

Article

Deformation Simulation Analysis of Mechanical Seal with Superconducting Magnetic Assistance Under Multi-Field Coupling and Its Influence on Lubrication Performance

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Abstract

This study investigates the influence of thermoelastic deformation on the lubrication performance of a superconducting magnetically assisted turbopump mechanical seal, with particular emphasis on film thickness and leakage. A spiral-groove hydrodynamic lubrication model based on the Reynolds equation and film energy equation is used to calculate the liquid-film load capacity and leakage, while a superconducting magnetic-force model based on the Bean model and electromagnetic finite-element analysis is employed to determine the magnetic-force contribution. The calculated mechanical loads and thermal boundary conditions are subsequently introduced into a thermoelastic finite-element model to obtain the deformation of the rotating and stationary seal rings. The results show that increasing the sealing clearance causes both the superconducting magnetic force and hydrodynamic liquid-film force to decrease nonlinearly. At a fixed clearance, the magnetic force is nearly independent of rotational speed, whereas the liquid-film force increases with increasing speed. The rotating ring exhibits greater deformation than the stationary ring, resulting in a non-parallel, approximately conical seal face. The deformation-induced change in the seal-face geometry is incorporated into the film-thickness model using the minimum and maximum film thicknesses, h_{min} and h_{max} . The results show that considering seal-face deformation leads to a smaller predicted minimum film thickness and a larger leakage rate than those obtained from the undeformed model. These findings indicate that thermoelastic deformation should be considered in the lubrication-performance evaluation of superconducting magnetically assisted mechanical seals, particularly under low-speed operating conditions.

Keywords: turbopump; one-way sequential coupling; lubrication performance; simulation analysis; minimum film thickness



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1. Introduction

The turbopump is the heart of the liquid rocket engine, which ensures that the low-temperature liquid fuel and oxidant are transported in an orderly manner according to the required pressure and flow [1,2]. The mechanical seal is an important functional component of a turbopump, which has the characteristics of reliable operation and long life. The fluid lubrication film between the stationary ring and the rotating ring of the mechanical seal resists the axial force and axial vibration of the system, which greatly reduces the occurrence of rubbing [3]. As liquid rocket engines evolve toward high-parameter operating conditions, the increase in operating parameters exacerbates wear between sealing surfaces, significantly impacting sealing stability [4]. Therefore, sealing reliability is particularly important.

Thermoelastohydrodynamic lubrication simulation methods have been widely applied in the analysis of sealing performance. Shahin et al. [5] carried out numerical simulation and experimental research on various end-face grooves under high pressure and high speed, indicating that it is feasible to form a stable full-film lubrication at high pressure and high speed. Djamaï et al. [6] developed a thermohydrodynamic numerical model for mechanical face seals to investigate the thermal and lubrication behavior of the fluid film. Brunetière and Modolo [7] investigated heat transfer in mechanical face seals and analyzed the thermal behavior of the sealing interface. Ayadi et al. [8] conducted experimental and numerical investigations on the lubrication regimes of liquid mechanical seals. Sua et al. [9], based on the finite element method and multiphysics software programming, developed a thermo-fluid–solid coupled model for the numerical simulation of mechanical seals, analyzing and discussing the sealing performance, thermal fields, deformations, and stress distributions of the solids. Zhao et al. [10] established the thermal-fluid–solid coupling simulation model of the mechanical seal of the ultra-high-speed gearbox and studied the temperature rise and deformation of the radial double-end mechanical seal end face of the ultra-high-speed transmission system. Zhang et al. [11] analyzed the extreme working conditions, such as high speed and low-viscosity medium lubrication, of the dynamic and static combined mechanical seal of a high-speed turbopump and completed the optimization of the key structural parameters of the mechanical seal. Lebeck et al. [12] considered the waviness, taper, and roughness of the mechanical seal end face and established a mechanical seal leakage model. Migout et al. [13] investigated fluid-film vaporization in a mechanical face seal using a transient two-phase numerical model and analyzed the influence of operating conditions on seal behavior. Xie et al. [14] realized the coupled solution between film pressure, film thickness, temperature, and deformation, and studied the influence of rotational speed, sealing pressure, and medium temperature on the deformation of the end face of the upstream pumping mechanical seal. Ran et al. [15] considered the influence of thermo-fluid–solid interactions on sealing performance and compared the performance of different models under various operating conditions, providing a theoretical basis for selecting suitable mechanical seal simulation models. Xiang et al. [16] studied a reciprocating rod seal with textured rods, where the design of the texture significantly improved sealing performance and reduced leakage rates. Ben-Abdelwahed et al. [17] showed that small-sized seals under low pressure and low Reynolds numbers in liquid pumps of turbo-jet engines exhibit more complex thermal deformation behavior. Recent studies have also demonstrated the importance of tribological and thermal-performance regulation under complex operating conditions, indicating that the coupled effects of lubrication, friction, and thermal behavior should be carefully considered in the performance evaluation of advanced tribological systems [18].

High-temperature superconducting magnetic levitation bearings have shown great industrial application prospects due to their advantages, such as passive self-stability,

non-contact operation without mechanical friction, and high rotational speed [19]. The combination of superconducting bearings with fluid bearings allows for complementary advantages, meeting operational conditions during start-stop phases, while ensuring stable load-bearing performance [20,21]. In our previous study [22], a superconducting magnetically assisted mechanical seal was proposed, and the numerical analysis mainly focused on the seal design, the combined contributions of the superconducting magnetic force, and the hydrodynamic liquid-film force to the supporting performance of the seal. However, the thermoelastic deformation of the rotating and stationary seal rings and the resulting change in seal-face geometry were not included in the previous model. Therefore, based on the previously proposed seal configuration, the present study further investigates the combined effects of the magnetic, hydrodynamic, and thermoelastic fields within a sequential multiphysics framework. Before the subsequent performance and deformation analyses, the hydrodynamic liquid-film model and superconducting magnetic-force model are validated against experimental measurements. The thermoelastic deformation of the rotating and stationary rings is then analyzed, and the deformation-induced non-parallelism of the seal faces is incorporated into the film-thickness model through the minimum and maximum film thicknesses, h_{min} and h_{max} . In this way, the effects of seal-face deformation on the minimum film thickness and leakage are quantitatively evaluated.

2. The Structure and Working Conditions of a Magnetic-Liquid Composite Mechanical Seal

The superconducting, magnetically assisted mechanical-seal configuration investigated in the present study was originally proposed in our previous work. The present study retains this basic seal configuration and further focuses on thermoelastic deformation of the seal rings and its influence on lubrication performance. The low-temperature sealing medium provides a favorable cryogenic environment for the superconducting blocks. The rotating ring incorporates the permanent-magnet component, whereas the stationary sealing ring is made of graphite. The superconducting blocks are embedded in the graphite stationary ring, and the graphite–superconductor assembly is mounted in a stainless-steel supporting substrate. Therefore, the permanent magnet, graphite, superconducting blocks, and stainless-steel substrate are distinct structural components of the seal, rather than alternative material definitions of the same sealing ring. Their respective locations are illustrated in Figure 1.

