Numerical Calculation of Composite Water Lubricated Bearing Considering Effect of Elastic Deformation

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Abstract. Traditional bearing design neglects the influence of elastic deformation of materials on bearing lubrication characteristics, based on a new type of polymer composite water-lubricated bearing materials, the influence of elastic deformation of bearing material is introduced into the simulation. The lubrication performance of the water-lubricated bearings is studied with respect to material parameters and structural parameters. The effects of the elastic deformation on the lubrication characteristics such as the maximum pressure value and the minimum film thickness were compared and analyzed under different bearing clearance structures of a certain type of stern bearings. The results show that, the maximum water film pressure of bearing increases with the clearance ratio, and the maximum water film pressure of elastic bearing is less than that of rigid bearing. With the increase of clearance ratio, the elastic deformation of bearing increases linearly. The increase of bearing elastic deformation results in the difference between minimum water film thickness and maximum water pressure between the elastic bearing and rigid bearing.

1. Introduction

As a critical component of the ship propulsion system, the friction and lubrication characteristics of stern tube bearings have an important impact on the service performance and life reliability of the ship. Due to the low-speed and heavy-load operating conditions of water-lubricated bearings combined with the low viscosity of the water medium, bearings usually run in mixed lubrication or boundary lubrication, which will inevitably lead to direct contact between the shaft and the bearing friction surface and wear of bearing material. Therefore, compared with other types of sliding bearings, water-lubricated bearings have requirements on the wear resistance of materials. In addition, the elastic modulus of water-lubricated bearing material is only a few hundredths of that of metal material, which makes the bearing elastic deformation and the water film thickness in the same order of magnitude. Hence, the influence of elastic deformation on bearing lubrication performance and structural design cannot be ignored.

At present, new water-lubricated bearing materials are mostly made of high molecular polymer [1]. The low hardness and small elastic modulus of high molecular polymer combined with the low speed and heavy load operating characteristics usually make high molecular polymer bearings generate significant elastic deformation [2-3] during operation. Simultaneously, due to the low viscosity of the water medium and poor dynamic pressure bearing capacity, such bearings are more likely to run in
mixed lubrication under low speed, heavy load and start-stop conditions, which results in severe wear of bearing material [4-6]. The matching scale has an important influence on the friction and wear performance of the friction pair [7-8]. The traditional bearing simulation method does not consider the influence of the elastic deformation of the bearing on the bearing friction and lubrication characteristics. The simulation analysis results have a large error compared with the engineering practice, which cannot meet the demand of the current water-lubricated bearing simulation analysis. Therefore, based on the operation characteristics of water-lubricated high molecular polymer bearings, it is necessary to introduce the elastic deformation simulation method into the water-lubricated bearing simulation model to establish a suitable numerical analysis method for the calculation of the lubrication characteristics of ship water-lubricated bearings.

In this paper, based on the modified UHMWPE bearing material parameters [9-10], the related lubrication theory of water-lubricated bearings is analyzed firstly. The boundary conditions and the flow state of the lubricating medium in the numerical calculation process of water-lubricated bearings are studied. The bearing lubrication performance analysis method considering elastic deformation is established. And considering the elastic modulus of the polymer composite material, the water-lubricated bearing pressure, minimum film thickness, deformation, friction coefficient and eccentricity under different clearances are calculated. Finally, the calculation results are compared and analyzed with the rigid bearing.

2. Water-lubricated bearing model considering elastic deformation

2.1. Calculation method

For the water-lubricated bearings, the principle of the dynamic pressure formation is shown in Figure 1. Under the effect of the external load, the certain eccentric distance $e$ will be generated between the center of the rotating shaft $(O_j)$ and the center of the bearing $(O_b)$, thereby forming a wedge-shaped gap in the water film convergence region, while the rotating shaft drives the water medium with a certain viscosity relative to the bearing. The movement meets three conditions of fluid dynamic pressure lubrication formation, hence establishing dynamic pressure lubrication and supporting the external load.

![Figure 1. Principle of Bearing Dynamic Pressure Formation](image)

The fluid particle motion equation is expressed as:
\[
\begin{align*}
\rho \left( \frac{\partial v_x}{\partial t} + v_x \nabla v_x \right) &= \rho X + \frac{\partial \tau_{sx}}{\partial x} + \frac{\partial \tau_{sx}}{\partial y} + \frac{\partial \tau_{sx}}{\partial z} \\
\rho \left( \frac{\partial v_y}{\partial t} + v_y \nabla v_y \right) &= \rho Y + \frac{\partial \tau_{sy}}{\partial x} + \frac{\partial \tau_{sy}}{\partial y} + \frac{\partial \tau_{sy}}{\partial z} \\
\rho \left( \frac{\partial v_z}{\partial t} + v_z \nabla v_z \right) &= \rho Z + \frac{\partial \tau_{sz}}{\partial x} + \frac{\partial \tau_{sz}}{\partial y} + \frac{\partial \sigma_z}{\partial z}
\end{align*}
\]  

(1)

The mass conservation equation:

\[
\frac{\partial \rho}{\partial t} + \frac{\partial (\rho v_x)}{\partial x} + \frac{\partial (\rho v_y)}{\partial y} + \frac{\partial (\rho v_z)}{\partial z} = 0
\]

(2)

