Analysis of the Tribological and Dynamic Performance of the Self-Adapting Water-Lubricated Stern Bearing

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Abstract: According to the design requirements of load equalization and vibration reduction for the stern bearing, a water-lubricated stern bearing with self-adaptation capacity is proposed. The bearing is mainly composed of three parts: the bearing bush, the elastic element, and the damping alloy. The elastic element is used to realize static and dynamic load sharing of the stern bearing, reduce the edge effect of the stern bearing, and make the contact pressure evenly distributed in the axial direction, thereby improving the service life of the bearing and reducing the frictional excitation of the bearing. Damping alloy is used to attenuate the shaft vibration transmission from the bearing to the foundation to optimize the vibration transmission characteristics. The revised lubrication models for such bearings are put forward. By analyzing the vibration characteristics of the stern bearing, the results show that the vibration transmission characteristics of the thruster excited to the bearing node are optimized, and the vibration response at the first-order fixed frequency is significantly reduced. A moderate increase in the support stiffness of the foundation can significantly reduce the vibration response of the bearing.

Keywords: self-adaptation; water lubrication; stern bearing; vibration characteristics

1. Introduction

Propulsion shafting is the rotating equipment with the largest weight and longest length at the stern of the ship. The shafting vibration caused by various excitations of the propeller and main shaft is transmitted to the stern hull through the bearing and base and transmitted to the water to form radiated noise. The stern bearing of the propulsion shaft system is the main way for the propeller excitation force to transmit to the hull vibration through the shaft system. Its main function is to support and ensure the safe operation of the shaft system under a long longitudinal span. Therefore, the stern bearing is a key piece of equipment that affects the acoustic performance of the propulsion system and even the sound radiation of the hull stern [1].

Under the cantilever load of the propeller, the bearing will have an “edge load” effect which leads to the uneven distribution of the bearing load. That means the load is mainly concentrated at the stern of the bearing, and the bow load is very small or even empty. The resulting effects include: (a) Local overload accelerates bearing wear and reduces bearing service life; (b) Eccentric wear of the bearing causes misalignment of the bearing and affects shafting vibration; (c) Local overload makes rubber bearings prone to friction noise [2–6]. In order to solve this problem, it is necessary to improve the uniformity of load distribution in the bearing through bearing load sharing technology. Generally, based on the static alignment calculation results of the shafting, measures of tilting the bearing are taken to make the bearing comply with the bending changes of the shaft end to realize a uniform
load [7–9]. However, during the actual operation of ship shafting, the shafting centering state is constantly changed by the influence of dynamic factors such as hull deformation, water temperature change, different loading capacity, and speed change. Based on the static alignment theory, the method of inclined bearing fails to consider the dynamic effect, and its load-sharing effect is limited. Therefore, how to improve the support interface characteristics of propeller bearing, reduce the adverse impact of edge load on bearing and shafting, and carry out vibration reduction on this basis is one of the important directions to improve the service life of bearing and optimize the vibration of propeller shafting.

According to the bearing state of the water-lubricated stern bearing, an adaptive water-lubricated bearing structure integrating static load sharing and dynamic vibration reduction is proposed in the present study. The static and dynamic load-sharing performance of the stern bearing can be effectively increased. The edge effect of the stern bearing is improved, and its contact pressure is evenly distributed along the axial direction. Additionally, its vibration isolation performance is analyzed through the dynamic performance of the shaft system.

2. Structural Design of New Water-Lubricated Stern Bearing

2.1. Traditional Water-Lubricated Bearing Structure

The traditional water-lubricated support bearing structure is shown in Figure 1. It consists of an outer bearing bush and an inner bearing bush. The inner bearing bush material is generally composed of rubber or polymer materials. It is fastened on the inner side of the bushing to support the shafting. The bearing is installed on the hull structure through the outer bushing. The lateral vibration of the shafting operation will be transmitted to the hull through the bushing-lining structure. At the same time, the radiated noise will be formed.