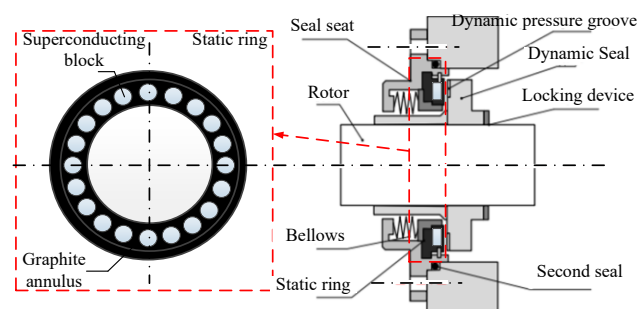


Figure 1. Structural diagram of mechanical seal with superconducting block.

The working principle diagram of the rocket engine turbine pump is shown in Figure 2a. The main function of the turbine pump is to transport the propellant to the main thrust chamber to generate gas and generate thrust at high speed from the nozzle. The structure diagram of the mechanical seal of the turbine pump is shown in Figure 2b.

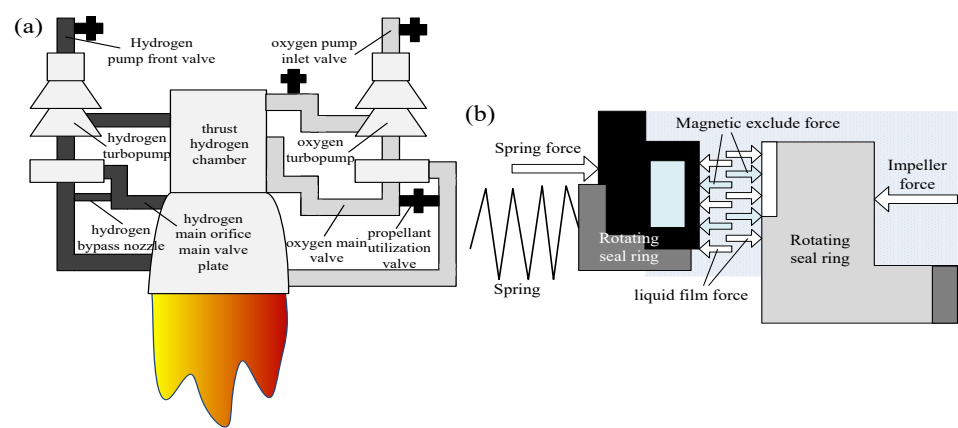


Figure 2. (a) The turbine pump working principle diagram; and (b) the working principle diagram for the rocket engine turbine pump.

The superconducting elements used in the present seal are cylindrical YBCO bulk superconductors. Sixteen superconducting blocks are embedded in the stationary ring, with each block having a diameter of 5 mm and a thickness of 10 mm. The permanent-magnet component of the rotating ring is made of NdFeB. The magnetic force generated by the sixteen superconducting blocks is obtained by first calculating the force of a single cylindrical superconducting block based on its B–H constitutive relation under the action of the NdFeB permanent magnet, and then superimposing the contributions of all of the superconducting blocks. The basic structural parameters of the superconducting mechanical seal are summarized in Table 1.

Table 1. Basic structural parameters of superconducting mechanical seals.

Parameter	Value	Parameter	Value
Outer diameter of rotating ring/mm	60	Groove depth $h_g/\mu\text{m}$	5
Inner diameter of rotating ring/mm	40 mm	Number of rotating ring grooves	15
Sealing surface width	10 mm	Superconducting block diameter/mm	5
Radius of groove bottom circle/mm	49.5	Superconducting block thickness	10
Slot width ratio	0.5	Number of superconducting blocks	16
helix angle/ $^\circ$	15		

The working condition parameters of the mechanical seal that is used for the turbopump are listed in Table 2. The sealing medium is liquid oxygen. Because the temperature of the liquid oxygen increases as it passes through the sealing region, the operating temperature of the seal is higher than $-160\text{ }^\circ\text{C}$. The sealing medium is liquid oxygen, and its actual temperature under the investigated operating condition is approximately $-160\text{ }^\circ\text{C}$.

Table 2. Turbine pump mechanical seal working condition parameters.

Parameter	Value	Parameter	Value
Rotor speed n/r/min	0~30,000	Pressure difference/MPa	0.5~5.0
Working temperature/ $^\circ\text{C}$	>-160	Axial load/kN	5~10
Medium viscosity/ $10^{-3}\text{ Pa}\cdot\text{S}$	0.6~1.5	Medium temperature/ $^\circ\text{C}$	-160

3. Force Analysis of Mechanical Seal with Superconducting Magnetic Force in Operation

The superconducting magnetically assisted mechanical seal will receive the action of multiple physical fields in its operation. The multiple physical fields include the fluid field,

magnetic field, stress field, and temperature field. Before carrying out the deformation analysis of the mechanical seals, it is necessary to analyze the stress of the mechanical seals in multiple physical fields.

3.1. Force Analysis Model of Mechanical Seal with Superconducting Magnetic Auxiliary

The hydrodynamic lubrication model and superconducting magnetic-force model employed in this section are based on the framework established in our previous study. The present study adopts a sequential multiphysics coupling strategy. First, the hydrodynamic liquid-film force and superconducting magnetic force are calculated separately, using the corresponding lubrication and electromagnetic models. The resulting mechanical loads, together with the prescribed thermal boundary conditions, are then transferred to the thermoelastic finite-element model to calculate the deformation of the rotating and stationary seal rings. The calculated deformation is subsequently used to correct the film geometry through the minimum and maximum film thicknesses, $h_{\min}$ and $h_{\max}$. The deformation-corrected film geometry is subsequently fed back into the hydrodynamic lubrication model, and the Reynolds equation is re-solved to obtain the updated film pressure, load-carrying capacity, and leakage rate. Therefore, the present model is sequentially coupled rather than fully coupled. The overall analysis procedure is shown in Figure 3.

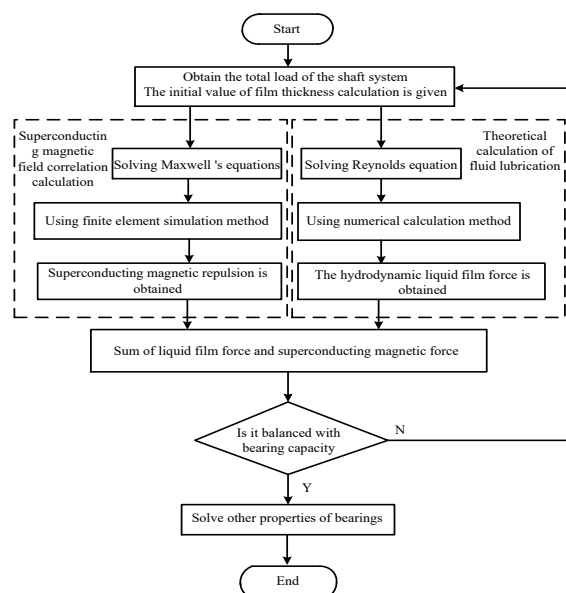


Figure 3. Superconducting mechanical seal performance analysis process.

