Numerical Analysis on the Loading Characteristic of Hydrodynamic Water-lubricated Bearing

Huang Chonghai\textsuperscript{1,2}, Zhang Yousen\textsuperscript{3}, Li Yubing\textsuperscript{4}, Chen Kai\textsuperscript{1,2}, Wang Wei\textsuperscript{1,2}, Ke Hanbing\textsuperscript{1,2}, Xiao Qi\textsuperscript{1,2,*}

\textsuperscript{1}Science and Technology on Thermal Energy and Power Laboratory, Wuhan, Hubei, 430205, P.R. China;
\textsuperscript{2}Wuhan Second Ship Design and Research Institute, Wuhan, Hubei, 430205, P.R. China;
\textsuperscript{3}Shenhua Guohua(Beijing) Electric Power Research Institute Co. Ltd., Beijing, 100025, P.R. China;
\textsuperscript{4}Wuhan Electric power technical college, Wuhan, 430079, P.R. China

zhiyan7@sina.com

Abstract. A fixed pad seawater-lubricated bearing has been designed according to the structure parameters and loading requirements of axial flow pump. Water film lubrication model was built and 3-D simulations were performed for water-lubricated bearings. The pressure distribution inside the bearing predicted by simulations agrees very well with the theoretical predictions. The formation mechanism of dynamic pressure inside water–lubricated bearing was revealed. Then the pressure distribution characteristics were analyzed. It was found that the loading capacity would increase linearly with the angular speed, and the designed bearing could meet the loading requirements even at the lowest angular speed of the given axial flow pump.

1. Introductions

Journal bearings are usually applied for orientating and supporting the rotator of integral axial flow pumps, and reducing the frictional loss [1]. In general, journal bearing could be classified into oil-lubricated and water-lubricated journal bearing according to lubricants. Oil-lubricated journal bearing is usually made of metallic antifriction materials such as babbit alloys, bronzes or aluminum alloys. Environmental pollution problems may be induced when lubricant oil leakage. To avoid the problems caused by lubricant oil leak, sometimes complex structures are adopted for seal. Increasing ecological awareness and stringent requirements for modern ships makes water-lubricated bearings attract more and more attentions in recent years [2-4]. There are two unique advantages for water-lubricated journal bearings [5,6]: firstly water is a safe and environment friendly lubricant, and can be obtained easily and cheaply without any lubrication pump; secondly, no special bearing location space and extra lubrication oil delivery systems are needed. Compared to oil-lubricated bearing, the structure of water-lubricated bearing is simpler and its operation and maintenance cost is much lower. In recent years, the water-lubricated journal bearing has been widely applied in ships, hydraulic turbines, pumps, shipping locks and agricultural machinery [7,8].

However, due to the low water viscosity, the formation of hydrodynamic lubrication is difficult
[9,10]. For example, the kinematic viscosity of water is only one percent of that of ISOVG lubrication oil. It contributes to low hydrodynamic load-carrying capacity, which increases with the fluid viscosity and decrease with the thickness of lubricant film. The thickness of water lubricant film is only one tenth compared to the ISOVG oil lubricant film under the same conditions. To make the water-lubricated journal bearing works under different stages, the minimum thickness of water lubricant film should be larger than the sum of roughness values of the bearing surface and bearing location surface, which brings great challenges for water-lubricated bearing design.

Hydrodynamic load-carrying capacity of water-lubricated bearing depends on the water film pressure distributions, which could hardly be obtained by the experimental or theoretical methods. CFD simulations, which could offer detail field information by computational method, have been gradually applied for bearing investigations in recent years. Gao et al. [11] evaluated the hydrodynamic load-carrying capacity of water-lubricated journal bearing by 3D CFD analysis. The relationship between hydrodynamic load-carrying capacity and the magnitude of the transition-arc structure dimension is established for different L/D. Hu et al. [12] investigated the flow field distribution of liquid film of water-lubricated bearing in start transients by 3D unsteady simulations. The influence of eccentricity on the film thickness was analyzed.

In this paper, a fixed pad water-lubricated bearing was designed based on the structure parameters and loading requirements of the axial flow pump. Then a 3-D numerical model was established for the designed bearing. The pressure field inside the bearing was obtained and then load-carrying characteristics of the water-lubricated bearing were verified.

2. Bearing design

According to the design requirements in this paper, the hub diameter is 100 mm. The angular speed is 1500 rpm. The bearing load is 196 N and fix pad is adopted. The pump is designed to be operated under different angular speeds, includes 1/2 full angular speed, 3/4 full angular speed and full angular speed. In general, the fluid dynamic supporting capacity offered by the hydrodynamic journal bearing would decrease with the angular speed. The minimum design angular speed is 750 rpm. The calculation process for water-lubricated journal bearing was shown in table 1.

