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Retraction

Retracted: Friction Performance Analysis of Reactor Coolant Pump Shaft Seal Based on Sensor and Computer Simulation

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This article has been retracted by Hindawi following an investigation undertaken by the publisher [1]. This investigation has uncovered evidence of one or more of the following indicators of systematic manipulation of the publication process:

(1) Discrepancies in scope
(2) Discrepancies in the description of the research reported
(3) Discrepancies between the availability of data and the research described
(4) Inappropriate citations
(5) Incoherent, meaningless and/or irrelevant content included in the article
(6) Manipulated or compromised peer review

The presence of these indicators undermines our confidence in the integrity of the article's content and we cannot, therefore, vouch for its reliability. Please note that this notice is intended solely to alert readers that the content of this article is unreliable. We have not investigated whether authors were aware of or involved in the systematic manipulation of the publication process.

Wiley and Hindawi regrets that the usual quality checks did not identify these issues before publication and have since put additional measures in place to safeguard research integrity.

We wish to credit our own Research Integrity and Research Publishing teams and anonymous and named external researchers and research integrity experts for contributing to this investigation.

The corresponding author, as the representative of all authors, has been given the opportunity to register their agreement or disagreement to this retraction. We have kept a record of any response received.

References

[1] Q. Gou, S. Xiao, and X. Chen, "Friction Performance Analysis of Reactor Coolant Pump Shaft Seal Based on Sensor and Computer Simulation," *Journal of Sensors*, vol. 2022, Article ID 7838491, 11 pages, 2022.

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Research Article

Friction Performance Analysis of Reactor Coolant Pump Shaft Seal Based on Sensor and Computer Simulation

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In recent years, the state has put forward a grand plan to develop nuclear power in order to control environmental pollution and carbon emissions. In nuclear power plants, mechanical seals for reactor coolant pump play an important role in nuclear power safety production. However, China cannot independently produce such mechanical seals, and they are blocked by foreign related technologies, which has seriously affected China's nuclear power development plan and the overall safety of China's nuclear power operation. Under this background, this paper studies the friction performance of the shaft seal of the reactor coolant pump, uses the sensor to obtain the data of friction factors such as low-pressure leakage, effectively monitors the low-pressure leakage through calculation and simulation modeling, truly reflects the operating state of the reactor coolant pump, and provides a new research direction and experimental basis for further analysis of the friction performance of the shaft seal of the reactor coolant pump. The results show that under the joint action of the pressure difference and the force generated by the deformation of the moving ring plate, the cone angle formed on the seal end face is 1300.9 μ convergence gap of rad. When the inlet water temperature is 65°C, the leakage rate is 1867.8 L/h. The deformation of the moving ring deformation ring plate can hinder the increase of the deformation cone angle of the moving ring. The greater the thickness of the moving ring deformation ring plate, the greater the seal leakage rate. The inlet temperature of the sealing medium and the friction performance of the material also have an impact on the sealing performance.

1. Introduction

Energy is the material basis of economic and social development. With the growth of population and economy and the improvement of electrical automation, the demand for energy will continue to grow. Taking fossil energy as the main energy has caused the deterioration of the global ecological environment, so it is urgent to promote the transformation of energy to clean and renewable direction, and the growth of nuclear power is a key component of degree to encourage the transformation of power structure in the coming days [1, 2]. Compared with fossil energy, nuclear energy is a clean, environmental friendly, low-carbon energy, and will not face the risk of resource depletion for the time being. Therefore, the development of nuclear power should meet the demand for clean energy. Nuclear power plays a key component of part in neat energy. The potential safety endanger of utilizing nuclear power lies in the leakage of nuclear substances. At present, most pressurized water reactor nuclear power plants in China use shaft seal coolant pumps. Shaft seal is an important component of the coolant pump, which ensures that the radioactive reactor coolant will not leak out through the coolant pump under normal working conditions. The shaft seal design structure is complex, and the processing is precise. At present, only a few coolant pump suppliers in the world have mastered this technology. The shaft seal of the reactor coolant pump prevents radioactive cooling water from entering the air [3]. The seal of the reactor coolant pump is easy to wear due to its extreme working conditions. The shaft seal technology of the reactor coolant pump is closely narrated to the safety and dependable manner of the reactor coolant pump. However, the average localization rate of nuclear power equipment in China is 45%, and some key parts are almost all imported. The localization of nuclear power equipment is imperative.