In this paper, the method for calculating the lubrication performance of the mechanical seals is based on the theory of fluid lubrication. The core calculation model is the modeling of bearing capacity and the leakage of mechanical seals with spiral grooves using the Reynolds equation and the energy equation. This method has been used in many studies and is a relatively simple classical digital modeling method. Figure 4 is the spiral groove structure diagram of the end face of the mechanical seal rotating ring studied in this paper.

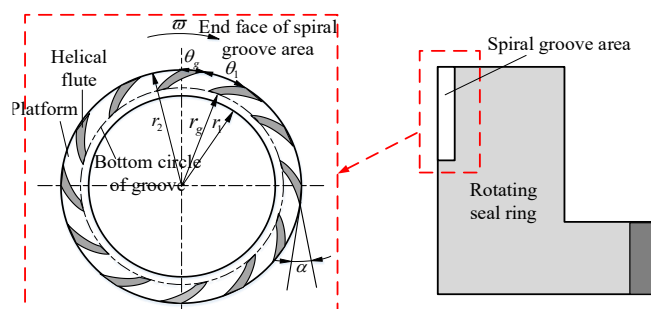


Figure 4. Spiral groove structure drawing of rotating ring.

The polar coordinate form of the generalized Reynolds equation of the spiral groove seal is shown in Equation (1):

$$\frac{\partial}{\partial r} \left(\frac{\rho r h^3}{12\mu} \frac{\partial p}{\partial r} \right) + \frac{\partial}{\partial \theta} \left(\frac{\rho h^3}{12\mu r} \frac{\partial p}{\partial \theta} \right) = \frac{\omega r}{2} \frac{\partial(\rho h)}{\partial \theta} + \rho r V \quad (1)$$

Considering the effect of superconducting magnetic force F_M , and under the combined action of F_M and the liquid-film force F_L , the stationary ring and the rotating ring of the mechanical seal are in an equilibrium state. The force equilibrium of the mechanical seal is illustrated in Figure 5. At this time, the liquid-film pressure p is the difference between the axial load F and F_M of the shaft system, that is, F_L . In Equation (2), φ is the projection angle of the angle between the rotating ring axis and the z -axis on the yz plane, γ is the projection angle of the rotating ring axis and the z -axis on the xz plane, and h_g is the depth of the spiral groove. When there is a spiral groove with a depth of h_g on the rotating ring, $h_g = 0$ in the non-groove area and $h_g = h_e$ in the groove area.

$$h = h_0 - \varphi r \cos \theta - \gamma r \sin \theta + h_g \quad (2)$$

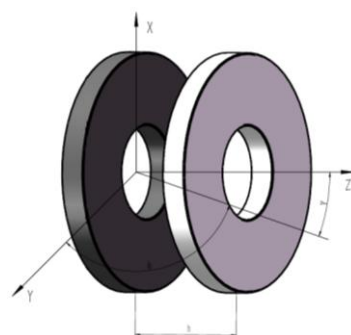


Figure 5. Mechanical seal equilibrium position force diagram.

When the liquid film is formed, the frictional resistance distance of the end face of the sealing ring can be obtained by integral of Equation (3):

$$M_f = \int_{r_1}^{r_2} \int_0^{2\pi} \left(\frac{h \partial p}{2r \partial \theta} + \frac{r}{h} \right) r^2 dr d\theta \quad (3)$$

$$Q = \int_0^{2\pi} \int_{r_1}^{r_2} \frac{h^3}{12\mu} \frac{\partial p}{\partial r} r dr d\theta \quad (4)$$

The bearing capacity F of the liquid film is obtained by integrating the liquid-film pressure p on the end face area.

$$F = \iint_{\Omega} p r dr d\theta \quad (5)$$

The magnetic properties of superconductors can be analyzed theoretically and experimentally. The magnetic calculation of a single superconducting block is the basis of the electromagnetic force calculation of the superconducting seal. The B-H and E-J electromagnetic constitutive relations of the superconductor are calculated, and the bearing capacity and stiffness provided by a single superconducting block are obtained. The calculation results of each superconducting block are superimposed. The electromagnetic properties of superconductors can be described by Maxwell's equations, as shown in Equation (6), where D is the electric displacement vector, ρE_e is the charge density, E is the electric field strength, B is the magnetic induction strength, H is the magnetic field strength, and J is the current density.

$$\left. \begin{aligned} \nabla D &= \rho E_e \\ \nabla \times E &= -\partial B / \partial t \\ \nabla B &= 0 \\ \nabla H &= J + \partial D / \partial t \end{aligned} \right\} \quad (6)$$

In general, the superconducting magnetic properties of superconductors are mainly determined by the external electromagnetic field and the internal induced current I . The superconducting current I is determined by the magnetic induction intensity B , that is, $I \propto E$, $D \times E = -\partial B / \partial t$, and B can be obtained by using Equation (7), where μ_m is the permeability of the medium.

$$B = \mu_m \bullet H \quad (7)$$

To obtain the differential equation of the static magnetic field, the magnetic vector potential A is introduced. The magnetic flux density B is expressed as the curl of A , as given in Equation (8).

$$B = \nabla \times A \quad (8)$$

The static magnetic field differential equation can be obtained by the simultaneous Equations (6)–(8), as shown in Equation (9), where J_m is the magnetization current of the permanent magnet, J_c is the critical current density of the superconductor, σ is the dielectric conductivity, and Φ is the electric scalar vector.

$$\nabla \times \left(\frac{1}{\mu_m} \nabla \times A \right) = J_m + J_c = J \quad (9)$$

$$J_c = \sigma (\partial A / \partial t + \nabla \Phi) \quad (10)$$

When calculating the critical current density of superconductors, there are many models to choose from, such as the Bean model, Kim model, exponential model, etc. In this paper, the most commonly used Bean model is selected. Let J_c be a fixed value, the energy function of the static magnetic field is shown in Equation (11), where Γ is the magnetic field boundary, n is the outer normal unit vector of the boundary, and Ω is the magnetic field region. In order to use the finite element solution, Equation (11) can be transformed into a conditional variable problem, such as in Equation (12), where A_0 is a known quantity.

$$\begin{aligned} f(A) &= \int_{\Omega} \left(\int_0^B H \cdot dB \right) d\Omega - \int_{\Omega} \left(\int_0^A J \cdot dA \right) d\Omega \\ &\quad - \int_{\Gamma} \left(\int_0^A \frac{1}{\mu_m} \nabla \times A \times n \cdot dA \right) d\Gamma \end{aligned} \quad (11)$$

$$\left. \begin{aligned} \Omega : \min f(A) &= \int_{\Omega} \int_0^B H dB d\Omega - \int_{\Omega} \int_0^A J dA d\Omega + \\ &\quad \int_{\Gamma_1} \int_0^A (\lambda A - q) dA d\Gamma \\ \Gamma_1 : A &= A_0 \end{aligned} \right\} \quad (12)$$

The superconducting material used in the present study is YBCO, and the permanent-magnet component is NdFeB. The electromagnetic model follows the framework established in our previous study, and is used here to obtain the magnetic force–clearance relationship. In the Bean model, the critical current density J_c is treated as a constant parameter. The temperature- and magnetic-field dependence of the superconducting properties is not explicitly considered in the present model.

