Analysis on the thrust bearing lubrication in pumped storage unit based on fluid-solid-thermal interactions

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Abstract. The bidirectional thrust bearings are used to balance the thrust load of the entire shaft system which play an important role in the pumped storage units. The lubrication is usually analyzed by simultaneously solving the Reynolds equation, energy equation, film thickness equation of the film and the heat conduction equation, elastic equilibrium equation of the pad. However, in this method the assumptions for the inlet temperature of the film and the wall heat coefficients on the pad greatly affect the numerical accuracy. This study used fluid-solid-thermal interaction method, built three dimensional (3D) FEM model for the pad and collar and the 3D CFD model for the oil film, and then analyzed the influence of the thrust loads and the Babbitt layer on the bearing performance. The results show that increasing the thrust load will rapidly decrease the minimum film thickness, increase the maximum pressure and maximum temperature on the pad. Increasing the thickness of the Babbitt layer on the pad sliding surface will increase the minimum thickness of the film, decrease the maximum temperature on the pad which improve the bearing lubrication.

1. Introduction
The pumped storage unit can start and stop simply and quickly, and it is a reliable power supply for the frequency and the peak regulation of power grid. Unlike conventional hydrogenerator set, generator and pump-turbine in the storage unit need to operate in two modes, i.e., forward power generation at peak power consumption and reverse pumped storage at low power consumption. With the increase of power grid capacity in China, pumped storage power stations are accelerating constructed to meet the growing demand for the power regulation. By the end of 2014, China had built pumped storage units with a total installed capacity of 21,800 MW in Guangzhou, Xilongchi, Huizhou and Xianyou. As an important part of pump-turbine unit, bidirectional thrust bearing balances the axial hydraulic thrust while supporting the weight of the whole rotating component. Its lubrication performance directly affects the safe operation of the unit and the stability of power grid supply.

In conventional hydropower units, the pad of one-way thrust bearing is usually eccentrically supported, and the maximum thrust load is obtained when the angular eccentricity is 58% ~ 60%.
However, the bearing pad of the pumped storage unit must be centrally supported to take into account the turbine mode and the pump mode, which makes the bidirectional thrust bearing unable to obtain a large load bearing capacity, and limits the development of larger capacity unit. On the other hand, the frequent start-stop and high-speed operation of the pumped storage unit make the operating condition of bidirectional thrust bearings worse. In recent years, some pumped storage units have been forced to shut down because of the failure of thrust bearings [1-3]. Therefore, it is necessary to calculate the lubrication performance of thrust bearings accurately in the design stage to meet the requirements of operation and to ensure the safe and stable operation of the units. Huang Bin et al. [4,5] and Wu Zhongde et al. [6] carried out three dimensional (3D) numerical thermal-elastic-hydrodynamic (TEHD) calculation of bidirectional thrust bearings through the simultaneous of oil film Reynolds equation, energy equation, thickness equation, heat conduction equation and elastic equilibrium equation, and compared with the experimental results. In the calculation process, it is necessary to assume the oil inlet temperature and the convective heat transfer coefficient on the pad surface and collar surface, which is an important factor affecting the accuracy of calculation. With the development of computational fluid dynamics (CFD), foreign scholars began to apply fluid-solid interaction (FSI) method to the TEHD lubrication analysis of journal bearings [7,8] and thrust bearings [9,10]. Wodtke et al. [11] established 3D lubricant CFD model and 3D pad FEM model and used FSI method to analyse TEHD lubrication performance of journal bearings and thrust bearings. Wodtke’s method didn’t need to assume the oil inlet temperature and the convective heat transfer coefficient, instead, it can be obtained automatically in the calculation process. Pajaczkowski et al. [12] used FSI method to calculate the transient lubrication characteristics of a single disk thrust bearing of pumped storage unit. The model considered the oil groove on the pad surface and the chamfering of the inlet and outlet edge. The results showed that the minimum oil film thickness in the transient process was quickly stabilized, while the deformation of the pad and the mirror plate took more time to stabilize.