![Figure 1. The conventional structure of a water-lubricated stern bearing. 1 lining 2 bushing.](image)

During the actual navigation of the ship, the thrust generated by the propeller is transmitted to the propulsion shafting, and due to the influence of the navigation environment and operating conditions, the axis presents different bending forms under different operating states. At present, the design length-to-diameter ratio of a traditional water lubricated stern bearing is recommended to be greater than three. Meanwhile, coupled with the cantilever load of the stern propeller, there is an obvious eccentric load phenomenon in the stern bearing—the edge effect. This effect cannot ensure the good lubrication of the stern bearing under different operating conditions. This effect worsens the local lubrication state and even causes local friction of the bearing. As shown in Figure 2, these conditions will directly aggravate the wear of the stern bearing. In addition, these conditions will also aggravate the friction vibration of the stern bearing, produce abnormal noise, and affect the operational safety and stealth performance of the propulsion shafting.
As a part of the stern bearing, it is included in the rotating system. The elastic element is used to realize the static and dynamic load sharing of the stern bearing, reduce the edge load-sharing technology. Therefore, according to the operational requirements of a real ship, an adaptive stern bearing is designed in this paper. As shown in Figure 3, the structure of the adaptive stern bearing is mainly composed of three parts: bearing bush, elastic element, and damping alloy. The physical properties of the elastic element and damping alloy are added to the liner in the bearing. This means that the material of the liner is a composite. As a part of the stern bearing, it is included in the rotating system. The elastic element is used to realize the static and dynamic load sharing of the stern bearing, reduce the edge effect of the stern bearing and make its contact pressure evenly distributed along the axial direction to improve the service life of the bearing and reduce the friction excitation of the bearing. Damping alloy is used to attenuate the vibration transmission of shafting vibration from the bearing to the foundation to optimize the transmission characteristics of vibration. The bearing can be changed with the actual deflection state of the propulsion shafting. Finally, the stern bearing can realize load sharing, improve the bearing lubrication performance, reduce friction, and reduce vibration under each operating condition.

Figure 2. Partial wear of conventional water-lubricated stern tube bearing.

2.2. Design of Adaptive Water-Lubricated Stern Bearing

In order to solve the problem of eccentric wear of the above traditional bearings, it is necessary to improve the uniformity of load distribution in the bearings through the bearing load-sharing technology. Therefore, according to the operational requirements of a real ship, an adaptive stern bearing is designed in this paper. As shown in Figure 3, the structure of the adaptive stern bearing is mainly composed of three parts: bearing bush, elastic element, and damping alloy. The physical properties of the elastic element and damping alloy are added to the liner in the bearing. This means that the material of the liner is a composite. As a part of the stern bearing, it is included in the rotating system. The elastic element is used to realize the static and dynamic load sharing of the stern bearing, reduce the edge effect of the stern bearing and make its contact pressure evenly distributed along the axial direction to improve the service life of the bearing and reduce the friction excitation of the bearing. Damping alloy is used to attenuate the vibration transmission of shafting vibration from the bearing to the foundation to optimize the transmission characteristics of vibration. The bearing can be changed with the actual deflection state of the propulsion shafting. Finally, the stern bearing can realize load sharing, improve the bearing lubrication performance, reduce friction, and reduce vibration under each operating condition.

Figure 3. The structure of adaptive stern bearing.

3. Calculation and Analysis of Dynamic Parameters of the Stern Bearing Considering Lubrication

The stiffness and damping of a liquid film (dynamic parameters of bearing) are the main parameters of bearing dynamic performance, and these parameters are important factors affecting the vibration characteristics of shafting. Based on the micro disturbance method [10], the calculation method of the dynamic parameters of a water-lubricated stern bearing is presented in this paper. The dynamic model of bearing based on linear
theory is shown in Figure 4. The stiffness coefficient of the bearing is represented by $K$. It is composed of main stiffness and cross stiffness $K_{xx}, K_{yy}, K_{xy}, K_{yx}$, respectively. The damping coefficient is represented by $C$. It is composed of main damping and cross damping $C_{xx}, C_{yy}, C_{xy}, C_{yx}$, respectively.

![Figure 4. The model of water-lubricated bearing.](image)

3.1. Calculation of Liquid Film Stiffness

During the operation of the shafting, the position of the journal in the bearing is shown in Figure 5. Point $O$ is the center position of the sliding bearing, and point $O_1$ is the static balance position of the main shaft during normal operation. $\theta$ and $e$ are the attitude angle and eccentricity at the equilibrium position. It is assumed that axis $O_1$ is displaced to point $O_2$ by a slight disturbance along the left direction of the $X$-axis. The new eccentricity and attitude angle can be obtained by using the cosine theorem in the $\Delta OO_2O_1$. Then, the liquid film force components $F_{x1}$ and $F_{y1}$ at this time can be calculated. The expressions of $e_1$ and $\theta_1$ are:

![Figure 5. The position of the bearing center.](image)
The liquid film force is equivalent to the load-carrying capacity. The related calculation equation is presented as follows:

\[
\begin{align*}
F_x &= \iiint p \sin \varphi \, dx \, dy \\
F_y &= \iiint p \cos \varphi \, dx \, dy \\
F &= \sqrt{F_x^2 + F_y^2}
\end{align*}
\]

(1)

\(F_x, F_y\) are the component forces of load-carrying capacity in \(x\) and \(y\), respectively.