Based on the above model, the magnetic field distribution is obtained by using electromagnetic field finite element analysis software, and then the magnetic levitation force of the superconducting tile is obtained by using the space integral method, that is, the bearing capacity, as shown in Equation (13).

$$F = \int_V J \times B dV \quad (13)$$

Considering the thermal effect of the liquid film, the energy equation is expressed as shown in Equation (14).

$$\rho c_F \left(u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} + w \frac{\partial T}{\partial z} \right) = k_F \left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} \right) + s \quad (14)$$

This equation describes the temperature distribution of the lubricating film considering heat convection, heat conduction, and viscous dissipation.

3.2. Simulation Model Validation

In order to ensure the correctness of the simulation results, the simulation model is verified before the simulation. The liquid-film force and superconducting magnetic force were verified by experiments. The test of the liquid-film force was carried out on the mechanical seal test bench, which is shown in Figure 6. The test was driven by a turbine, and the test speed was 30,000 r/min. Through the test, the opening force of the sealing end face can be obtained; that is, the liquid-film force F_y , the sealing clearance H_0 , the inlet pressure P_1 , the outlet pressure P_2 , the leakage Q , and the end face temperature T .

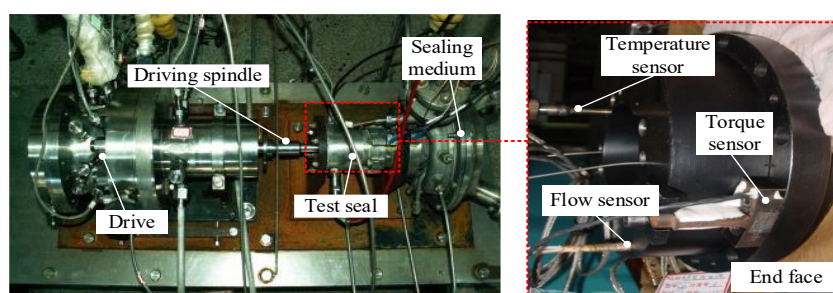


Figure 6. The mechanical seal test bench constructed with reference to the experimental setup described in [1].

To evaluate the repeatability of the experiment, each test was repeated three times under the same operating conditions. The differences among the three repeated measurements were within 5%, indicating good experimental repeatability. The data presented in Figure 7 are the average values obtained from the three repeated tests. The data obtained through the experiment are shown in Figure 7a–c. The test was instantaneously accelerated to 30,000 r/min, and the power was cut off when stable operation was reached after 200 s, and the free deceleration was stopped. Figure 7a shows the change in F_y and H_0 with the test time. It can be seen that when the F_y measured in the stable operation stage is about 2.1 kN, H_0 is about 8 μm , which is relatively stable in this stage. After deceleration, both F_y and H_0 decrease until it decreases to 0. It can be seen from Figure 7b that the sealing inlet

pressure P_1 , that is, the pressure of the sealed medium, remains at about 0.9 MPa during the test, and P_1 increases to 1.35 MPa after the speed decreases to 0. The sealing outlet pressure P_2 is basically stable at 0.1~0.2 MPa during the test, and suddenly increases to 0.4 MPa at 100 s, which may be caused by the internal pressure change during the medium gasification process. In Figure 7c, it can be seen that the leakage of the seal in the stable operation stage is relatively stable, the Q is 4.5~8.0 g/s, and the Q is also reduced to 0 after the speed is reduced to 0, because the sealing stationary ring and the rotating ring are closed. The end-face temperature T was measured during the experiment. As shown in Figure 7c, the minimum measured temperature is approximately $-160\text{ }^{\circ}\text{C}$, corresponding to the liquid-oxygen temperature. As the liquid oxygen passes through the sealing region, heat transfer between the sealing medium and the seal faces causes the local end-face temperature to gradually increase.

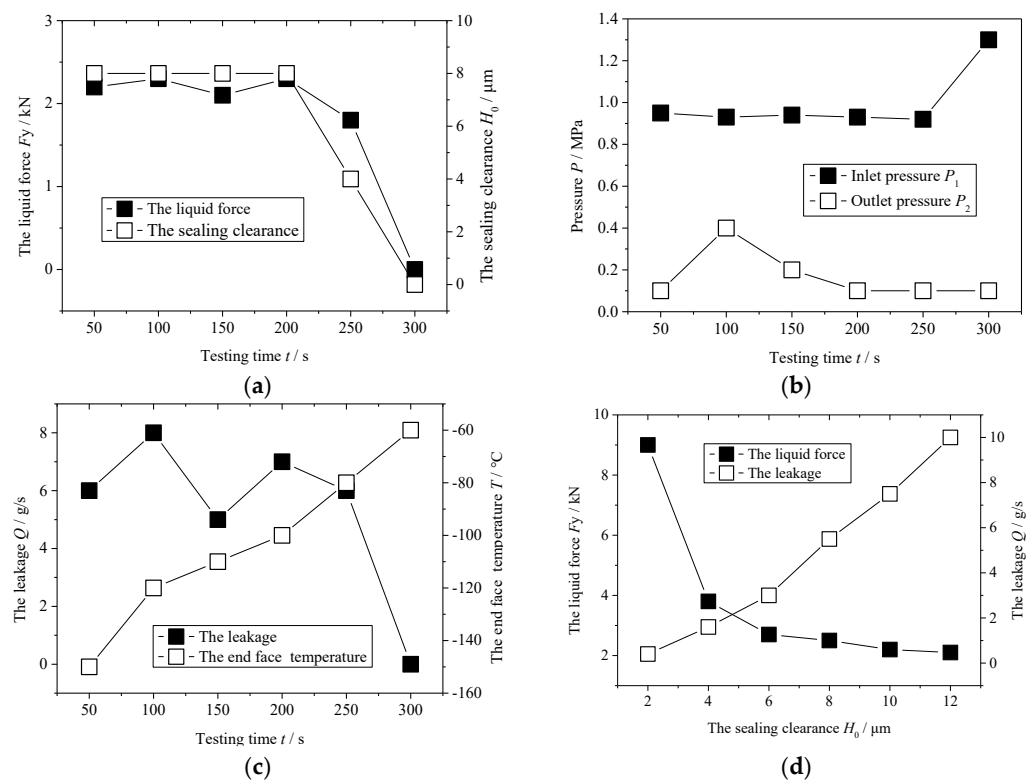


Figure 7. (a) The change in F_y and H_0 with the test time; (b) the change in P_1 and P_2 with the test time; (c) the change in Q and T with the test time; and (d) the change in F_y and Q with the H_0 obtained by simulation.

