Particulates Distribution in the Dynamic Pressure Sliding Bearing under the Oil Lubrication Condition

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Abstract: In the dynamic pressure sliding bearing under the oil lubrication condition, the lubrication oil between the journal and the bearing bush maintains the normal operation of the bearing, which is affected by the complete oil film, the boundary lubrication and the mixed film. With the increase of oil input and journal speed, it is not uniform towards the dynamic pressure distribution inside the sliding bearing. At the same time, cavitation is occurring, causing that particulates fall off the bearing surface, as well as the increase of the improper noises, which reduce the service life of the sliding bearing. The experimental result shows that the wear particles in the oil film are affected by the rotation speed of the shaft journal, the external load of the bearing and the size of the wear particles, so the wear particles in the bearing oil film are concentrated at the working face of the bearing bush and the minimum thickness of the oil film by centrifugal motion. Therefore, the study of the physical and chemical characteristics, which are talked on wear particles in the lubricating oil film of dynamic pressure sliding bearings, will play an important role in the fault diagnosis and condition monitoring of sliding bearings in the mechanical field.

1. Introduction
Under working conditions, there are some factors such as poor sealing, air erosion and complex working conditions in the sliding bearing, which lead to the presence of solid particles in the lubrication oil of the bearing. With the change of journal speed and load pressure, the solid particles suspended in the oil film have a great influence on the lubrication and mechanical properties of sliding bearing. LI[1] used LBM theory to study the influence of abrasive particles distribution suspended in lubricating oil on oil film pressure and oil flow velocity, and found that the more solid particles distributed in the direction of oil film thickness, the stronger the abrasive particle distribution was in the ability to obstruct the flow of lubricating oil. GAO[2] et al. used Archard modified model to conduct finite element analysis of bearing wear, and obtained the friction factors and coefficient of bearing steel under boundary lubrication by carrying out ball disc friction and wear test, and analyzed the influence of running time, radial load and other factors on bearing wear. Due to the presence of lubricating oil between sliding bearing journal and bearing bush friction pair, LIN[3] et al. proposed a bearing wear statistics method based on Archard wear model and adhesive wear mechanism. Polygonal or polyhedral particles can be used as a finite number of block elements. At present, it is not practical to simulate the large number of particle groups, so the dominant particles are spheres. [4-9] In order to better evaluate the mechanical properties and service life of oil-lubricated sliding bearing and reduce the mechanical damage caused by particle wear.
and adhesive wear, it is necessary to conduct in-depth analysis of wear particles of sliding bearing by combining Archard wear model and finite element analysis method.

2. Cavitation in sliding bearing

In the process of running at high speed, the sliding bearing will produce unstable dynamic pressure on the lubricating oil between the journal bearing bush. As oil lubrication bearing axis neck load at high speed, a small amount of mixed in lubricating oil by high pressure air escape form bubbles, the bubbles burst in journal - bearing clearance, and produce mechanical damage and chemical corrosion to the bearing metal surface. These phenomena have produced metal particles, which accelerate the sliding bearing friction and wear and aging damage. Thus, these actions are causing air cavitation.

Under the steady-state condition of dynamic pressure sliding bearing, with the rotation direction of the journal, the position of the minimum oil film thickness can be taken as the boundary line, and the oil film pressure in the wedge zone reaches the peak value gradually and then decreases rapidly in the divergent zone, forming the vacuum zone. By comparing the influence of four groups of different rotational speeds on the pressure area of sliding bearing, it can be seen that the position of the maximum positive pressure area basically remains unchanged and has nothing to do with the rotational speed of the journal. The maximum positive pressure value and the maximum negative pressure value have a linear relationship with the spindle speed, and the pressure value decreases with the increase of the spindle speed.

The temperature variation of friction surface of sliding bearing is calculated by finite element method, and the steady-state calculation results can be obtained by FLUENT software in version 19.2. In this paper, sliding bearing model parameters and cavitation conditions are selected, which have been shown in Table 1.

| Inside Diameter | Outer Diameter | Spindle Diameter | Eccentricity | Diameter of Oil Port |
|-----------------|----------------|------------------|-------------|---------------------|
| 980 mm          | 975 mm         | 979.9 mm         | 0.016 mm    | 4 mm                |
| Width           | Speed          | Viscosity        | Density     | Cavitation Pressure |
| 30 mm           | 65 rpm         | 0.048 Pa/s       | 890 Kg/m3   | 7550 Pa             |

The model has been imported into Fluent software for the calculation of steady-state multiphase flow. Considering the cavitation effect, the phase of lubricating oil is gasified to form cavitation in local surface under the influence of the constant speed of axial neck that is under the dynamic load. The inlet pressure of the oil port is set at 0.5MPa, and the rotating surface of the journal is set at low speed.