Reactor coolant pump (RCP) is the “heart” of nuclear power plant, and its main function is to pump the coolant required by the reactor into the reactor system, so that the coolant can be recycled in the reactor system [4, 5]. According to the different sealing forms of the RCP, the RCP is divided into shaft seal pump and shielded pump. Shield pump is designed to integrate the sealing device and pump as a whole, which can ensure that the coolant in the pump will not leak, but its efficiency is relatively low, and it cannot continue to be used after the power failure of the nuclear power plant. Therefore, it is gradually replaced by the shaft seal pump in the development process of the sealing technology of the RCP, but some types of shield pumps are still used in nuclear power plants. Just now, shaft seal pumps are mostly utilized during the second and third generation nuclear power plants. Shaft seal is an intention to temperance the leakage of shaft seal pump, and its performance is closely narrated to the approach safety, patch up, and maintenance cycle of nuclear power plant [6, 7]. The shaft seal in the shaft seal pump belongs to mechanical seal according to the classification of seal, which is the key auxiliary component in the RCP. Once the nuclear power plant leaks, it will cause huge losses and even disasters. In 1979, the reactor of the Three Mile Island nuclear power plant in the United States leaked, causing huge economic losses. The Chernobyl accident in Ukraine in 1986 is known as the worst nuclear power disaster in history. In 2011, the power failure of Fukushima nuclear power plant in Japan caused the RCP to stop working due to the earthquake, resulting in the leakage of radioactive substances, causing a global crisis. The mechanical seal at the shaft end of the RCP is used to limit the leakage of the nuclear reactor coolant along the pump shaft and force it to flow to the chemical volume control system or the nuclear reactor coolant drain tank as much as possible. The failure of the seal of the RCP will not only bring huge economic losses but also pose a potential threat to the safety and health of the public. In recent years, with the development of nuclear power in the world, mainly the speedy growth of nuclear power in China, the seal of RCP has got grand attention. Many scholars have conducted more in-depth research on the seal of RCP, and a large number of relevant documents have been published. With the development of rubber and various auxiliary materials, shaft sealing materials are constantly improved to meet the harsh sealing conditions. Its friction and wear performance directly determines the serviceability and service life of the seal ring. In order to ensure the safety and reliability of the shaft seal under the actual working conditions, the experimental research on the friction and wear characteristics of the shaft seal material is of great significance.

In the RCP, the mechanical seal is a vulnerable equipment, which is also prone to failure. The failure caused by the mechanical seal accounts for more than half of the coolant pump accidents, so the mechanical seal, as the key equipment of the nuclear power plant, plays a key part in the trustworthy and dependable practice of the nuclear power plant. Therefore, we should master the key technology of the design of the mechanical seal for the RCP as soon as possible and get rid of the bondage of foreign mechanical seals on the development of nuclear power in China; it is an important point and a need to swiftly encourage the advancement of national nuclear power industry. Based on the above situation, these notes present a survey on the friction performance of RCP shaft seal based on sensors and computer simulation. Taking the new streamline groove mechanical seal for RCP as the research object, the lubrication model of streamline groove mechanical seal is established, the influence law of end face parameters on sealing performance is analyzed, and the end face parameters are optimized by single variable method and orthogonal experiment method, which provides effective support for improving the basic theory of RCP sealing technology. Experiments show that the seal face produces circumferential waviness and radial taper. With the increase of seal pressure, the wavy deformation and taper deformation of seal face increase. The thickness of lubricant film between seal faces increases with the increase of seal pressure, and the friction coefficient decreases with the increase of seal pressure.

