

**Transient TEHD Analysis of a Thrust Bearing Based on Two-Way Fluid-Solid-Thermal Interaction**

Jingwei Cao¹, Liming Zhai³, Yongyao Luo¹, Zhengwei Wang¹,*

¹ Department of Energy and Power Engineering, Tsinghua University, Beijing 100084, China
² Engine Business Division, FAW Jiefang Automotive Co., Ltd., Wuxi, 214026, China
³ School of Vehicle and Mobility, Tsinghua University, Beijing 100084, China
* Corresponding author. E-mail: wzw@tsinghua.edu.cn

**Abstract.** Thrust bearing is a key component of hydroelectric generator and bears the axial load of the unit. During transient processes, such as start and stop of the unit, thrust bearing usually occurs wear phenomenon at pad surface, which affects the stable operation of the unit and reduces service life of the bearing. Based on two-way fluid-solid-thermal interaction, this paper did transient thermal-elastic-hydrodynamic (TEHD) analysis of a tilting pad thrust bearing under the pump mode. For the fluid region, the Reynolds equation and the viscosity-temperature equation are solved. For the solid region, transient heat conduction equation and basic thermal-elasticity equation are considered. The data transferred in the fluid-solid interface satisfy the fluid-solid-thermal interaction equation. The variation of the lubricant oil temperature, the oil film force, the pad surface inclination angle and axial velocity of pad and mirror plate are analysed, which provide theoretical basis for the design of bearing and the safe operation of the unit.

**Keywords.** Thrust bearing; transient analysis; thermal-elastic-hydrodynamic; fluid-solid-thermal interaction

1 Introduction

Different from the start-up process of the turbine mode, the unit usually adopts frequency conversion start under the pump mode. The static frequency conversion (SFC) start-up device is used to drive the unit from 0 speed to rated speed to realize the stable start-up of the synchronous machine. The China large-scale energy storage units have reached 300 MW and are developing towards higher capacity unit, this means higher requirement to thrust bearings to minimize the deflections duo to unit weight and thrust load[1]. Tilting pad thrust bearing is an important part of pump-turbine unit, it bears the axial load and ensure the safe operation of the unit[2]. Bearing lubrication parameters, such as minimum oil film thickness and pad deformation, are vital to estimate the bearing performance[3]. During transient processes[4, 5], bearing lubrication performance is very complex, and due to differences between turbine mode and pump mode, it is important to analyse bearing transient lubrication performance under pump mode. Therefore, bearing lubrication parameters in start-up and shutdown processes are simulated based on fluid-solid-thermal interaction method, which is important to
understand bearing transient lubrication performance and ensure the safe operation of the unit.

2 Governing equations
Considering the fluid volume force and inertial force, Reynolds averaged continuity equation, momentum equation and total energy equation are solved using ANSYS CFX. Based on the Reynolds average N-S equations \(^6\), Shear Stress Transport (SST) model \(^7\) is used as the turbulence model. Based on fluid-solid interaction method, governing equations of fluid region and solid region are solved alternately using ANSYS multi-physics coupling system MFX\(^8\). Fluid region is solved by CFX and solid region is solved by Mechanical APDL\(^8\).

2.1 Fluid region equations

2.1.1 Reynolds averaged continuity equation

\[
\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x_j} (\rho U_j) = 0
\]  

2.1.2 Reynolds averaged momentum equation

\[
\frac{\partial \rho U_i}{\partial t} + \frac{\partial}{\partial x_j} (\rho U_i U_j) = -\frac{\partial \rho u_i}{\partial x_j} + \frac{\partial}{\partial x_j} (r_i - \rho u_i u_j) + S_m
\]  

Where \( r \) is molecular stress tensor, \( S_m \) is body force, \(-\rho u_i u_j\) is Reynolds stress of incompressible fluid.

2.1.3 Reynolds averaged total energy equation

\[
\frac{\partial \rho h_{tot}}{\partial t} + \frac{\partial}{\partial x_j} (\rho U_j h_{tot}) = \frac{\partial}{\partial x_j} \left[ \lambda \frac{\partial T}{\partial x_j} - \rho u_i h \right] + \frac{\partial}{\partial x_j} \left[ U_i (r_i - \rho u_i u_j) \right] + S_e
\]

where \( \rho u_i h \) is additional turbulent flux, \( \frac{\partial}{\partial x_j} \left[ U_i (r_i - \rho u_i u_j) \right] \) is viscous effect term induced by viscous stress, \( h_{tot} \) is total enthalpy, \( h_{tot} = h + \frac{1}{2} U_i U_j + k \), \( k \) is turbulence kinetic energy, \( k = \frac{1}{2} \rho u_i u_j \).