| nomenclature         | numeration                           | value       |
|----------------------|--------------------------------------|-------------|
| loading $F$          | given                                | 196 N       |
| bearing inner diameter $D$ | decided by hub diameter          | 6 cm        |
| bearing width-diameter ratio $B/D$ | 0.4~1.0                          | 1.0         |
| bearing width $B$    | $B=D (B/D)$                          | 6 cm        |
| designed angular speed $n_0$ | 1/2 full speed                   | 750 r/min   |
| angular velocity $\omega_0$ | $\omega_0 = 2\pi n_0$              | 78.5 /s     |
| bearing wrap angle $\alpha$ | given                            | 360         |
| clearance ratio $\Psi$ | $\Psi = \frac{D}{2}$               | 0.002       |
| radius clearance $c$ | $c = \Psi D/2$                      | 0.006 cm    |
| average pressure $p_m$ | $p_m = F/(BD)$                    | 5.44N/cm²   |
| lubricant            | 30°C sea water                      | sea water   |
| kinematic viscosity $\eta$ | as 30°C water                | 0.9×10⁻⁷ N.s/cm² |
| Sommerfeld number $C_p$ | $C_p = \frac{p_m \Psi^2}{(\eta \omega)}$ | 3.08 |
| eccentricity ratio $\epsilon$ | decided by $C_p^{[6]}$            | 0.8         |
| eccentricity $e$     | $e=\epsilon c$                      | 0.0048 cm   |
minimum thickness of water film $h_{\text{min}}$ 

$$h_{\text{min}} = c(1-\varepsilon) \quad 0.0012\text{cm}$$

roughness of shaft journal 

given 

average roughness height of shaft journal \( R_1 \) 

decided by material 

0.00032\text{cm}

roughness of bearing shell 

given 

0.8

deflection of shaft journal \( y_1 \) 

neglected when \( p_m < 30\text{N/cm}^2 \) 

0

offset of shaft journal \( y_2 \) 

neglect 

0

allowable minimum thickness of water film 

$$[h_{\text{min}}] = 1.5(R_1 + R_2 + y_1 + y_2) \quad 0.00096\text{cm}$$

checking terms 

$h_{\text{min}} \geq [h_{\text{min}}]$ satisfy

friction coefficient in load zone \( C_f \) 

view photograph [6] 

3.0

coefficient \( \xi \) 

given 

1.0

friction coefficient \( C_\mu \) 

\( C_\mu = C_f \) 

3.0

friction force in load zone \( F_\mu \) 

\( F_\mu = F_C \xi \Psi \) 

1.18

power consumption \( N \) 

\( N = F_\rho \omega \) 

0.00278\text{kW}

discharge coefficient in load zone \( k_{Q1} \) 

view photograph [6] 

0.24

water supply pressure \( p_s \) 

given 

0.03\text{MPa}

coefficient \( \xi \) 

view photograph [6] 

0.25

total discharge \( Q \) 

\( Q = 2k_{Q1} \Psi \rho \omega B \) 

4.07\text{cm}^3/\text{s}

water temperature rise \( \Delta t \) 

\( \Delta t = 590N/Q \) 

0.40 \text{°C}

---

3. Numerical model

The structure of the simulated dynamic water-lubricated bearing is shown in Fig. 1. The liquid film between the shaft journal and bearing shell would be simulated by commercial software ANSYS Fluent. As shown in Table 1, the shaft radius \( r \) is 30 mm. The bearing width \( L \) is 60 mm, and eccentricity \( e \) is 0.06 mm. The minimum film thickness \( h_{\text{min}} \) is 0.012 mm and the maximum \( h_{\text{max}} \) 0.108 mm. Structural grid is adopted and the grid system with a scale of 20×400×100 is used in our simulations after the grid-independent test.

As the dynamic water-lubricated journal bearing is connected with the axial flow pump, the inlet pressure of the dynamic water-lubricated journal bearing is set to be that of the outlet of axial flow pump. The axial flow pump head is 3 m, and the inlet pressure of the bearing is 0.03 MPa. Outflow condition is applied at the bearing outlet as the pressure and velocity at outlet is unpredictable before simulations. The outer wall would be set as moving wall and the angular speed is 78.5 rad/s as the minimum rotational speed of the simulated bearing is 750 rpm. Non-slip wall condition is set for the inner wall as shown in Fig.1.

4. Results and discussion

Firstly, the simulation results would be evaluated by comparing with the theoretical predictions. According to the Reynolds equations, the dynamic pressure could be calculated as follows [7]:

$$p = \frac{6\mu \omega \varepsilon (2 + \varepsilon \cos \theta) \sin \theta}{\Psi^2 \left(2 + \varepsilon^2\right)(1 + \varepsilon \cos \theta)^2}$$

(1)

The circumferential distribution of the fluid pressure predicted by Eq.1 is presented in Fig. 2. It could be found in that the simulation results agree very well with the theoretic predictions. The maximum dynamic pressures predicted by the theoretic method and CFD simulations are 0.2289 MPa and 0.2368 MPa respectively. The difference is less than 3.5%, which means that the CFD prediction
accuracy is acceptable. It could also be observed in Fig. 2 that: when $0 < \theta < \pi$, the dynamic pressure is positive, while when $\pi < \theta < 2\pi$, it turns to be negative. In fact the liquid film would be broken when the dynamic pressure is negative, so the dynamic pressure should be 0 when $\pi < \theta < 2\pi$.