**Figure 2.** Water Film Boundary Conditions

2.2. Model design and analysis

The water film boundary conditions are set as shown in Figure 2. The left and right end faces of the water film are the pressure inlet and the pressure outlet respectively. The outer surface of the water film is in contact with the bearing, which is a fluid-solid coupling surface. The inner surface of the water film is in contact with the shaft, which is a rotating wall surface. Due to the special structure of the water film, the radial dimension is several tens of micrometers or even several micrometers, which is much smaller than the axial and circumferential dimensions. In order to accelerate the convergence of the calculation results and increase the stability of the calculation process, the structured grid is used for the fluid in this calculation. The solid mesh is divided in the ANSYS Workbench, and the material parameters and boundary conditions are set at the same time, finally the *.inp file containing the solid domain information is output. The fluid grid and the *.inp file of the solid are imported into CFX for fluid-solid coupling calculations.

The bearing solid elements are deformed and displaced under the effect of the water film pressure, and the motion follows the Newton's Second law.

\[
\rho_i \ddot{d}_s = \nabla \cdot \sigma_s + f_i
\]

(3)

Where \(\rho_i\) is the solid density, \(\ddot{d}_s\) represents the acceleration of the solid element, \(\sigma_s\) is the solid stress tensor and \(f_i\) is the volume force vector. At the interface between the fluid and the solid, the fluid pressure deforms the solid, which changes the boundary shape of the fluid in turn. The elements at the interface between the fluid and the solid follows the equality of variables such as displacement, stress, heat flow and temperature. This paper neglects the temperature variation of the water medium in the calculation,
so the change in heat flow and temperature is ignored. Therefore, the equilibrium equations are expressed as:

\[
\begin{align*}
\mathbf{d}_f - \mathbf{d}_s &= \mathbf{0} \\
\mathbf{n} \cdot \mathbf{\tau}_f - \mathbf{n} \cdot \mathbf{\tau}_s &= \mathbf{0}
\end{align*}
\]

(4)

Where \(d_f\) and \(d_s\) are fluid displacement and solid displacement respectively, \(\mathbf{\tau}_f\) and \(\mathbf{\tau}_s\) are the stress of fluid particles and solid elements respectively. The governing equations of the fluid and solid domains are coupled to the same system matrix for solution:

\[
\begin{bmatrix}
\mathbf{A}_{ff} & \mathbf{A}_{fs} \\
\mathbf{A}_{sf} & \mathbf{A}_{ss}
\end{bmatrix}
\begin{bmatrix}
\Delta \mathbf{X}_f^k \\
\Delta \mathbf{X}_s^k
\end{bmatrix} =
\begin{bmatrix}
\mathbf{F}_f^k \\
\mathbf{F}_s^k
\end{bmatrix}
\]

(5)

Where \(\mathbf{A}_{ff}\) and \(\mathbf{A}_{ss}\) are system matrices of the fluid and solid domains respectively, \(\mathbf{A}_{fs}\) and \(\mathbf{A}_{sf}\) are fluid-solid coupling matrices, \(\Delta \mathbf{X}_f^k\) and \(\Delta \mathbf{X}_s^k\) are the pending solutions of the fluid and solid domains respectively, \(\mathbf{F}_f^k\) and \(\mathbf{F}_s^k\) are the external loads acting on the fluid and solid domains respectively, \(k\) is the number of iteration steps.

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3. Results analyze

The clearance has an important influence on the bearing carrying capacity and friction coefficient. Therefore, it is important to study the effect of different clearances on the bearing load and lubrication performance. According to the compression test, the modified UHMWPE bearing material has the compressive elastic modulus of 1000MPa and the Poisson's ratio of 0.44, which are used as the material parameters when studying the influence of structural parameters on the lubrication performance of the bearing. In Figure 3(a)-(e), the water-lubricated stern bearing with the diameter of 80mm and the length-diameter ratio of 1.0 was taken as the research object to study the influence of different clearance ratios on maximum water film pressure, minimum water film thickness, deformation, friction coefficient and eccentricity with and without elastic deformation.

(a) Maximum Water Film Pressure Comparison

(b) Minimum Water Film Thickness Comparison
Figure 3. Effect of Different Clearance Ratio on Lubrication Performance of Bearings

4. Conclusion
The results show that when the bearing clearance ratio increases from 0.06% to 0.25%, the bearing eccentricity increases from 0.4 to 0.95 under constant load and speed conditions, which indicates that the bearing carrying capacity decreases and the change of the eccentricity gradually slows down with the increase of the clearance ratio, and the influence of the clearance on the bearing carrying capacity is gradually reduced at the same time. The maximum water film pressure increases with the increase of the clearance ratio. The maximum water film pressure of the elastic bearing is smaller than that of the rigid bearing, and the difference between them increases with the increase of the clearance ratio. In addition, the minimum water film thickness and friction coefficient reduce with the increase of the clearance ratio. The minimum water film thickness of the elastic bearing is larger than that of the rigid bearing, and the difference between them increases with the increase of the clearance ratio. Simultaneously, with the increase of the clearance ratio, the bearing elastic deformation increases linearly. After the bearing clearance ratio increases from 0.06% to 0.25%, the bearing elastic deformation increases by 3.5 times. It is also because of the increase of the bearing elastic deformation that the difference between the minimum water film thickness and the maximum water film pressure of the elastic bearing and the rigid bearing gets larger.

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