2.1.4 Viscosity-temperature equation
The viscosity of the lubricating oil is influenced by temperature, the relationship can be described as:

\[
\mu = \mu_0 \left( \frac{20 + T_0}{20 + T} \right)^{\gamma}
\]

where \( \mu_0 \) is the dynamic viscosity of lubricant oil at \( T_0 \), and \( T \) is the absolute temperature.

2.2 Solid region equations

2.2.1 Transient heat conduction equation
For isotropic homogeneous materials, the internal heat transfer equation without a heat source can be expressed as:

\[
\Delta T = \frac{\rho c}{k} \frac{\partial T}{\partial t}
\]

where \( k \) is the solid thermal conductivity, \( \rho \) is the solid density, \( c \) is the specific heat capacity.
2.2.2 Basic thermal-elasticity equation

In the solution of elastic mechanics, there are generally two methods of the displacement method and the stress method. The displacement method is used because of its strong adaptability.

Equilibrium equation:\(\textbf{(6)}\):

\[
(\lambda + G)\nabla \theta + GV^2u_\theta + F_s = \frac{\alpha_0 E}{1 - 2\nu} \frac{\partial T}{\partial y} = \rho \frac{\partial^2 u_n}{\partial t^2}
\]

where \(\lambda\), \(G\) are the Lame constant, \(y\) is the coordinate value of the Cartesian coordinate system, \(E\) is the elastic modulus, \(v\) is the Poisson ratio, \(F\) is the volume force, \(u\) is the displacement, \(\alpha_0\) is the linear expansion coefficient of the material, \(n=1,2,3\).

2.3 Fluid-solid-thermal interaction equation

In the fluid-solid-thermal bidirectional coupling calculation, the data transferred in the fluid-solid interface need to satisfy the pressure, displacement, heat fluxes and temperature conservation:

\[
\begin{align*}
\tau_j \cdot n_j &= \tau_s \cdot n_s \\
u_j &= u_s \\
q_j &= q_s \\
T_j &= T_s
\end{align*}
\] \(\textbf{(7)}\)

where \(f\) represents the fluid, \(s\) represents the solid.

3 Model

Tilting pad thrust bearing used in this paper is consisted of 10 fan-shaped pads, the model is shown in Figure 1, the structure parameters is shown in Table 1, which is as same as reference \([10]\), and the numerical model is shown in Figure 2.

![Thrust bearing](image1.png)

**Figure 1** Thrust bearing

![Numerical model](image2.png)

**Figure 2** Numerical model

Bearing lubrication performance is monitored by monitoring points, including 4 monitoring points at pad surface (p1 ~ p4), 4 monitoring points at collar surface (c1 ~ c4), and 4 temperature monitoring points (t1 ~ t4), as shown in Figure 3. 9.

| Table 1 Structural parameters |
|-----------------------------|
| Item                        | value |
| Pad outer radius R1/mm      | 1335  |
| Pad inner radius R2/mm      | 775   |
Solid226 Structural-Thermal element is used in solid domain, each node contains 4 degrees of freedom ($U_x$, $U_y$, $U_z$ and Temperature). Oil tank supports the pad and the pad can tilt freely in circumferential and radial direction, thus Link180 element is used to simulate the support. All the external surfaces of the pad and the working surface of the mirror plate are set as fluid-solid interaction surfaces, which are used to exchange physical quantities such as temperature, heat flow, pressure and displacement with the lubricating oil field. The displacement of the surfaces on both sides of the collar is coupled to keep same deformation. Because the collar rotates rapidly, its circumferential temperature is almost equal, so its circumferential temperature degrees of freedom is coupled.
The distribution of temperature on pad surface and collar surface are shown in Figure 5. It can be seen that the high temperature zone located at the surrounding of the oil hole, and the pad surface has the same distribution with the collar surface. This is because of the compression of clearance between the pad surface and the collar surface, the high-pressure oil flows with high velocity and produces a large amount viscous heat. Due to the collar does not rotate, the heat is not circumferentially consistent distribution.