In this study, sensor and computer simulation technology are used to study the friction performance of RCP shaft seal. This study will elaborate the research process of shaft seal friction of RCP from five aspects. The first part introduces the background and significance of the research on the sealing of RCPs at home and abroad. The second part summarizes the shaft seal technology and research status of RCP. The third part is based on the sensor and computer simulation technology to study the friction performance of the RCP shaft seal. Section 4 is the experimental analysis and result comparison. Section 5 analyzes and summarizes the full text and prospects the future research direction.

2. Related Work

Mechanical seal is an axial rotating sealing device that relies on the joint action of elastic elements and sealing medium pressure to maintain the fit of two sealing faces, so as to realize sealing. Mechanical seals can be divided into contact type and noncontact type according to whether the end faces of the two seal rings are in contact. Contact mechanical seal has the benefits of easy structure and low leakage rate, but because of its large end wear and calorific value, it is not suitable for the application of high temperature, high pressure, and other occasions. Noncontact sealing refers to the phenomenon that the noncontact sealing device does not contact with the shaft, does not produce friction, and has oil grooves, oil slingers, labyrinth seals, etc. The existing noncontact sealing technologies include gas film seal and liquid film seal. The medium delivered by the RCP is the coolant with high temperature, high pressure, and high radiation. The ordinary contact mechanical seal cannot meet the requirements of its service conditions, so the mechanical seal for the main pump often adopts the noncontact mechanical seal with deep groove on the sealing end confront or taper on the end confront [8, 9]. In normal operation, because the two seal faces are not in contact, the wear of the seal face is small, and it can operate stably in high-pressure occasions for a long time.

Because of the speedy advancement of science and tech and the high parameter requirements of sealing conditions, the design of mechanical seals presents a more severe challenge. At present, mechanical seals are developing rapidly in the direction of high parameters such as high pressure and high temperature. Using sensors to obtain shaft seal friction data in real time and then using computer simulation technology to simulate and analyze the data is the current hot research direction of this research, which is more conducive to real-time, accurate, and specific analysis. Generally, the deformation of the seal end confront and the thickness of the lubricating liquid movie are overall in similar bid of importance [10]. Under high parameters, the deformation of the mechanical seal end confront has a great impact on the thickness of the lubricating liquid film, so it will also affect the sealing performance. Mechanical seal face deformation is mainly caused by force deformation (mechanical deformation) and thermal deformation (temperature deformation). The mechanical deformation of the seal ring is mainly caused by the joint action of medium pressure, liquid film pressure, spring force, contact load, etc. When the mechanical seal works normally, the temperature distribution of the seal ring will change due to the effect of factors such as the temperature of the seal medium and the friction heat caused by the relative movement of the two seal faces, and the seal ring will be deformed to varying degrees due to uneven heating. In particular, the end face will produce taper and waviness due to deformation.