Considering heat transfer, the energy equation is opened and the boundary heat transfer conditions are set. Meanwhile it has been reasonable chosen to set the axis as the heat source, take heat transfer in the form of heat flux, set the heat flow 0 on the outer surface of the bearing bush, and set the coupling surface as the outer surface of the journal for heat transfer 650. Then it is coherent to calculate from the outer surface of the journal, and give a small initial speed of 0.005m/s in X and Z directions, and determine the initial volume fraction of air that can be set to 0.1. Finally, the software can calculate this bearing model and set these parameters to calculate for enough steps.

By comparing the distribution of the temperature inside the bearing at the above three different speeds, it can be obtained that the maximum temperature value of the bearing increases with the increase of the speed. It can also be seen from the figure that the area of the high temperature area gradually expands with the increase of the spindle speed, so that the high temperature area slowly approaches the oil port.

At the same time, when observing the temperature distribution of the wedge region and the divergence zone, it can be found that the temperature of the wedge region is significantly lower than that of the divergence zone, and with the increase of the spindle speed, the divergence zone temperature is more likely to increase than the divergence zone.
Therefore, it can be concluded that the influence of shaft speed on the temperature field of bearing oil film is mainly in the divergence zone, where the shaft neck is prone to thermal damage and frictional wear which is due to the formation of high temperature accumulation zone.[10]

![Fig.1 Air corrosion at 50rpm.](image1)

![Fig.2 Air corrosion at 80rpm.](image2)

By comparing the distribution of cavitation zone within the bearing under the above three different speeds, it can be summarized that with the increase of the journal speed, the cavitation zone generated on the working surface of the bearing expands continuously along the bearing circumference, and the corrosion degree of the central cavitation zone intensifies. According to Reynolds equation, the wedge area and divergence area of the sliding bearing under dynamic load provide conditions for cavitation to occur, as shown in FIG.1 FIG.2.

At the same time, the pressure and temperature distribution of the oil film is not uniform, and the center of the journal must believe that the increase of motion, leading to the lubrication film rupture. These actions are making the cavitation phenomenon in the bearing more serious, so that these make the bearing surface off, accelerate wear, increase noise, and reduce the service life of the bearing.[11]

3. Archard wear model

Due to the lubricating oil filling between the journal and bearing bush friction pair, the friction pair's wear is mainly adhesive WEAR. When adopting ARCHARD WEAR model, it is necessary to determine the WEAR factor of the friction pair and take into account such conditions as external load, working temperature and allowable clearance inside the shaft.

During the wear process of sliding bearing, the shape, size, interaction, physical and chemical characteristics of particles as well as other factors will be introduced into the model calculation, which are regarded as initial value conditions, and the wear condition and particles concentration distribution of sliding bearing can be simulated by EDEM software.

The mathematical expression of ARCHARD WEAR model is:

\[
dV = k \frac{d(P \times L)}{H}
\]  

(1)

Where, dV is wear volume differential; D (P) is the product differential of the normal pressure P and the tangential relative slip distance L between the journal and the bearing bush; K is the wear factor; H is Brinell hardness of bearing bush material.

At the same time, the wear volume between the journal and the bearing bush friction pair can be replaced by a cylinder, which has the following formula:

\[
dV = d(h \times S)
\]  

(2)

Where: H is the average wear depth between the upper surface of the bearing bush and the lower surface of the journal; S is the contact area between friction pairs. The high speed running of the journal under working conditions will lead to the increase of friction heat between the bearing friction pair, and the increase of the temperature of the friction pair members will make the metal on the journal and bearing bush surface softened by heat, thus reducing the metal surface hardness and increasing the roughness of the friction pair contact surface.

Therefore, there is no doubt that ARCHARD WEAR model needs to be modified. In view of this situation, Hahamir et al constructed K(T) and H (T) functions with H13 mold steel as the experimental
material and temperature as the independent variable, and proposed a more complete WEAR calculation formula:

\[ h(T) = K(T) \frac{\delta u}{H(T)} \, dt \]  

(3)

The K(T) and H(T) functions in the modified model can be characterized by empirical formulas fitted by experiments on high temperature wear and hardness.