The paper mainly focuses on the water-lubricated stern bearing, so the governing equation in the fluid domain, RNG \(k - \varepsilon\), is given as follows:

\[
\begin{align*}
\frac{\partial (\rho k)}{\partial t} + \frac{\partial (\rho k u_i)}{\partial x_i} &= \frac{\partial}{\partial x_j} \left[ \alpha_k \mu_{\text{eff}} (\frac{\partial k}{\partial x_j}) \right] + G_k - \rho \varepsilon \\
\frac{\partial (\rho \varepsilon)}{\partial t} + \frac{\partial (\rho \varepsilon u_i)}{\partial x_i} &= \frac{\partial}{\partial x_j} \left[ \alpha_\varepsilon \mu_{\text{eff}} (\frac{\partial \varepsilon}{\partial x_j}) \right] + \frac{C_1 \mu_{\text{eff}} - C_2 \mu \varepsilon^2}{k}
\end{align*}
\]

(2)

\(k\) turbulent kinetic energy, \(C_{1k}, C_{2k}\) empirical constant, \(\varepsilon\) turbulent dissipation rate, \(\mu_{\text{eff}}\) diffusion coefficient, \(\alpha_k, \alpha_\varepsilon\) relaxation factor, \(u_{ij}\) time average velocity, \(\rho\) the density of liquid film, and \(G_k\) the turbulent kinetic energy term under the velocity gradient.

\[
\begin{align*}
\varepsilon_1 &= \sqrt{\varepsilon^2 + (\Delta x)^2 - 2\varepsilon \Delta x \cos(90^\circ + \theta)} \\
\sin(90^\circ + \theta) &= \frac{\Delta x}{\sin \alpha_1} \\
\theta_1 &= \theta + \alpha_1
\end{align*}
\]

(3)

In order to obtain accurate \(K_{xx}\) and \(K_{yy}\) results, the axis \(O_1\) can be moved to point \(O_3\) by shifting a small disturbance displacement to the right \(\Delta x\). Similarly, in the \(\Delta O O_3 O_1\), the sine and cosine theorems are used to calculate the eccentricity \(e_2\) and attitude angle \(\theta_2\) and angle \(\alpha_2\) when the axis is at \(O_3\) position. The liquid film force components \(F_{x2}\) and \(F_{y2}\) at this time can be calculated.

\[
\begin{align*}
\varepsilon_2 &= \sqrt{\varepsilon^2 + (\Delta x)^2 - 2\varepsilon \Delta x \cos(90^\circ - \theta)} \\
\sin(90^\circ - \theta) &= \frac{\Delta x}{\sin \alpha_2} \\
\theta_2 &= \theta + \alpha_2
\end{align*}
\]

(4)

To sum up, the exact calculation formulas of the liquid film stiffness coefficients \(K_{xx}\) and \(K_{yy}\) can be written as:

\[
\begin{align*}
K_{xx} &= \frac{F_{x1} - F_{x2}}{2\Delta x} \\
K_{yy} &= \frac{F_{y1} - F_{y2}}{2\Delta y}
\end{align*}
\]

(5)

Similarly, the perturbation momentum is applied in the \(y\) direction to obtain \(F_{x3}, F_{y3}, F_{x4}\), and \(F_{y4}\).

\[
\begin{align*}
K_{xy} &= \frac{F_{x3} - F_{x4}}{2\Delta y} \\
K_{yx} &= \frac{F_{y3} - F_{y4}}{2\Delta y}
\end{align*}
\]

(6)

The stiffness coefficient \(K\) calculated by the film force derivatives \([11]\) has a certain error compared with the actual stiffness coefficient. And the error is mainly related to the disturbance displacements \(\Delta x\) and \(\Delta y\). In order to make the results accurate enough, the values of \(\Delta x\) and \(\Delta y\) should be as small as possible. Generally, the disturbance displacement values of \(\Delta x/h_0\) and \(\Delta y/h_0\) are between 0.005 and 0.01, and \(h_0\) is the thickness of the liquid film.