As for the study of mechanical deformation of seal ring, Gemma ring theory, pitzino ring theory, gram body torque theory, finite element method, finite difference method, and finite volume method are used as usual. Zhang et al. [11] simplified the seal ring into a cylinder according to Gemma ring theory and obtained the deformation and rotation angle of the seal ring. In addition, the author used the finite element method to calculate the thermal deformation of the mechanical seal and found that the temperature difference of the seal face will deform the flat seal face. Although the finite element software has powerful functions, it also has certain limitations. For example, for mechanical seals with shallow grooves on the end face, the groove depth is micron level and the groove width is millimeter level, which brings some difficulties to modeling and meshing. Literature [12] studied the influence of force deformation and thermal deformation on the performance of hydrostatic mechanical seals by establishing a two-dimensional thermoelastic hydrodynamic lubrication model (TEHD). Literature [13] simplifies the seal ring into an axisymmetric model and analyzes the force deformation of the seal ring by using the ring theory and two-dimensional finite element model, respectively. The results show that this method has a good effect on calculating the deformation of complex seal rings. This method simplifies the results calculated from the modeling analysis and simplifies the complex engineering problems that feel there is no place to start. But the accuracy fluctuates greatly. Based on the level and boundary conditions of modeling, whether the simulation of load cases is true, etc. Gong et al. [14] used the general finite element method to discuss the influence of seal interface wear on tropical migration. The wear simulation program is used to calculate the wear of the seal ring, and the thermoelastic effect is also included in the simulation model. The results show how wear affects the migration of hot rolled strip, and the wear of sealing surface leads to more complex tropical migration state, including the division and combination of tropics. The proposed simulation method and program are suitable for trend analysis of seal design. Wittmaack et al. [15] used the finite element method to study the mechanical deformation of the mechanical seal ring of the high parameter compressor. In this method, the continuous solution domain is discretized into a combination of a group of elements, and the approximate function assumed in each element is used to represent the unknown field function to be solved in the solution domain. The approximate function is usually represented by the numerical interpolation function of the unknown field function and its derivatives at each node of the element. The results show that the working pressure difference causes the force deformation of the seal ring. Under high pressure, the force deformation of the seal ring has a great impact on the sealing performance, which cannot be ignored. Literature [16] uses the boundary element method to calculate the deformation of the mechanical seal. The deformation of the seal ring is numerically calculated and compared with the experimental value. It is found that the boundary element method can better solve the end face deformation. Literature [17] put forward a TEHD numerical analysis model of mechanical seal considering the influence factors such as heat, inertia effect, and turbulence and analyzed the thermal deformation of the seal ring by calculating the heat conduction between the fluid film and the seal end face. The results show that the deformation of the end face has an important influence on the thickness of the lubricant film on the seal end face. TEHD method is to consider not only the comprehensive influence of temperature change on fluid pressure generation and viscosity change but also the influence of pressure and temperature gradient in bearing bush on thrust bearing performance in numerical analysis block deformation, so the simulation results obtained by TEHD method are more accurate. Ding et al. [18] established a fluid solid coupling theoretical analysis model of deep groove end face density and obtained the pressure distribution, liquid film thickness, and leakage rate of the liquid film by using the numerical calculation method. It was found that the wavy deformation of the seal end face would occur due to the effect of the deep groove, resulting in the hydrodynamic pressure effect due to the extrusion of the liquid film at the convergence gap in the direction of speed rotation.

At present, domestic and foreign scholars' research on the sealing ring mainly focuses on the design and optimization of the sealing structure, as well as the unilateral performance of the sealing ring material, such as the aging performance of composite materials. Most of them mainly focus on the aging mechanism of materials, mechanical properties of materials, etc., but there is less research on the change law of friction and wear of composite materials after aging. Through the analysis of the above research status, it can be found that at present, China's research on mechanical seals for RCP mostly stays in theoretical research, and there is little research on experimental analysis and application introduction, which indirectly explains the development stage of RCP in China, but it is through these stages that China can consolidate the theoretical foundation and further promote the development of RCP sealing technology and pave the way for the complete autonomy of the RCP seal in China. This paper presents an analysis method of shaft seal performance based on sensor and computer simulation technology. Over theoretical calculation, it is found that its performance parameters are improved compared with the existing mechanical seal with groove on the end face, mainly in terms of opening force, stiffness, and bearing capacity. At the same time, its surface hydrodynamic effect is also better, so it is applied to the RCP seal.

3. Simulation Analysis of Shaft Seal Friction Using Sensor Data

3.1. Sealing Performance Parameters. When the streamline groove mechanical seal works, due to the setting of the groove on the sealing end face, there is a convergence gap, and the relative rotation of the sealing end face forms a fluid film with a certain stiffness, so as to achieve the sealing effect [19]. To study the sealing performance of mechanical seals is essentially to study the application of hydrodynamic lubrication theory in mechanical seals. Hydrodynamic lubrication refers to the lubrication state in which the fluid between the two end faces of the friction pair depends on the uneven shape of the end faces of the friction pair, and there is a convergence gap. When the two end faces move with each other, the fluid between the end faces forms a fluid film due to the hydrodynamic effect, which separates the moving end faces from each other. In 1883, tower tested the bearings of train axles and observed the phenomenon of hydrodynamic pressure for the first time. In 1886, Reynolds put forward the basic equation of lubrication theory based on the knowledge of fluid mechanics, successfully revealed the mechanism of fluid film, and laid the foundation for modern fluid lubrication theory. From the perspective of mathematics, the main content of hydrodynamic lubrication theory is the solution of Reynolds equation.

