pressure. In the contact area, the speed of the lubricating oil is constant. The lubricating oil passing through the contact area is small and shows an outward tend, thus the oil film pressure is reduced.

4.3 Analysis of Contact Changes of Flow Velocity of Oil Film

(a) Overall images of oil film flow rate

(b) Contact load less scroll body oil film flow ges
Figure 10(a), the velocity cloud map of the oil film, shows that the speed of oil film in the intersection area between the fluid and the solid is significantly greater than any other areas, including the free surface. This is because movement of the lubricating oil caused by the movement of the inner ring of the bearing and the rolling elements; the oil film in the contact area is thinner, hence better entrainment effect and larger flow rate. As oil film gets thinner from the inlet to the outlet of the contact area, the lubricating oil moves toward this area; as it gets thinker after bypassing the contact area, most of the lubricating oil bypasses the contact area. Analysis of the less loaded part of the rolling element is shown in Figure 10(b), which shows that the velocity of the oil film at the inlet is greater than that at the outlet. Analysis of the less loaded part of the rolling element is shown in Figure 10(c), which also shows that the oil film velocity at the inlet is larger than that at the outlet. That is because as the oil film in the contact area gets thinner, more lubricating oil accumulates at the inlet and less at the outlet. Figure 10 (d) is a radial sectional view of the oil film.

5. Conclusion
This paper takes the bearing NJ2232 at the output end of the rocker arm of thin coal seam shearer as the research object to establish the bearing lubrication model. Based on contact theory and elastic fluid dynamics lubrication theory, this paper analyzes the lubrication effect of low-speed heavy-duty bearings. The conclusions are as follows:

(1) The fluid-solid coupling model of the bearing is established. Selected parameters of the bearing and bearing lubricant and contact conditions are applied to the simulation model, thus verifying the correctness of the fluid-solid coupling model.

(2) The lubrication performance of the contact interface of low-speed heavy-duty bearings is analyzed. Results show that the thickness of the oil film features rectangular distribution in the cross section; the rectangular area of the oil film thickness between the rolling element and the inner ring is smaller than that between the outer ring and the outer ring. The oil film pressure at the inlet of the contact zone increases, so the pressure at the inner rolling contact is greater than that at the outer raceway. The velocity of the oil film at the intersection part between the fluid and the solid is greater
than that of the free surface and is significantly greater than any other areas. In different working conditions, as the rotational speed increases, the thickness and pressure of oil film increase; as the load increases, oil film thickness decreases while oil film pressure increases.

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