3.2. Calculation of Liquid Film Damping

A small perturbation velocity increment \(+\Delta \dot{x}\) and \(-\Delta \dot{x}\) can be given \(\dot{x}\), respectively, during the calculation the damping coefficients \(C_{xx}\) and \(C_{yx}\). Substitute the Reynolds equations expressed in \(\dot{x}\) and \(\dot{y}\), and calculate the corresponding liquid film reaction
components $F_{x5}$, $F_{y5}$, $F_{x6}$, and $F_{y6}$, respectively. According to the definition of damping coefficient, the damping coefficients $C_{xx}$ and $C_{yx}$ are obtained as:

$$
\begin{align*}
C_{xx} &= \frac{F_{x6} - F_{x5}}{2\Delta x} \\
C_{yx} &= \frac{F_{y6} - F_{y5}}{2\Delta x}
\end{align*}
$$

(7)

Similarly, a small perturbation velocity increment $+\Delta \dot{y}$ and $-\Delta \dot{y}$ can be given $\dot{y}$, respectively. Substitute the Reynolds equations expressed in $\dot{x}$ and $\dot{y}$, and calculate the corresponding liquid film reaction components $F_{x7}$, $F_{y7}$, $F_{x8}$, and $F_{y8}$, respectively. Then, the damping coefficients $C_{yy}$ and $C_{xy}$ are respectively:

$$
\begin{align*}
C_{yy} &= \frac{F_{y7} - F_{y8}}{2\Delta y} \\
C_{xy} &= \frac{F_{x7} - F_{x8}}{2\Delta y}
\end{align*}
$$

(8)

Disturbance velocity $\Delta \dot{x}/(\omega h_0)$ and $\Delta \dot{y}/(\omega h_0)$ are usually between 0.005 and 0.01. $\omega$ is the rotational angular velocity of the journal, and the corresponding four damping coefficients can be obtained.

### 3.3. Solution Procedure

From the flow chart (as shown in Figure 6), we can see that firstly, the parameters of the bearing-rotor system are input. Next, for the film thickness, attitude angle, and pressure distribution, the assumption is carried out. According to the actual state, if the results are not satisfied, the film thickness and attitude angle will be modified. The film pressure and attitude angle are obtained by solving the Reynolds equation. If the pressure and attitude angle all satisfy the convergence criterion, the modes and load-sharing effect of the shaft can be calculated. Finally, combined with the above analysis, the analysis of the vibration absorption performance is implemented.

![Flow chart](image-url)
4. Performance Analysis of Adaptive Water-Lubricated Stern Bearing

4.1. Model Construction

As shown in Figure 7, the vibration characteristics of shafting are calculated and analyzed in this paper. The shafting is driven by the propulsion motor, and it transmits the motive power to the propeller through the elastic coupling, intermediate shaft, thrust shaft, stern shaft, propeller shaft, and inter shaft connecting flange. The structural parameters are shown in Table 1. As shown in Figure 3, the stern bearing adopts the adaptive structure, and its outer layer is a damping alloy with a measured damping ratio of 0.03.

![Propulsion shafting model](image)

**Figure 7.** The propulsion shafting model.

| Item                          | Value |
|-------------------------------|-------|
| Bearing width/L (m)          | 0.22  |
| Bearing diameter/D (m)       | 0.08  |
| Operating speed/V (r/min)    | 0–300 |
| Elastic element length/L (m) | 0.032 |
| Lubricant                    | water |
| Propeller weight/G (kg)      | 54    |

4.2. Analysis of Load-Sharing Effect of Adaptive Stern Bearing

The bearing lubrication model under the flexural state established in reference [12] is used to compare and analyze the lubrication state of the adaptive stern bearing and the traditional bearing. Compared with the literature [12], the water-lubricated stern bearing is in a flexural state in the present study. In Figure 8, the liquid film pressure distribution of the two bearings at 100 r/min is shown. It can be seen that the liquid film pressure distribution of adaptive stern bearings is more symmetrical than that of conventional bearings, and its maximum liquid film pressure is reduced from 0.059 MPa to 0.03 MPa. The position that appears also moves from the place of bearing one quarter away from the middle section to the axial middle section of the bearing. Its corresponding minimum liquid film thickness increases from 20 µm to 25 µm. The results show that the journal of traditional water-lubricated bearings is inclined due to cantilever support, and the bearing load near the end face will increase, resulting in a unilateral effect. Compared with traditional water-lubricated stern bearing in which the wear at the end of the bearing increases and the lubrication performance of bearings is affected [13,14], the edge effect of the adaptive stern bearing, in this manuscript, is weakened, and the lubrication condition of
the adaptive stern bearing is improved obviously. Therefore, the service life of the bearing is improved, and the wear of the bearing is reduced.

**Figure 8.** The difference in film pressure in two types of bearings. (a) Adaptive liquid film pressure distribution of stern bearing. (b) Liquid film pressure distribution of traditional water-lubricated bearing.