3.2. Continuity Equation. In fluid mechanics, the continuity equation is the specific expression of the law of conservation of mass [20]. The premise of its application is that the velocity and density of fluid medium are continuous. If the flow between seal faces is stable, the mass of fluid flowing out of a certain area must be equal to the mass of fluid flowing in. If the flow between the seal faces is in an unstable state, the difference between the mass of the fluid flowing into a certain area and the mass of the fluid flowing out must be the value that causes the mass change in that area. In mathematical form, it is the continuity equation. As shown in Figure 1, assuming that the area is a cube, the continuity equation is solved according to the average flow model.

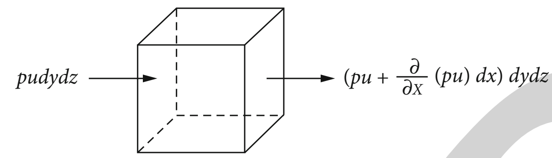

Figure 1: Micro flow diagram.

Taking the x direction as an example, the mass flowing in from the left side per unit time is *pudydz*, and the mass flowing out from the right side per unit time is as shown in the following formula:

$$\left[pu + \frac{\partial}{\partial x}(pu)dx\right]dydz. \tag{1}$$

The mass difference of flow in and out in X direction in unit time is

$$\left[pu + \frac{\partial}{\partial x}(pu)dx\right]dydz - pudydz = \frac{\partial}{\partial x}(pu)dxdydz. \tag{2}$$

Similarly, the quality difference of outflow and inflow in Y direction and Z direction is

$$\left[pv + \frac{\partial}{\partial y}(pv)dy\right]dxdz - pvdxdz = \frac{\partial}{\partial y}(pv)dxdydz, \tag{3}$$

$$\left[pw + \frac{\partial}{\partial z}(pw)dz\right]dxdy - pwdxdy = \frac{\partial}{\partial z}(pw)dxdydz. \tag{4}$$

The mass increased by density per unit time is $(\partial p/\partial t)$ $dxdydz$. That is

$$\frac{\partial p}{\partial t}dxdydz + \left[\frac{\partial(pu)}{\partial x} + \frac{\partial(pv)}{\partial y} + \frac{\partial(pw)}{\partial z}\right]dxdydz = 0. \tag{5}$$

The general form is

$$\frac{\partial p}{\partial t} + \frac{\partial(pu)}{\partial x} + \frac{\partial(pv)}{\partial y} + \frac{\partial(pw)}{\partial z} = 0, \tag{6}$$

where pu, pv, and pw refer to the flow per unit length in the x, y, and z directions, respectively; $\partial/\partial x(pu)dx$, $\partial/\partial y(pv)dy$, and $\partial/\partial z(pw)dz$ refer to the flow change rate in the x, y, and z directions, respectively.

3.3. Expression of Mechanical Seal Performance Parameters

3.3.1. Opening Force. The opening force refers to the bearing capacity of the fluid film [21]. After the pressure distribution of the sealing face is obtained, the opening force is obtained by integrating it on the whole sealing face. The opening force is used to describe the force that keeps the seal face open. When the opening force is balanced with the closing force, the seal operates stably. The calculation formula of opening force is