Thrust bearings may be rubbed during start, stop, overload and accidental shock period, which makes the bearing in mixed lubrication or even oil-poor lubrication and affects the safe and stable operation of units seriously. Babbitt has excellent performances [13,14], especially when the lubricating oil supply is temporarily interrupted, it still has good self-lubricating and abrasion resistance performances, so it is widely used in hydroelectric unit thrust bearings. Usually, the certain thickness of Babbitt is directly cast on the pad base, then the surface is scraped to improve the operating and fit accuracy, increase the contact area and improve the load bearing capacity. The current units, like the Babbitt pads used in the Three Gorges, do not need to scrape the surface [15].

Based on the basic principle of fluid-solid-thermal coupling, a 3D FEM model of thrust bearing pad and collar, and a 3D CFD model of surrounding lubricant for a pump-turbine thrust bearing are established. The influence of thrust load and babbitt coating thickness on the TEHD lubrication performance of the bidirectional thrust bearing are analysed by the bidirectional coupling method, which has theoretical and engineering reference significance.

2. Basic Theory of Fluid-Solid-Thermal Bidirectional Coupling
The numerical calculation model of thrust bearing lubrication is composed of fluid domain (oil film and surrounding lubricant) and solid domain (pad and collar), the fluid-solid-thermal bidirectional coupling method is adopted.
2.1. Governing equation in fluid domain
Unlike the traditional method of solving simplified Reynolds equation to calculate oil film lubrication characteristics, this paper solves momentum equation, continuity equation and energy equation simultaneously on the basis of the full 3D transient turbulent N-S equation. The influence of fluid inertia and volume force are taken into account, and the viscosity-temperature effect on oil is considered. The lubricating oil is assumed to be single-phase and incompressible Newtonian fluid, and the effect of temperature and pressure on density is neglected.

Reynolds time-averaged continuity equation:
\[
\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x_j} \left( \rho U_j \right) = 0
\]  

(1)

Reynolds time-averaged momentum equation:
\[
\frac{\partial \rho U_i}{\partial t} + \frac{\partial}{\partial x_j} \left( \rho U_i U_j \right) = -\frac{\partial \rho}{\partial x_i} + \frac{\partial}{\partial x_j} \left( \tau_{ij} - \rho u_i u_j \right) + S_M
\]

(2)

Where \( \rho \) is the fluid density, \( \tau \) is the molecular stress tensor, \( S_M \) is the volume force, \( \rho u_i u_j \) is the Reynolds stress.

Reynolds time-averaged total energy equation:
\[
\frac{\partial \rho h_{tot}}{\partial t} - \frac{\partial \rho}{\partial x_j} \left( \rho U_i h_{tot} \right) = \frac{\partial}{\partial x_j} \left( \rho \frac{\partial T}{\partial x_j} - \rho \overline{u_i h} \right) + \frac{\partial}{\partial x_j} \left[ U_i \left( \tau_{ij} - \rho u_i u_j \right) \right] + S_E
\]

(3)

Where \( \rho \overline{u_i h} \) is the additional turbulent flux, \( \frac{\partial}{\partial x_j} \left[ U_i \left( \tau_{ij} - \rho u_i u_j \right) \right] \) is the additional turbulent flux, \( h_{tot} \) is the total enthalpy.

The viscosity of oil decreases with the increase of temperature, and the variation is more obvious at lower temperature. The viscous-temperature relationship can be described by the following formula:
\[
\mu = \mu_0 \left( \frac{20+T_0}{20+T} \right)^3
\]

(4)

Where \( \mu_0 \) is the dynamic viscosity of oil at \( T_0 \). \( T \) is the absolute temperature.