### 4.3. Analysis of Vibration Absorption Performance of Adaptive Stern Bearing

For the shafting bench shown in Figure 7, a dynamic model of the shafting is established based on the stern bearing lubrication model established in reference [12] and in combination with the expression of liquid film stiffness and damping derived in Section 2. The material properties of each part are shown in Table 2.

| Table 2. Material Properties of bearing parts. |
|-----------------------------------------------|
| Structure Name      | Young's Modulus (MPa) | Poisson's Ratio | Density (kg/m³) |
| principal axis      | 210,000               | 0.30           | 7850           |
| bushing             | 305                   | 0.37           | 2200           |
| lining              | 110,000               | 0.35           | 8800           |
| Inner damping alloy | 90,000                | 0.27           | 7330           |
| Elastic element     | 6                     | 0.27           | 1000           |
| Outer damping alloy | 90,000                | 0.27           | 7330           |

Firstly, the influence of the adaptive stern bearing on the critical speed of shaft whirling vibration is analyzed. The calculated results are shown in Table 3, and the vibration modes of each order are shown in Figure 9. The results show that the influence of the adaptive bearing on critical speed is mainly concentrated on the first-order shaft frequency and blade frequency. The reduction is 4.3% and 3.3% from 1406.85 r/min and 182.96 r/min to 1346.81 r/min and 176.93 r/min, respectively, and the effect on the modes of whirling vibration is relatively small [15,16].

| Table 3. Critical speed of axial frequency and blade frequency of cyclotron vibration with adaptive stern bearing (r/min). |
|---------------------------------------------------------------|
| Order          | Steel Bearing |         | Adaptive Bearing |         |
|                | Axial Frequency forward Cyclotron | Blade Frequency forward Cyclotron | Axial Frequency forward Cyclotron | Blade Frequency forward Cyclotron |
| First order    | 1406.85       | 182.96  | 1346.81           | 176.93  |
| Second order   | 2027.62       | 288.73  | 2027.60           | 288.73  |
| Third order    | 3656.71       | 508.94  | 3248.78           | 463.01  |
| Fourth order   | 12,058.70     | 1612.66 | 10,999.80         | 1271.21 |
| Fifth order    | 14,922.30     | 1848.04 | 14,282.80         | 1613.46 |
The transverse unit excitation force is loaded at the center of gravity of the propeller, and the frequency range is 1~100 Hz. The vibration velocity response at the stern bearing position is extracted to evaluate the vibration transmission characteristics from the propeller to the stern bearing [17,18].

Figure 10 shows the speed response of the stern bearing joint with different bearing structures. It can be seen that the vibration response at the stern bearing is dominated by the fixed frequency of first-order and third-order whirling vibration. After the adaptive stern bearing is used, the response at the first-order fixed frequency decreases from 0.0056 mm/s to 0.003 mm/s with a decrease of 46.43%. Therefore, the vibration energy transmitted by the foundation is reduced.

In addition, the calculation results show that the foundation stiffness is also a key factor affecting the bearing vibration response. The original support stiffness of the foundation is $K_0$, and the foundation stiffness is adjusted to $2K_0$ and $3K_0$, respectively. The speed response of the stern bearing under different stiffness is shown in Figure 11. The results show that the vibration response of the stern bearing gradually decreases with the increase in foundation stiffness [19–21], and the decreases in the vibration response near 20 Hz are 51.13% and 67.66%, respectively. Therefore, on the basis of the adaptive stern bearing, the bearing vibration response can be further reduced by properly increasing the support stiffness of the hull structure. Consequently, it is more advantageous to operate stably.
In addition, the fixed-frequency offset is hardly affected by different foundation stiffness, mainly because the bearing stiffness is weaker than the foundation stiffness. Two stiffness are connected in series, and the bearing stiffness is the main comprehensive stiffness.

![Graph showing speed response of stern bearing under different foundation stiffness.](image)

**Figure 11.** Speed response of stern bearing under different foundation stiffness.

5. Conclusions

In this paper, an adaptive stern bearing structure is designed to achieve dynamic load sharing of the water-lubricated stern bearings, and the lubrication performance and shafting vibration characteristics of the adaptive stern bearing are theoretically analyzed. The results show that:

(1) Compared with the traditional water-lubricated bearing structure, the stern bearing with an adaptive structure can change the bearing deflection angle with the inclination of the shafting to achieve the bearing load sharing. The maximum liquid film pressure is significantly reduced, and the lubrication state of bearings is significantly improved. It can also effectively reduce bearing wear and improve bearing service life.

(2) After adopting the adaptive stern bearing, the vibration transmission characteristics from propeller excitation to the bearing node are optimized. The decrease in vibration response at the first fixed frequency is 46.43%. At the same time, if the support stiffness of the hull structure at the stern bearing is properly increased, the vibration response of the stern bearing will be further reduced.

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