Then the influence of angular speed on the pressure distribution would be investigated. As shown in Figure 3, the thickness of liquid film would reach to the maximum value at the top and minimum value at the bottom. The film form at left is dovetail and divergent at the right. Fig. 4 shows the liquid film pressure distribution inside the journal bearing when the angular speed is 78.5 rad/s. It could be found that the pressure value is positive in the dovetail zone, which means that the liquid film could be established in this zone. As the liquid film pressure value is negative in the divergent zone, the fluid film would break and the pressure value in this zone turns to be 0. It could also be found in Fig. 4 that the pressure does not get the highest/lowest values at the narrowest/widest place. The pressure maximum value appears at the narrowest point in dovetail zone, while the minimum at the narrowest point in divergent zone.
Fig. 5 shows the liquid film pressure distribution along the circumferential direction. It could be observed that the pressure distribution of liquid film is similar for different angular speeds. The pressure value of liquid film is positive in the dovetail zone and negative in the divergent zone. The maximum value increases with the angular speed and the minimum decreases with $\omega$. We could also find that the extreme values appear almost the same positions under different angular speeds. The pressure gradient would also increases with the angular speeds.

Fig. 6 shows the thrust force of journal bearing induced by the dynamic pressure of liquid film under different angular speeds. It could be observed that the thrust force linearly increases with the rotational speeds. As the journal bearing loading capacity is 196 N, the angular speed should exceed 61.3 rad/s to realize the required loading capacity. For the designed water-lubricated journal bearing, the full rotation speed is 157 rad/s and the minimum angular speed is 78.5 rad/s, which is higher than the required angular speed. Therefore, the designed dynamic water-lubricated journal bearing can satisfy the bearing load requirement.

5. Conclusions

In this paper, a fixed pad water-lubricated bearing was designed according to the structure parameters and the loading requirements of the axial flow pump. 3-D numerical model is established to simulate the designed bearing. It is found that the predicted dynamic pressure distribution by CFD method agrees well with the theoretical prediction. Then forming process of the dynamic pressure inside the liquid film and the pressure distribution is investigated. The simulation results show that the thrust force induced by the liquid film dynamic pressure increases linearly with the angular speed. The extreme values appear almost the same positions under different angular speeds. The designed bearing can offer the loading capacity even at the lowest rotating speed of the axial flow pump.

References

[1] B. Manshoor, M. Jaat, Zaman Izzuddin, Khalid Amir, CFD analysis of thin film lubricated journal bearing [J]. Procedia Engineering, 2013, 68: 56-62;
[2] Hua Xijin, Numerical analysis for fluid lubrication on longitudinal grooved water-lubricated bearings based on Fluent (in Chinese) [D], Chongqing University, 2009;
[3] Xiong Yongqiang, Numerical study of the water-lubricated journal bearings considering the effects of cavitation (in Chinese) [D], Shanghai Jiao Tong University, 2011;
[4] Xiong Yongqiang, Yin Zhongwei, Peng Yinghong, The effects of mechanical properties of bearing bush on water lubricated plastic bearing lubricating property (in Chinese) [J], Lubrication Engineering, 2011, 36(2): 9-11;
[5] Wang Fangfang, Chen Wei, Zhang Youfeng, Simulation on the flow fields of a novel water-lubricated static and dynamic hybrid bearing for high-speed spindle (in Chinese) [J], Lubrication Engineering, 2010, 35(12): 28-31;
[6] Tu Lin, Li Duomin, Duan Zihua. Pressure field numerical simulation of hydrodynamic bearing based on Fluent (in Chinese) [J], Lubrication Engineering, 2011, 36(4):82-86;
[7] Zhou Jianhui, Liu Zhenglin, Zhu Hanhua, Hai Pengzhou, Experimental study on frictional characteristic of rubber water-lubricated stern tube bearings [J], Journal of Wuhan University of Technology, 2008, 32: 842-844;
[8] Peng Yaling, Zhang Zhigu, Chen Rugang, Fang Chengyue, Study of CFD aid design of ship water journal bearing of shaft[J], Lubrication Engineering, 2008, 5: 72-76.
[9] Gao Gengyuan, Yin Zhongwei, Jiang Dan, Zhang Xiuli, Wang Yanzhen, Analysis on design parameters of water-lubricated journal bearings under hydrodynamic lubrication [J], Journal of Engineering Tribology, 2016, 230: 1019-1029;
[10] Wang Youqiang, Shi Xiujian, Zhang Lijiang, Experimental and numerical study on water-lubricated rubber bearings [J], Industrial Lubrication and Tribology, 2014, 66: 282-288;
[11] Gao Gengyuan, Yin Zhongwei, Jiang Dan, Zhang Xiuli, CFD analysis of load-carrying capacity of hydrodynamic lubrication on a water-lubricated journal bearing [J], Industrial Lubrication
and Tribology, 2015, 67: 30-37;
[12] Hu QL, Hu JN, Ye XY, Zhang DS, Zheng JB, Flow field distribution of liquid film of water lubricated bearing-rotor coupling systems [C], IOP Conf. Series: Materials Science and Engineering, 2016, 129: 012056.