There are two kinds of flow patterns in the fluid domain of thrust bearing: laminar flow in the oil film and turbulent flow around the pad. The SST turbulence model can obtain appropriate flow transport characteristics by restricting eddy viscosity, thus it’s suitable for the flow field calculation. Eddy viscosity is expressed as:
\[
\nu_t = \frac{a_k k}{\max \left( a_i \omega, SF_2 \right)}
\]

(5)

Where \( F_2 \) is the mixed function, \( S \) is the invariants of strain rate.

2.2. Governing equation in solid domain
Thermal-elastic constitutive equation:
\[
\frac{\partial Q}{\partial t} = T_0 \left\{ \beta \right\} + \frac{\partial \{\varepsilon\}}{\partial t} + \rho C_v \frac{\partial (\Delta T)}{\partial t} - [K]\nabla^2 T
\]

(6)
Where \( \{\varepsilon\} \) is the total stress vector, \( \Delta T = T - T_{\text{ref}} \), \( T_0 = T_{\text{ref}} + T_{\text{off}} \) is the absolute temperature, \( T_{\text{ref}} \) is the reference temperature, \( T_{\text{off}} \) is the difference between \( T_0 \) and \( T_{\text{ref}} \), \( \rho \) is the material density, \( \{\beta\} \) is the thermal-elastic coefficient vector.

2.3. Fluid-Solid-Thermal Coupling Equation

In fluid-solid-thermal bidirectional coupling calculation, the data transfer in the fluid-solid interface must satisfy the conservation of pressure, displacement, heat flow and temperature, i.e.:

\[
\begin{align*}
\tau_f \cdot n_f &= \tau_s \cdot n_s \\
d_f &= d_s \\
q_f &= q_s \\
T_f &= T_s
\end{align*}
\]  

(7)

where subscript \( f \) denotes fluid, subscript \( s \) denotes solid.

ANSYS multi-physical field coupling function MFX is used as a platform for TEHD lubrication analysis of the thrust bearing. It uses CFX to solve fluid domain (lubricant) and mechanical APDL to solve solid domain (pad and collar). Both of them are iterated to conduct TEHD bidirectional coupling analysis. In the calculation process, all the coupled data, such as temperature, pressure, displacement and heat flux, are transmitted through the fluid-solid coupling interface. The pressure and heat of the lubricating oil are transferred to the solid domain, which is used as the boundary condition for the calculation of the thermal-elastic deformation of the pad and the collar. Conversely, the calculated pad temperature is the boundary of the oil film in the next step. The thickness distribution of the oil film is influenced by the pad deformation, which further affects the pressure and temperature characteristics of the oil film. This method is the separation coupling method, i.e. the fluid domain and the solid domain are coupled by the data exchange at the interface. Therefore, the two solving domains can adopt different grid densities. In general, the fluid domain grid is denser, while the solid grid is sparser, which can satisfy their numerical accuracy while using less computing time.

3. Numerical model for calculation of thermal elastic flow of thrust bearing

The tilting pad thrust bearing of a pumped storage unit is analyzed, as shown in Figure 1. The bearing consists of a collar and ten thrust pads that are completely immersed in the lubricating oil. In order to simplify the calculation, assuming that the thrust load is evenly distributed on each pad surface, so the
rotational symmetry model can be used in the calculation, i.e., only 1/10 of the bearing model is established, which consists of one pad, 1/10 of the collar and 1/10 of the lubricating oil, as shown in Figure 2.