Figure 7d is a combination of the theoretical simulation results given by the test conditions. The simulation results show that the liquid-film force gradually decreases with the increase in the sealing gap, and the leakage amount gradually increases proportionally with the increase in the sealing gap. Compared with the experimental results, it can be found that H_0 is 8 μm , F_y is about 2.1 kN, and Q is 4.5~8.0 g/s; in the simulation results, when H_0 is 8 μm , F_y is 2.5 kN, and Q is 5.5 g/s. At a sealing clearance of $H_0 = 8\text{ }\mu\text{m}$, the experimentally measured liquid-film force is approximately 2.1 kN, whereas the simulated value is approximately 2.5 kN, corresponding to a relative error of approximately 19.05%. The experimentally measured leakage rate ranges from 4.5 to 8.0 g/s, while the simulated leakage rate is approximately 5.5 g/s, which falls within the experimental range. Considering the repeatability of the three independent tests, the numerical model shows reasonable agreement with the experimental measurements under the investigated operating conditions.

The superconducting magnetic test platform is shown in Figure 8. The test bench can realize different relative displacements between the superconductor and the permanent magnet by adjusting the positioning device and measuring the corresponding magnetic force through the force sensor installed on the device. The circular superconducting block is installed on the fixture, and the thrust disc is made of permanent magnets. Liquid nitrogen is injected into the test chamber during the test. The correctness of the theoretical calculation model can be investigated by using this test device. The displacement sensor can measure the distance between the superconductor and the permanent magnet. The force sensor can measure the superconducting magnetic force.

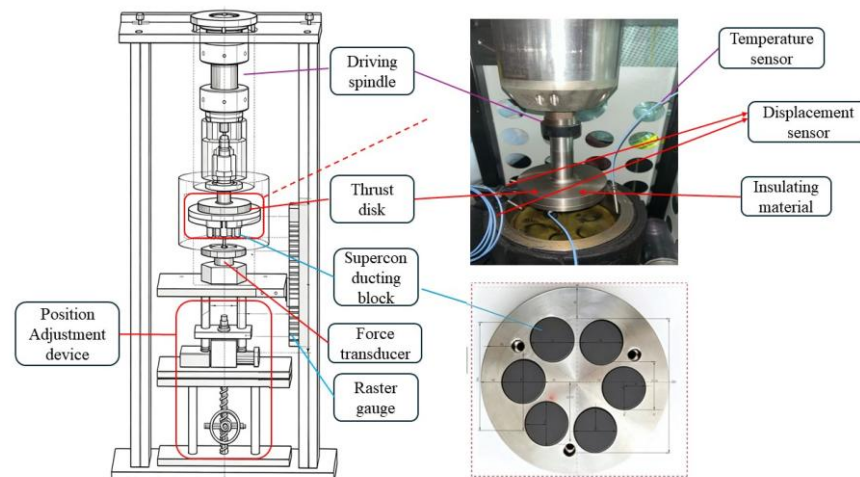


Figure 8. The superconducting magnetic-force test platform based on the experimental setup described in [23,24].

The comparison between the superconducting repulsive force under different gaps obtained by theoretical calculation and the superconducting repulsive force obtained by experimental test is shown in Figure 9. It can be seen from Figure 9 that both the theoretical calculation value and the measured value decrease rapidly with the increase in the gap. The calculated value is 18 N larger than the experimental value when the gap is 2 μm , and the calculated value is only 5 N larger than the experimental value when the gap is 12 μm . The calculated and measured superconducting magnetic forces exhibit the same decreasing trend with increasing gap. The absolute difference between the numerical and experimental results decreases from approximately 18 N at a gap of 2 μm to approximately 5 N at a gap of 12 μm . Together with the good repeatability of the three independent tests, these results indicate reasonable agreement between the numerical predictions and experimental measurements.

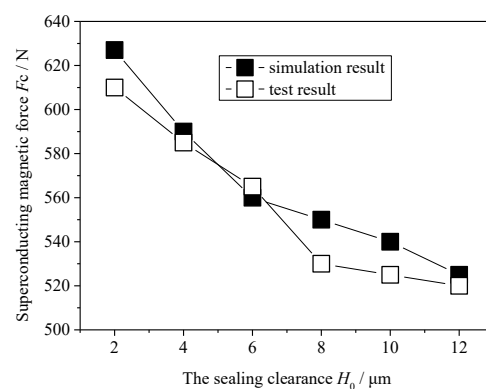


Figure 9. Comparison of simulated and experimental superconducting magnetic forces under different sealing clearances.

The superconducting magnetic-force test was also repeated three times under the same conditions. The differences among the repeated measurements were within 5%, and the data presented in Figure 9 are the average values of the three tests.

The liquid-nitrogen experiment was used to validate the overall magnetic force–clearance relationship predicted by the electromagnetic model under the investigated cryogenic condition. It should be noted that the detailed temperature-dependent superconducting properties and the differences between the liquid-nitrogen test condition and the actual turbopump thermal environment are not fully considered in the present model. These issues are regarded as limitations of the present study and will be addressed in future work.

3.3. Analysis of Simulation Results of Magnetic Fluid Conforming to Mechanical Seal Performance

The magnetic fluid obtained by simulation calculation conforms to the lubrication performance of the mechanical seal, as shown in Figure 10. Figure 10a is the change in the bearing capacity of the mechanical seal end face with the seal clearance at a rotational speed of 30,000 r/min. It can be seen from the figure that the superconducting magnetic force is not sensitive to the change in clearance but makes a certain contribution to the resultant force. Figure 10b shows the change in the bearing capacity of the mechanical seal end face with the rotational speed of the turbopump when the given load is 2.2 kN. It can be seen that the liquid-film force increases gradually with the change in rotational speed, and the superconducting magnetic force does not change. The contribution of the superconducting magnetic force to the total bearing capacity is obvious. Figure 10b shows the change in the bearing capacity of the mechanical seal end face with the rotational speed of the turbopump when the given load is 2.2 kN. It can be seen that the liquid-film force increases gradually with the change in rotational speed, and the superconducting magnetic force does not change. The contribution of the superconducting magnetic force to the total bearing capacity is obvious. Figure 10c shows the change in the thickness of the lubricating film of the mechanical seal with the rotational speed of the turbopump when the given load is 2.2 kN. It can be seen that the thickness of the lubricating film increases obviously with the increase in rotational speed, which is mainly provided by the liquid dynamic pressure effect. After increasing the superconducting magnetic force, the film thickness of the mechanical seal increases. Figure 10d shows that when the given load is 2.2 kN, the leakage of the mechanical seal changes with the rotational speed of the turbine pump. It can be seen in the figure that the leakage of the mechanical seal also increases with the increase in the rotational speed, while the leakage of the superconducting contribution does not change significantly. The leakage of the mechanical seal after the superconducting magnetic force will increase slightly.