In FIG. 3, A is the bearing bush, B is the solid particle, and C is the bearing journal. The solid particles move irregularly between the metal friction surfaces on both sides of the oil film. At the same time, under the influence of temperature, pressure, speed and other factors of the bearing, the friction pair will experience greater contact stress, resulting in fatigue wear or plastic deformation on the contact surface, which further weakens the mechanical properties and service life of the sliding bearing.[12]

The study on the wear mechanism of sliding bearing particles mainly focuses on four aspects, including namely micro-cutting, micro-fracture, extruding spalling and micro-fatigue. Moreover, the wear of bearing particles is affected by external load, relative hardness of materials, geometric characteristics of bearing particles and other factors. Through our study of sliding bearing case and our analysis of wear condition, the bearing has been influenced jointly by normal load, contact stress, furrow effect of particulate matter. At the same time, it is distinct to find that the damage is intensified because of the solid particles to the metal surface of the journal and bearing, including shear, plowing and cutting, thus causing irregular groove grinding marks in the sliding bearing. FIG. 4 has shown the abrasive wear of sliding bearing and verified the results discussed in this paper.

4. Analysis of particulates matter distribution

Through proper settings in the EDEM software, the full-period motion results can be calculated, which are about wear particles between journal and bearing bush from generation to quasi-steady state distribution. In order to ensure the precision and accuracy of the simulation results, rotational kinetic energy in the direction of z-axis can be applied to the journal surface, so that the rotational speed of the journal gradually increases from 0 rpm to 50 rpm. In addition, it is necessary to consider the Y-axis direction that can be set as -9.81 N/Kg in the direction of gravity acceleration of particulate matter. Secondly, FLUENT software is coupled with EDEM software to realize the comprehensive calculation of particle distribution in oil film of sliding bearing. The particles that have been caused by the friction and wear of bearing bush and journal are considered to move in the lubricating oil, affecting the stress distribution of oil film and journal.

In consideration of the effect of the Saftman effect and the pressure gradient force, the distribution of particulate matter in the oil film and the wall surface has been discussed under the condition that the shaft rotation speed is 50 rpm. As shown in FIG.5 and FIG.6.
In order to further analyze the effect of different rotating speeds on the stress at the bottom of the tail shaft, the stress data at three different rotating speeds of 50rpm, 70rpm and 90rpm were compared. It is critical to quantify the stress distribution at the bottom of the tail shaft, and the longitudinal (Z-axis) direction was selected to extract stress data. Then the calculated data was sorted out for data analysis in the Origin software, which was displayed through the three-dimensional waterfall diagram of Original software, as shown in Fig. 7. It can be seen from the figure that the shape of the two is roughly the same, indicating that the stress distribution of the tail shaft is roughly the same at low rotational speed, and the value of the two is obviously different. The maximum stress at the speed of 70rpm is 3596N, while the maximum stress at the speed of 50rpm is 5607N, indicating that the increase of speed will affect the mechanical properties of the friction pair between the journal and the bearing bush, leading to the decrease of stress between the friction pair. The tangential component of the contact force generated by the relative motion of the metal surface between the journal and the bearing friction pair leads to Coulomb friction. At the same time, the relative movement between the oil flow layer and the bearing metal due to different velocities leads to tangential contact force at the junction, thus forming fluid friction. The existence of Coulomb friction and fluid friction makes the surface material of oil-lubricated tail bearing friction pair peel off due to fatigue under cyclic contact stress, and the metal surface condition of the friction pair deteriorates continuously. Wear debris will be produced with the flow of lubricating oil with different degrees of displacement, and then under the action of load pressed into the friction surface to produce indentation, the plastic material surface extruded out irregular debris.
the lubricating oil velocity on the inner surface of the bearing bush does not change much under the influence of inertia, gravity and viscous force. The increase of spindle speed leads to the increase of centrifugal force on the particles, which reduces the average velocity of particles in the oil and concentrates on the inner surface of the bearing bush during the shaft neck operation.

5. Conclusion
(1) By combining the finite element analysis method and Archard Wear model, with the help of FLUENT and EDEM software, the ground generation and distribution process of particulate matter in sliding bearing can be accurately simulated.

(2) The simulation results show that the properties of metal materials, lubricating oil quality and bearing working state (including bearing speed, temperature, dynamic pressure and external load) can directly or indirectly affect the distribution of solid particles in the oil film of sliding bearing.

(3) Through the research and analysis in this paper, it can be summarized that in the failure mode of oil-lubricated sliding bearing, abrasive wear has a great impact on the mechanical properties and service life of the bearing.

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