Table 1. Structural parameters and operating conditions

| Item                                      | value          |
|-------------------------------------------|----------------|
| Pad outer radius $R_1$/mm                | 1335           |
| Pad inner radius $R_2$/mm                | 775            |
| Pad angle $\theta$/deg                   | 31             |
| Pad width $B$/mm                         | 560            |
| Pad thickness $H_1$/mm                   | 203            |
| Babbitt layer thickness, $H_2$/mm        | 3              |
| Number of pads $n$                       | 10             |
| Thrust load $F$/t                        | Up to 400      |
| Rotational speed $\omega$/rpm            | 500            |
| Radial pivot position $O_r$/mm           | 1065           |
| Circumferential pivot eccentricity $O_c$/%| 50             |
| Collar outer radius $R_3$/mm             | 1335           |
| Collar inner radius $R_4$/mm             | 775            |
| Collar thickness $H_c$/mm                | 660            |
| Oil capacity $Q/(L \cdot s^{-1})$         | 2.5            |
| Oil supply temperature $T$/°C             | 25             |

Bearing structural parameters and operating conditions are shown in Table 1, material properties are shown in Table 2.

Table 2. Material properties

| Material                      | Alloy steel | Babbitt | lubricant |
|-------------------------------|-------------|---------|-----------|
| Density $\rho$/kg·m$^{-3}$    | 7850        | 7420    | 890       |
| Young's modulus of the pad $E$/Pa | $2.10 \times 10^{11}$ | $5.30 \times 10^{10}$ | –         |
| Poisson's ratio $\nu$         | 0.33        | 0.33    | –         |
| Thermal conductivity $\lambda/(W \cdot m^{-1} \cdot K^{-1})$ | 50         | 38      | 0.145     |
| Thermal expansivity $\alpha/K^{-1}$ | $1.20 \times 10^{-5}$ | $2.20 \times 10^{-5}$ | $3.8 \times 10^{-4}$ |
| Specific heat $C/(J \cdot Kg^{-1} \cdot K^{-1})$ | 465       | 251    | 2000      |
| Dynamic viscosity $\mu_0/(Pa \cdot s)$ | –         | –      | $2.848 \times 10^{-2}$ |

The lubricating oil between the pads is divided into two parts, one adjacent to the upstream pad and one adjacent to the downstream pad, the two parts use the rotating periodic boundary conditions to simulate the entire bearing. By using this method, continuous numerical solutions considering flow and heat transfer between all pads can be obtained, and the real operating conditions of the thrust bearing can be simulated. The oil film thickness between the pad and the working surface of the mirror plate is initialized to 500 μm, which changes continuously during the iteration process until it reaches the static equilibrium position. After checking the grid independence, 167,000 eight-node elements are used in
the fluid domain, in which 15 layers are divided in the direction of oil film thickness. Sparser grids are used in the solid domain, with a total of 3,900 intermediate-node elements.

4. Results

4.1. Basic lubrication characteristics

The basic lubrication characteristics were analyzed and the result has been published in reference [16]. This part cites some results to describe the basic lubrication performance and it is helpful for understanding the thrust load and pad coating thickness’ effect.

4.1.1. Oil film pressure.

Firstly, the lubrication performance of the bearing is calculated when the thrust load is 260 t and the speed is 500 rpm. Figure 3(a) shows the oil pressure distribution on the pad surface, wherein the leading side on the right and the trailing side on the left. It can be seen that the maximum pressure is located near the center of the pad, which is different from the one-way thrust bearing (one-way thrust bearing’s maximum pressure is biased toward the trailing side). It is because the one-way thrust bearing pad is usually eccentrically arranged, and the bidirectional thrust bearing pad is centrally supported. Figure 3(b) is the oil pressure distribution on the mirror plate surface. Its distribution and numerical value are the same as the pad surface, i.e., the pressure gradient in the direction of oil film thickness is 0, which is consistent with the assumption that the pressure in the direction of oil film thickness is the same when solving the Reynolds lubrication equation. In addition, there is no obvious negative pressure zone in the oil film, which avoids the occurrence of cavitation. Due to the thrust bearing adopts the elastic tilting pad, the pad can incline along both circumference and radial direction, and has great adaptive ability, so that there is no obvious divergence area in the oil film (along the rotational direction), thus avoid the occurrence of cavitation.