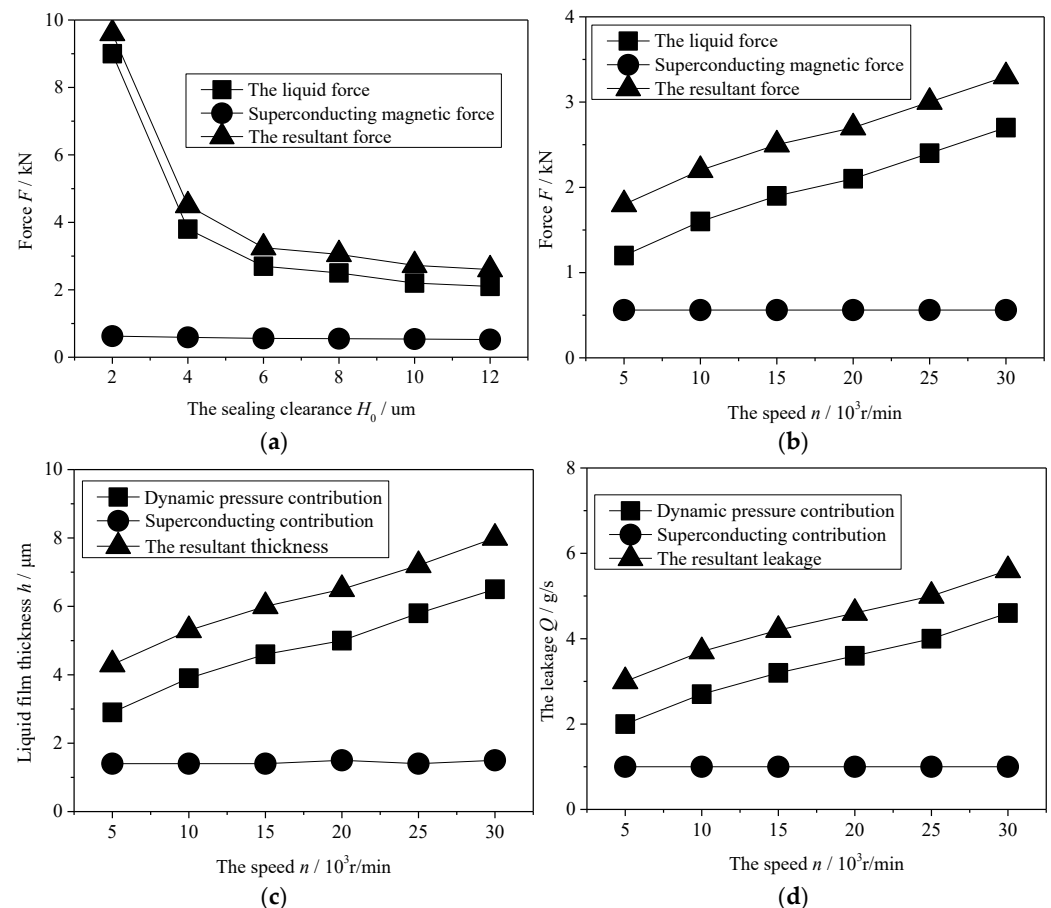


Figure 10. (a) The force capacity of the mechanical seal changes with the sealing clearance; (b) the force capacity of the mechanical seal changes with the sealing revolution speed; (c) the liquid-film thickness of the mechanical seal changes with the sealing revolution speed; and (d) the leakage of the mechanical seal changes with the sealing revolution speed.

4. Deformation Simulation Analysis of Mechanical Seal with Superconducting Magnetic Assistance Under Multi-Field Coupling

According to the structural parameters of the spiral groove mechanical seal above, the three-dimensional modeling of the rotating ring is carried out using SolidWorks 2023 software. The three-dimensional diagram of the rotating ring and the stationary ring is shown in Figure 11a,b. The section of the rotating ring is arranged with spiral grooves. In the modeling of the stationary ring, the graphite material is omitted, and the stationary ring is regarded as a metal whole. On the basis of the three-dimensional model, the coupling deformation simulation of the thermal deformation and stress deformation of the rotating ring and the stationary ring is carried out. In this paper, the ANSYS Workbench 2022R2 software is used as a tool to analyze the thermal deformation of the mechanical seal ring and the stationary ring. According to the theory of fluid lubrication and the structure and working conditions of the mechanical seal, the bearing capacity of the mechanical seal under the given working speed and working clearance is obtained by calculation, and then the calculation results are brought into the finite element model to obtain the liquid-film temperature. Finally, the size and distribution of the thermal deformation of the mechanical seal are obtained by finite element simulation. After the solid model is imported, it is necessary to specify the characteristics of the analysis object, that is, to define the unit type, which is used to simulate various structures and materials in the project. The elastic modulus and Poisson's ratio should be considered in the structural analysis, and the thermal conductivity should be considered in the thermal analysis. The relevant

parameters of the mechanical seal ring and the stationary ring are shown in Table 3. The imported model is opened from the ‘steady-state heat’ module, and the ‘engineering data’ is selected to define the material and related material coefficients of the rotating ring model. The rotating ring is modeled using the material properties of an NdFeB permanent magnet, while the stationary ring is simplified as impregnated graphite. This material configuration is consistent with the superconducting mechanical-seal structure reported in previous studies, in which the rotating-ring substrate is made of NdFeB permanent-magnet material, and the stationary ring is made of graphite with YBCO superconducting blocks embedded in it. The thermophysical properties listed in Table 3 are adopted as representative values for the thermoelastic analysis.

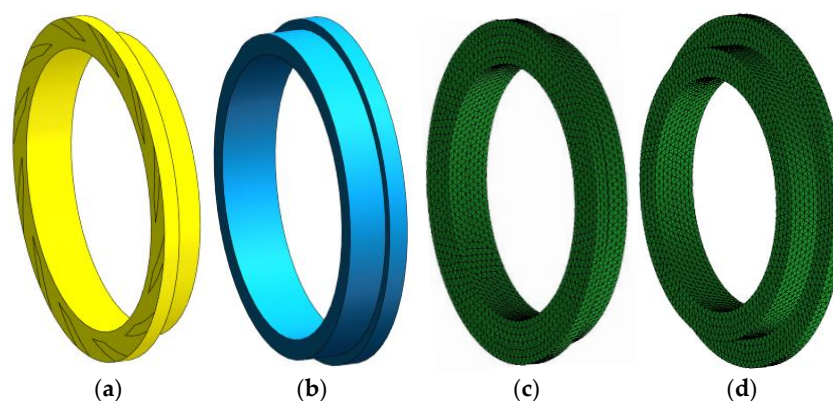


Figure 11. (a) The three-dimensional diagram of the rotating ring; (b) the three-dimensional diagram of the stationary ring; (c) the rotating ring model of the divided grid after the import; and (d) the stationary ring model of the divided grid after the import.

Table 3. Related material parameters of mechanical seals.

Parameter	Value	
	Rotating Ring	Stationary Ring
Elastic modulus/GPa	213	120
Poisson's ratio	0.254	0.270
Coefficient of linear expansion/ 10^{-6} K^{-1}	11.8	4.8
Thermal conductivity/ $\text{W/m}^{-1} \cdot \text{K}^{-1}$	20.0	10.8
Specific heat capacity/ $\text{J} \cdot \text{kg}^{-1} \cdot \text{K}^{-1}$	34.6	48.0

The thermophysical properties of the graphite material were determined from previous experimental measurements, including the coefficient of thermal expansion, specific heat capacity, thermal diffusivity, and thermal conductivity.