The deformation on pad surface and collar surface are shown in Figure 6. The pad surface protrudes at outer radius position, and the collar surface is mainly radial tilting.
4.2 Start-up Process

In this section, thermal-elastic-hydrodynamic numerical simulation of the thrust bearing during start-up process is carried out. The unit start from the static pressure state of high-pressure oil jacking at 1500 s, the relationship of rotational speed and high-pressure oil flow rate with time is shown in Figure 7. When the rotational speed increases to 90% \( \omega_{\text{rated}} \) (1770 s), the high-pressure oil stopped and the unit runs stably to 4000 s. The time step is 5 s, and the static pressure characteristic result of high-pressure oil jacking stage is taken as the initial condition of transient calculation. Table 2 shows operating parameters during start-up.

**Table 2 Operating parameters during start-up**

| Parameter                  | Value  |
|----------------------------|--------|
| Rotational speed \( \omega/\text{r} \text{ min}^{-1} \) | 0–500  |
| Speed-up time \( t/\text{s} \)            | 300    |
| Oil flow rate \( Q/\text{kg \cdot s} \)      | 1.5    |
| Oil pressure \( P/\text{bar} \)               | 80     |
| Oil temperature \( T/\text{ºC} \)             | 25     |
| Oil stop point 90% \( \omega_{\text{rated}} \) |        |

Figure 8 shows variation of bearing parameters with time during start-up. Figure 8 (a) shows the oil film force first decreases slightly with the increase of rotational speed, and then increases gradually. Finally, it’s higher than external load. This is mainly due to the slow rise of the rotational speed, the flow field disturbance caused by the rotation of the mirror plate has little impact on the hydrostatic flow field of the oil film, thus the oil film force slightly reduced. Then, the hydrodynamic oil film force gradually increases with the increase of the rotational speed, making the total oil film force higher than external load.

Figure 8 (b) shows due to the slowly variation of oil film force, the axial speed at each monitoring point has small disturbance. When the high-pressure oil stops, there is a significant disturbance due to the sudden decrease of the oil film force.

Figure 8 (c) shows the circumferential inclination of the pad surface increases with the increase of the
rotational speed, decreases slightly after 90% of the rated speed, and then rebounds until reach a stable value. The main reason is that after the unit is started, the dynamic oil film force formed at the leading side increases the inclination angle of the pad. However, because of the slow speed rise, an obvious dynamic pressure area has been formed near the trailing side when the rotational speed reaches 90% \( \omega_{\text{rated}} \), which makes the circumferential inclination angle slightly decreases under the influence of dynamic oil film force when the high-pressure oil suddenly stops.

Figure 8 (d) shows that the oil temperature increases gradually, and the maximum temperature increases from 40.3 \( ^\circ \text{C} \) to 62.5 \( ^\circ \text{C} \) at the rated speed and finally stabilizes at 74.3 \( ^\circ \text{C} \) at 4000 s. The temperature difference between pad surface and pad body increases rapidly in the speed-up process, and also produces significant transient thermal deformation.

![Figure 8](image-url)  
**Figure 8** Variation of bearing parameters during start-up

4.3 Shutdown Process
In this section, thermal-elastic-hydrodynamic numerical simulation of the thrust bearing during shutdown process is carried out. The unit stop from the stable operation condition at 4000 s, the relationship of rotational speed and high-pressure oil flow rate with time is shown in Figure 9. When the rotational speed is reduced to 90% $\omega_{\text{rated}}$ (4007.2s), start the high-pressure oil system to ensure sufficient lubrication of the bearing. Then the electric brake is started at 50% $\omega_{\text{rated}}$, and the mechanical brake is started at 5% $\omega_{\text{rated}}$. Table 3 shows operating parameters during start-up.

| Parameter           | Value     |
|---------------------|-----------|
| Rotational speed $\omega/\text{r} \cdot \text{min}^{-1}$ | 500-0     |
| Speed down time $t$/s | 77        |
| Oil flow rate $Q_s$/kg $\cdot$ s$^{-1}$    | 1.5       |
| Oil pressure $P_s$/bar       | 80        |
| Oil temperature $T_s$/ºC     | 25        |
| Oil start point            | 90% $\omega_{\text{rated}}$ |

Figure 10 shows the variation of bearing parameters with time during shutdown process. Figure 10 (a) shows that the hydrodynamic oil film force decreases gradually. At 90% $\omega_{\text{rated}}$ (4007.2s), the sudden injection of high-pressure oil makes the oil film force rapidly increases to about 7.5 MN, and the oil film enters the dynamic and static pressure mixing stage. As the speed continues to decrease to 0 (4077 s), the oil film completely enters the static pressure stage.