![Figure 3. Pressure distribution in the oil film (thrust load 260t)](image-url)
4.1.2. Thermal-elastic deformation. The action of pressure and temperature gradient of the oil film induces the mechanical and thermal deformation of the pad and the collar. Figure 4(a) is the comprehensive axial thermal-elastic deformation of the pad surface. The pad surface is convex upward and the maximum deformation occurs at the outer radius (near the trailing side), reaching 176.34 μm. The thermal deformation plays a major role because it usually causes the pad surface to protrude, while the mechanical deformation usually causes the pad surface to sink. Figure 4(b) is the axial thermal-elastic deformation of the mirror plate surface, which changes slightly along the rotational direction, but has obvious deformation along the radial direction, showing the shape that the inner radius sinking and the outer radius warping, and the inner radius deformation is obviously larger than that of the outer radius.

![Figure 4. Deformation distribution of the oil film (thrust load 260t)](image)

The maximum deformation value of pad surface is about 170 μm, while the mirror plate surface is about 360 μm, which reaches the magnitude of the oil film thickness and significantly changes the oil film thickness distribution, as shown in Figure 5. The minimum oil film thickness is about 142 μm, which locates at the trailing side (near the inner radius) of the pad. Along the rotating direction, the oil film thickness gradually decreases and forms a convergent wedge, which is conducive to generating enough hydrodynamic pressure to meet the great thrust load requirements of the units.

![Figure 5. Film thickness distribution (unit: μm)](image)
4.1.3. Oil film temperature. Figure 6(a) shows the temperature distribution on the pad surface. The temperature gradually increases along the rotational direction, and the maximum value reaches 68.35 °C at the outer radius (near the trailing edge). Figure 6(b) shows the temperature distribution on the mirror plate surface, and the temperature keep equal along the rotational direction. The mirror plate rotates at a speed of 500 rpm, although it passes through the high-temperature and low-temperature regions of the pad surface continuously during the rotation process, the temperature of the mirror plate keeps the same along the circumferential direction. This is because that the heat conduction rate of the mirror plate is much lower than its speed and the circumferential points are approximately uniformly heated. The maximum temperature of the mirror plate is 61.85 °C, which is 6.5 °C lower than the maximum temperature of the pad surface, indicating that there is a large temperature gradient along the thickness direction in the oil film.

![Figure 6. Temperature distribution in the film (unit: μm)](image)

Figure 7(a) shows the circumferential section temperature at 50% pad width, it can be seen that the oil film temperature increases gradually along the wedge-shaped oil film. Along the direction of the oil film thickness, there is a large temperature gradient near the pad surface and mirror plate surface, while it remains substantially unchanged in the middle of the oil film. Figure 7(b) shows the radial section temperature at 50% pad angle, and the temperature increases gradually along the radial direction. Be similar to the circumferential section, the temperature gradient near the pad surface and mirror plate surface is larger along the thickness direction, but the variation in the middle is slower. In addition, the radial inclination angle of the mirror plate and pad surface are basically the same, and the two surfaces are almost parallel. Due to the pad can tilt freely along the radial direction, the pad surface can tilt adaptively in the light of the oil film force. When a circumferential wedge-shaped oil film is formed, the radial inclination of the pad surface is kept parallel to the mirror plate surface in the radial direction, which offsets the radial warpage of the mirror plate surface, and ensures the formation of the dynamic pressure oil film force in the rotational direction.
4.2. Thrust Load Effect on Lubrication Performance

The thrust load of the pumped storage unit is mainly composed of the weight of the rotating components and the axial hydraulic-thrust on the runner. Rotating components include pump-turbine runner, spindle and generator rotor, which weigh about 450 t and remain constant during operation. The axial hydraulic-thrust on the runner is the superposition of the hydro-pressure on the inner surface of the runner and the hydro-pressure on the outer surface of the crown and the band. Unlike conventional turbine runners, the over-flow passage of the pump-turbine runner is narrow, long and appears flat. The band is almost perpendicular to the axis, which results in a large upward hydraulic-thrust in the clearance of the band, and the total hydraulic-thrust of the runner is reduced or even directed upwards, thereby reducing the total thrust load of the unit. The inner radius of the pump-turbine runner is 2.25 m, the outer radius is 3.97 m, the crown clearance is 37 mm, and the band clearance is 35 mm. Firstly, the CFD method is used to establish the flow model of volute, runner and draft tube. At the same time, considering the clearance flow between the crown and the band, the full 3D flow field of the unit is simulated, and the thrust load under the main operating conditions is obtained, as shown in Table 3.