For the meshing part, the mesh module was selected, and the size of the solid mesh is defined as 0.5 mm. Since there is no problem of excessive proportion, the meshing of the rotating ring can be divided by the commonly used hexahedral mesh or hybrid mesh. In order to facilitate the simulation calculation of the temperature field of the sealing ring, the following assumptions are made: (1) There is an ideal horizontal gap between the end faces of the sealing ring, and the heat is evenly distributed on the surface of the model. (2) Representative constant material properties are adopted in the present thermoelastic analysis. This simplification is introduced because complete temperature-dependent material data over the entire investigated cryogenic temperature range are not available. The influence of temperature-dependent material properties on the predicted deformation is therefore regarded as a limitation of the present model. (3) The convective heat transfer coefficient of the fluid between the sealing medium and the external air does

not change with temperature. (4) Ignoring the rubbing, friction, and wear heat between the seal ring end faces, the seal end faces are completely fluid-lubricated. (5) The seal end face is in full contact with the lubricating liquid film; there is no gap, and the liquid film and the wall surface are fully heat exchanged. (6) Do not consider the influence of thermal radiation and gravity factors. These assumptions are introduced to simplify the thermoelastic analysis and isolate the first-order effect of thermal and mechanical loads on seal-face deformation. The assumptions of constant material properties and heat-transfer coefficients may affect the predicted temperature distribution and thermoelastic deformation, while neglecting frictional heat, radiation, and gravity may introduce additional uncertainty. Therefore, their influence on the quantitative deformation results is acknowledged as a limitation of the present study. These assumptions are introduced to simplify the thermoelastic analysis and isolate the first-order effect of thermal and mechanical loads on seal-face deformation. Their possible influence on the quantitative deformation results is acknowledged as a limitation of the present study. The model of the rotating ring model of the divided grid after the import is shown in Figure 11c, and the stationary ring model of the divided grid is shown in Figure 11d.

Before the deformation simulation, the grid independence test was first carried out. The test method is that the size of the grid has little effect on the simulation results after applying a small temperature change and a small load. The number of grid elements affects the calculation results of the deformation. Therefore, taking thermal deformation and stress deformation as evaluation indices, the grid independence of different grid densities was studied while taking into account the calculation accuracy and efficiency. The results obtained by using coarse meshes are sensitive to mesh size. When the number of mesh elements increases to 180,000, the changes in these two parameters become very small. This shows that the results are basically grid-independent. Considering the calculation accuracy and computational efficiency, this study finally used a grid with a grid number of about 180,000, and the calculation step size was set to 500.

Due to the existence of the fluid field and the magnetic field on the end face of the dynamic and stationary rings, the liquid-film force and superconducting magnetic force are considered when setting the boundary conditions. The liquid film contacts and bears the dynamic pressure of the liquid film on the end face, so the end face of the rotating ring is set as the fluid–solid coupling surface in the calculation. The loading pressure on the outer and back sides of the rotating ring is the same as that of the sealing medium. According to the test data in Figure 7b, the inner diameter side of the sealing ring is set to standard atmospheric pressure, and the outer pressure is set to 0.95 MPa. The inner side of the rotating ring needs to be fixed to the rotor. In the calculation, the inner side of the motion ring is constrained by the ring and fixed on its back. According to the simulation data in Figure 11a, the magnetic fluid composite force of the rotating ring and the stationary ring section is set to a uniform load of 2.5 kN. According to the experimental data in Figure 7c, the temperature boundary condition applied to the inner side of the rotating ring is $-90\text{ }^{\circ}\text{C}$, and the temperature boundary condition applied to the outer side of the rotating ring is $-160\text{ }^{\circ}\text{C}$. The temperature boundary conditions are applied along the thickness direction of the rotating ring under the temperature gradient of $-160\text{ }^{\circ}\text{C}\sim-90\text{ }^{\circ}\text{C}$. Figure 12 is a schematic diagram showing the applied boundary conditions.

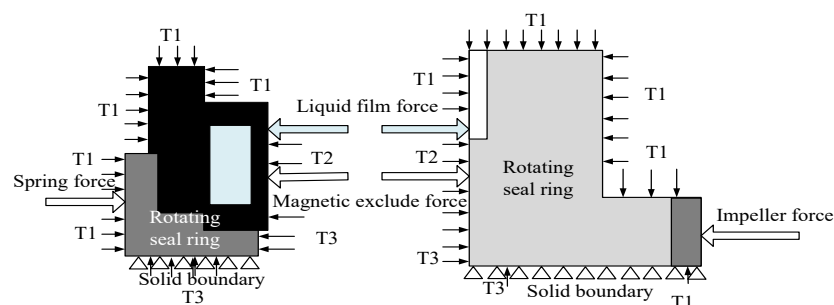


Figure 12. Schematic diagram of applying boundary conditions.

The finite element analysis results of the thermal deformation of the rotating ring and the stationary ring are shown in Figure 13. Figure 13a shows the simulated deformation of the rotating ring. The larger deformation occurs on the outer side, with a maximum value of $8.3 \mu\text{m}$. This maximum deformation is equivalent to the thickness of the liquid film when the mechanical seal is working. Figure 13b shows the simulation results of the stationary ring. It can be seen from the figure that the stationary ring of the turbopump mechanical seal is made of impregnated graphite material. The linear expansion coefficient is $4.8 \times 10^{-6} \text{ K}^{-1}$, which is far less than the thermal expansion coefficient of the metal. The deformation after heating is very small, and the plasticity is not strong. The deformation after stress is also very small. Therefore, the deformation obtained by the stationary ring simulation is very small. The maximum deformation is outer side the stationary ring, and the maximum deformation is $0.8 \mu\text{m}$. The predicted maximum deformation of approximately $8.3 \mu\text{m}$ has not been experimentally verified in the present study and will be further investigated in future work.

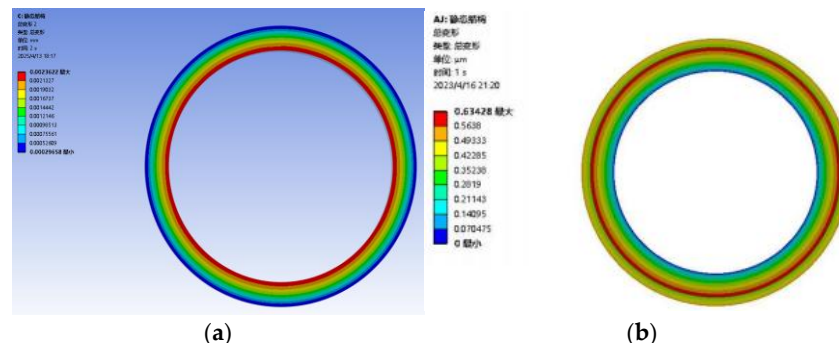


Figure 13. (a) The deformation of the rotating ring; and (b) the deformation of the stationary ring.

5. The Influence of Thermal Deformation of Mechanical Seal on Lubrication Performance

From the previous finite element simulation results, it can be seen that the deformation of the rotating ring is much larger than that of the stationary ring when the mechanical seal is working. The deformation of the rotating ring increases along the direction of the outer diameter, and the maximum deformation can reach $8.3 \mu\text{m}$. The shape of the rotating ring after deformation is that the side produces a large deformation. The maximum deformation is that the outer axial deformation is large under a large temperature difference, and the inner deformation is small, so that the whole seal end face is shown as a conical structure.