Figure 10 (b) shows the high-pressure oil injection induces the disturbance on the pad surface of pad and mirror plate suddenly increases, but rapidly decreases. But the disturbance lead to the axial speed rebounds slowly and increases to the maximum when the speed drops to 0. Finally, the axial speed gradually stabilizes to 0. Meanwhile, the circumferential inclination of the pad surface decreases as the rotational speed decreases, but the injection of high-pressure oil leads to slightly increases, as shown in Figure 10 (c).

Figure 10 (d) shows the temperature of the pad surface decreases slightly with the rotational speed decreases, but there was a jump rise at 90% $\omega_{\text{rated}}$ (4007.2s). As the rotational speed continues to decrease, the viscous friction power consumption gradually decreases, and the temperature also decreases rapidly. However, when the rotational speed drops to 0 (4077 s), the temperature decreases slowly. The temperature of t1 and t3 on the surface decrease rapidly to the below the temperature of t2 and t4 respectively, the thermal deformation generated offsets the previous bulge caused by the high temperature of pad surface, which is beneficial to avoid the wear of pad surface caused by large thermal-
elastic deformation.

Figure 10 Variation of bearing parameters during shutdown

5 Conclusions

In this paper, bearing transient lubrication parameters, including oil film force, axial speed, circumferential inclination angle and temperature, are analysed using fluid-solid-thermal interaction method.

During start-up and shutdown process, the high-pressure oil is injected to keep bearing lubrication and to help bearing to dissipate heat. In start-up process, there is a large disturbance after the oil is stopped, which is detrimental to the safe operation of the unit. During shutdown process, it is obvious that the oil helps changing the temperature difference between pad surface and pad body, and the generated thermal deformation offsets pad deformation caused by pad surface high temperature.

However, this paper uses a periodic cyclic symmetry model to simulate the bearing performance, the characteristics of aperiodic symmetry and uneven load on pads of the actual bearing operating condition are ignored. Therefore, it will be very meaningful to use the full 3D model of thrust bearing for numerical study.

6 References

[1] Huang B, Wu Z D, Wu J L, et al. Numerical and experimental research of bidirectional thrust bearings used in pump-turbines[J]. ARCHIVE Proceedings of the Institution of Mechanical Engineers Part J Journal of Engineering Tribology 1994-1996 (vols 208-210), 2012, 226(9): 795-806.
[2] Wodtke M, Schubert A, Fillon M, et al. Large hydrodynamic thrust bearing: Comparison of the calculations and measurements[J]. Proceedings of the Institution of Mechanical Engineers, Part J: Journal of Engineering Tribology, 2014, 228(9): 974-983.

[3] Liang X, Yan X, Ouyang W, et al. Thermo-Elasto-Hydrodynamic analysis and optimization of rubber-supported water-lubricated thrust bearings with polymer coated pads[J]. Tribology International, 2019, 138: 365-379.

[4] Monmousseau P, Fillon M. Transient thermoelastohydrodynamic analysis for safe operating conditions of a tilting-pad journal bearing during start-up[J]. Tribology International, 2000: 225–231.

[5] Pap B, Fillon M, Guillemont M, et al. Experimental and Numerical Analysis on the Seizure of a Carbon-Filled PTFE Central Groove Journal Bearing during Start-Up Period[J]. Lubricants, 2018, 6(1).

[6] J.S.Rao. Simulation Based Engineering in Fluid Flow Design[M]. Gewerbestrasse 11, 6330 Cham, Switzerland: Springer Nature, 2017: 159-166.

[7] Menter F R. Review of the shear-stress transport turbulence model experience from an industrial perspective[J]. International Journal of Computational Fluid Dynamics, 2009, 23(4): 305-316.

[8] Zhai L, Wang Z, Luo Y, et al. TEHD analysis of a bidirectional thrust bearing in a pumped storage unit[J]. Industrial Lubrication and Tribology, 2016, 68(3): 315-324.

[9] Guo Z. Research on Methodology of Thermal-Flow-Elastic Coupling Numerical Simulation in Air-Cooled Turbine with The Finite Difference[D]. Harbin Institute of Technology 2009.

[10] Zhai L, Luo Y, Liu X, et al. Numerical simulations for the fluid-thermal-structural interaction lubrication in a tilting pad thrust bearing[J]. Engineering Computations, 2017, 34(4): 1149-1165.