Table 3. The thrust load under different operating conditions (unit: ton)

| Operating modes     | $H_{total}$ | $H_{total}/W_{rotation}$ | $F_{thrust}$ |
|---------------------|-------------|--------------------------|-------------|
| Turbine mode, No load | 122         | 27.05                    | -330        |
| Turbine mode, 180 MW | 143         | 31.68                    | -310        |
| Turbine mode, 300 MW | 190         | 42.32                    | -260        |
| Pump mode           | 301         | 66.81                    | -150        |

*The symbol ‘−’ represents the downward load, $H_{total}$ is the total hydraulic-thrust, $W_{rotation}$ is the weight of the rotating parts, $F_{thrust}$ is the thrust load.

The results show that the clearance hydraulic load at the crown and the band is much larger than the internal hydraulic load of the runner. The total hydraulic-thrust of the runner is upward, and increases with the increase of the load under the turbine mode, while reaches the maximum under the pump mode, which is 66.81% of the weight of the rotating parts. This is in line with the actual operation of the unit, which is prone to occur the lifting phenomenon during the pump start process. Because the weight of the rotating parts remains unchanged, the thrust load of the unit decreases with the increase of the load under the turbine mode and reaches the minimum under the pump mode.
In the actual operation of the unit, a temperature sensor is arranged in the pad body below 20 mm of the pad surface center to monitor the pad temperature. Table 4 shows the comparison of measured and calculated temperature at this point under different thrust loads, wherein the mirror plate speed is 500 rpm. The error is controlled within 3%, and the numerical results is reasonable.

Table 4. Comparison between the numerical and measurement results

| Thrust load, $F_{thrust}$/t | Monitor point temperature/°C | Computational temperature/°C | Error/% |
|-----------------------------|-----------------------------|------------------------------|---------|
| 150                         | 56                          | 56.05                        | 0.09    |
| 260                         | 57                          | 58.17                        | 2.06    |
| 310                         | 58                          | 59.07                        | 1.84    |
| 330                         | 58                          | 59.42                        | 2.45    |

The bearing TEHD lubrication characteristics under different thrust loads are calculated, and the key performance results are shown in Table 5, keeping the speed of 500 rpm and the oil inlet parameters are the same as Table 1. With the increase of the thrust load, when the corresponding single-pad load increases from 0.15 MN to 0.33 MN, the maximum oil film pressure increases from 1.18 MPa to 2.71 MPa, the maximum pad surface temperature increases from 65.41 °C to 70.01 °C, the minimum oil film thickness decreases from 185.23 μm to 123.52 μm, the oil flow rate at the leading side decreases from 4.75 L/s to 3.85 L/s, but the friction power increases from 1050.38 KW to 1100.76 KW. Wherein the minimum oil film thickness and the maximum pad surface temperature are important indexes affecting the lubrication performance of the thrust bearing, and are also important factors limiting the increase of the thrust bearing load of the unit.