After the deformation, the surface of the rotating ring of the mechanical seal and the surface of the stationary ring are no longer parallel. The film thickness formula in Equation (2) for the lubrication performance calculation model of the mechanical seal introduced above needs to be corrected. At this time, the liquid-film thickness of the mechanical seal can be expressed by two quantities, the minimum film thickness $h_{\min}$

and the maximum film thickness $h_{\max}$. According to the deformation characteristics of the rotating ring, $h_{\min}$ is at the innermost end of the mechanical seal, and $h_{\max}$ is at the outermost end of the mechanical seal, as shown in Figure 14. The θ_1 in Figure 14 can be calculated from the deformation of the mechanical seal ring, according to the finite element simulation results.

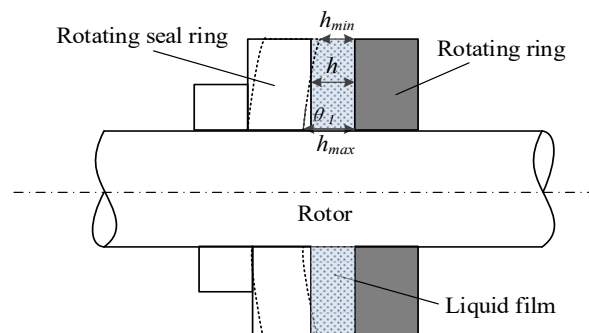


Figure 14. Lubrication film thickness model of mechanical seal considering deformation.

After updating the film–thickness distribution using the calculated seal-face deformation, the Reynolds equation is re-solved based on the corrected film geometry, and the resulting lubrication performance is used for the subsequent analysis. Assuming that the deformation varies linearly along the radial direction, the corrected film thickness h' at any radial position can be expressed by Equation (15). In Equation (15), R is the outer radius of the dynamic sealing ring, and r is the radius at any point.

$$\begin{aligned} h' &= h + (R - r) \sin \theta_1 \\ &= h_0 - \varphi r \cos \theta - \gamma r \sin \theta + h_g + (R - r) \sin \theta_1 \end{aligned} \quad (15)$$

In the calculation and analysis of lubrication performance, the minimum film thickness $h_{\min}$ after deformation is compared with the film thickness h before deformation, and the change in leakage before and after deformation is compared. The calculation results are shown in Figure 15.

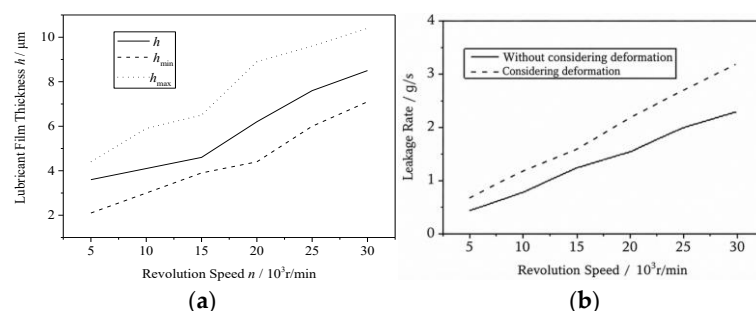


Figure 15. Film thickness and leakage change with rotational speed. (a) Variation in lubricating film thickness with rotational speed; (b) variation in leakage rate with rotational speed.

Figure 15a is the calculation result of film thickness changing with rotational speed. It can be seen that the lubricating film thickness before and after deformation increases with the increase in rotational speed n . When n is the same, h is between $h_{\max}$ and $h_{\min}$. If the influence of deformation is not considered in the calculation, the calculated film thickness is larger than the actual minimum film thickness. Although the reduced film thickness does not lead to direct face contact under the investigated conditions, it may increase the risk of contact and wear, particularly under additional impact loads. Direct contact and wear were not experimentally investigated in the present study. Figure 15b shows the calculation

results of the leakage Q with the change in the rotational speed n . It can be seen that the Q after considering the deformation is larger than that without considering the deformation. The reason is that the gap between the rotating ring and the stationary ring increases after the deformation, so that Q also increases.

As shown in Figure 15a, the calculated minimum film thickness remains positive under the investigated operating conditions. Therefore, direct contact between the rotating and stationary seal faces is not predicted in the present model, and the seal remains within the full-film lubrication assumption. However, the reduction in the minimum film thickness indicates an increased potential risk of local contact, particularly under additional impact or transient loads. Mixed lubrication and direct face contact are not considered in the present model.

The present framework remains sequential rather than fully coupled. The deformation-corrected film geometry is used to re-solve the Reynolds equation; however, the updated hydrodynamic load is not repeatedly transferred back to the structural model for further deformation iterations. In addition, the deformation-induced variation in the magnetic gap is not recalculated in the present model. The seal-face deformation is represented by a linear radial approximation rather than direct nodal mapping of the finite-element surface. These simplifications are regarded as limitations of the present study and will be addressed in future work.

6. Conclusions

- (1) Based on the superconducting magnetic-force and hydrodynamic lubrication framework established in our previous study, the present work further introduces the thermoelastic deformation analysis of the rotating and stationary seal rings. The deformation-induced change in seal-face geometry is incorporated into the lubrication-performance evaluation through the minimum and maximum film thicknesses, $h_{\min}$ and $h_{\max}$, thereby enabling the influence of thermoelastic deformation on film thickness and leakage to be assessed.
- (2) The superconducting magnetic force and liquid-film force at different rotational speeds are calculated and analyzed. Under the fixed gap, the superconducting magnetic force does not change with the change in rotational speed, and the liquid-film force increases with the increase in rotational speed. The superconducting magnetic force and the hydrodynamic liquid-film force under different working clearances are calculated and analyzed. With the increase in working clearance, both the magnetic force and the liquid-film force decrease rapidly, and the variation in both is nonlinear. Under the same clearance, the liquid-film force is larger than the magnetic force, but the magnetic contribution to the total bearing capacity cannot be ignored. The forces at different rotational speeds are further analyzed. Under a fixed clearance, the magnetic force is nearly independent of rotational speed, while the liquid-film force increases with increasing speed.
- (3) The deformation of the mechanical seal under the coupled magnetic force, liquid-film force, and thermal boundary conditions is obtained by finite-element simulation, and its influence on lubrication performance is investigated. The rotating ring deforms much more than the stationary ring; the maximum deformation occurs near the outer diameter and is on the order of the working film thickness, and the deformation increases along the radial direction, leading to a conical end-face shape and loss of face parallelism. Therefore, the film thickness should be described by the minimum and maximum values ($h_{\min}$ and $h_{\max}$). After considering deformation, the minimum film thickness becomes smaller, and the leakage becomes larger; neglecting deformation would overestimate $h_{\min}$ and underestimate leakage, and the reduction in minimum

film thickness may increase the potential risk of contact and wear, although direct wear was not investigated in the present study.

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