Table 5. Influence of the thrust loads on the thrust bearing performance

| Thrust load, $F_{thrust}$/t | 150 | 200 | 260 | 300 | 310 | 330 |
|-----------------------------|-----|-----|-----|-----|-----|-----|
| Single-pad load $F$/MN     | 0.15| 0.20| 0.26| 0.30| 0.31| 0.33|
| Maximum film pressure $P_{max}$/MPa | 1.18| 1.57| 2.08| 2.43| 2.53| 2.71|
| Maximum pad surface temperature $T_{max1}$/°C | 65.41| 66.81| 68.35| 69.31| 69.53| 70.01|
| Maximum mirror plate surface temperature $T_{max2}$/°C | 60.01| 60.90| 61.85| 62.43| 62.57| 62.86|
| Minimum film thickness $H_{min}$/μm | 185.53| 163.40| 142.73| 131.31| 128.68| 123.52|
| Oil inlet flow rate $Q$/L·s⁻¹ | 4.75| 4.41| 4.11| 3.96| 3.92| 3.85|
| Friction power $P$/KW       | 1050.38| 1069.82| 1090.36| 1102.20| 1105.20| 1110.76|

4.3. Pad Coating Thickness Effect on Lubrication Performance

In order to analyse the effect of Babbitt coating thickness on the lubrication performance of the thrust bearing, the TEHD lubrication performances with Babbitt coating thickness of 0 mm, 1 mm, 3 mm, 5 mm and 7 mm are calculated respectively under the thrust load of 260 t and the speed of 500 rpm, the results are shown in Table 6. As the Babbitt thickness increases from 0 to 7 mm, the maximum oil film pressure increases from 2.07 MPa to 2.10 MPa, the maximum pad surface temperature decreases from 68.49 °C to 68.07 °C, the maximum mirror plate temperature decreases from 61.98 °C to 61.66 °C, the
minimum oil film thickness increases from 142.33 μm to 143.29 μm, the oil flow rate at the leading side increases from 4.06 L/s to 4.20 L/s, and the friction power decreases from 1042.18 KW to 1039.38 KW. Therefore, properly increasing the Babbitt coating thickness is conducive to improving the lubrication performance of thrust bearings.

| Table 6. Influence of the Babbitt thickness on the thrust bearing performance |
|-----------------------------|-----|-----|-----|-----|-----|
| Coating thickness h/mm      | 0   | 1   | 3   | 5   | 7   |
| Single-pad load F/MN        | 0.26| 0.26| 0.26| 0.26| 0.26|
| Maximum film pressure Pmax/MPa | 2.07| 2.07| 2.08| 2.09| 2.10|
| Maximum pad surface temperature Tmax1/℃ | 68.49| 68.48| 68.35| 68.23| 68.07|
| Maximum mirror plate surface temperature Tmax2/℃ | 61.98| 61.94| 61.85| 61.77| 61.66|
| Minimum film thickness Hmin/μm | 142.33| 142.51| 142.73| 143.04| 143.29|
| Oil inlet flow rate Q/L·s⁻¹ | 4.06| 4.08| 4.11| 4.15| 4.20|
| Friction power P/KW         | 1042.18| 1041.71| 1041.22| 1040.08| 1039.38|

5. Conclusions
A 3D TEHD numerical analysis of the centrally supported bidirectional thrust bearing of a pump-turbine unit is carried out by using the fluid-solid-thermal coupling method. The following conclusions are obtained:

The deformation of the thrust bearing pad and the mirror plate is mainly caused by thermal effect, with the pad surface protruding and the mirror plate warping at outer radius. In the oil film rotation direction, wedge-shaped oil film is produced mainly by tilting deformation of the pad surface. While in the radial direction, the pad surface and the mirror plate surface are almost parallel, and the influence of both on the oil film cancel each other out.

Under the turbine mode, with the increase of generating load, the thrust bearing load decreases gradually and reaches the minimum at the pump mode. With the increase of the thrust load, the minimum oil film thickness decreases rapidly, and then the maximum oil film pressure and the maximum pad surface temperature increase rapidly, which becomes important factors restricting the increase of the thrust load.

Increasing the thickness of the Babbitt coating will lead to the increase of the minimum oil film thickness and the decrease of the maximum temperature of the pad surface, thus improving lubrication performances of the thrust bearing.

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