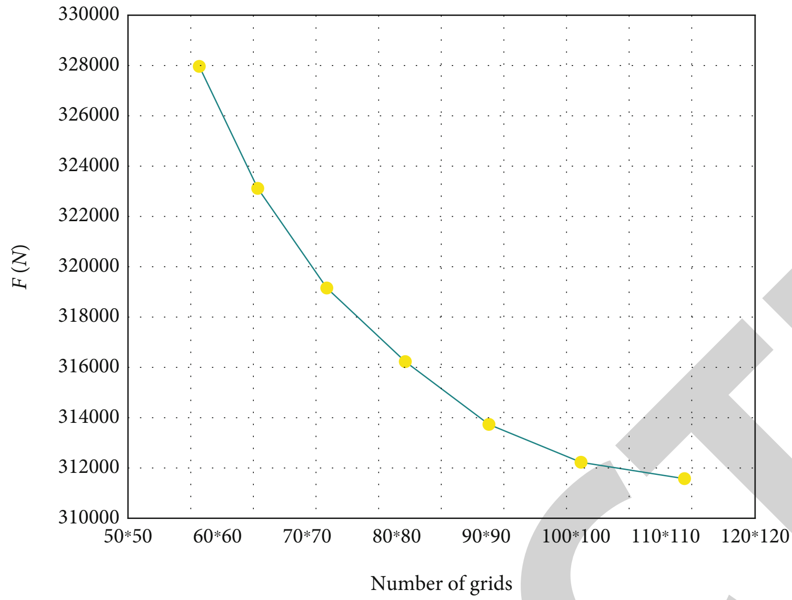

FIGURE 2: The grid independence test and verify.

$$F0 = \int_0^{2\pi} \int_{r0}^{ri} prdrd\theta, \quad (7)$$

where F_0 is the opening force; θ is the circumferential boundary angle; r_0 and r_i are the radial boundary.

3.3.2. Leakage Volume. Leakage is the most important performance parameter to evaluate the mechanical performance [22, 23]. It refers to the amount of sealing medium leaking between sealing ends in a unit time. At present, the leakage formula under mixed lubrication provided by Mayer in Germany is generally used in the study of leakage:

$$Q = -\int_0^{2\pi} \left(\frac{h^3}{12\mu} \frac{\partial p}{\partial r} \right) rid\theta, \quad (8)$$

where r_i is the dimensionless inner diameter; p is the dimensionless internal pressure.

The opening leakage ratio refers to the ratio of opening force and leakage rate, and its calculation formula is

$$Ef = \frac{F0}{Q}. \quad (9)$$

3.4. Numerical Solution of Reynolds Equation

3.4.1. Finite Difference Method (FDM). Reynolds equation contains partial differential items, so it is very troublesome to find its analytical solution. Although many experts and scholars have made in-depth research on it, the effect is very little. At present, the treatment method of Reynolds equation is mainly numerical calculation method. FDM, FEM (finite element method), FVM (finite volume method), and emerging geometric analysis methods are mostly used. Each method has its applicable occasions. Among them, FDM is the earliest kind of method. Its basic idea is to draw a differential grid in the solution area. The variables and their differential terms in the equation are expressed by nodes or difference quotients of nodes and are simplified into differential algebraic equations through transformation. FEM is used to deal with more complex solving regions, and FVM is suitable for solving complex regions and solving regions with flexible meshing. Surface modeling technology is developing rapidly, and the limitations of the above methods hinder its development and application. The emergence of geometric analysis overcomes the limitations of traditional finite element analysis in mesh generation and can not only improve the numerical accuracy of the analysis model but also greatly improve the analysis efficiency. FDM is selected as the regional comparison rule studied in this paper. Although other methods will improve the accuracy of numerical calculation, it will increase a lot of calculation. Therefore, FDM is used for comprehensive consideration.

3.4.2. Grid Independence Verification. The number of meshes will affect the accuracy of numerical analysis, so it is necessary to select the appropriate number of meshes to discretize the solution area. Too many grids will lead to too long calculation time, and the fluctuation range of calculation results is not large, but if the number is too small, it will seriously affect the accuracy of numerical simulation. Therefore, it is necessary to select an appropriate number of grids to reduce the deviation of calculation results caused by the number of grids.

As shown in Figure 2, the opening force first decreases rapidly, then decreases slowly, and finally tends to be stable with the increase of the number of grids. When the number of grids is greater than 100×100, the opening force changes little, and the error is less than 1%, so the selected number is 100×100, but in order to distinguish the difference between

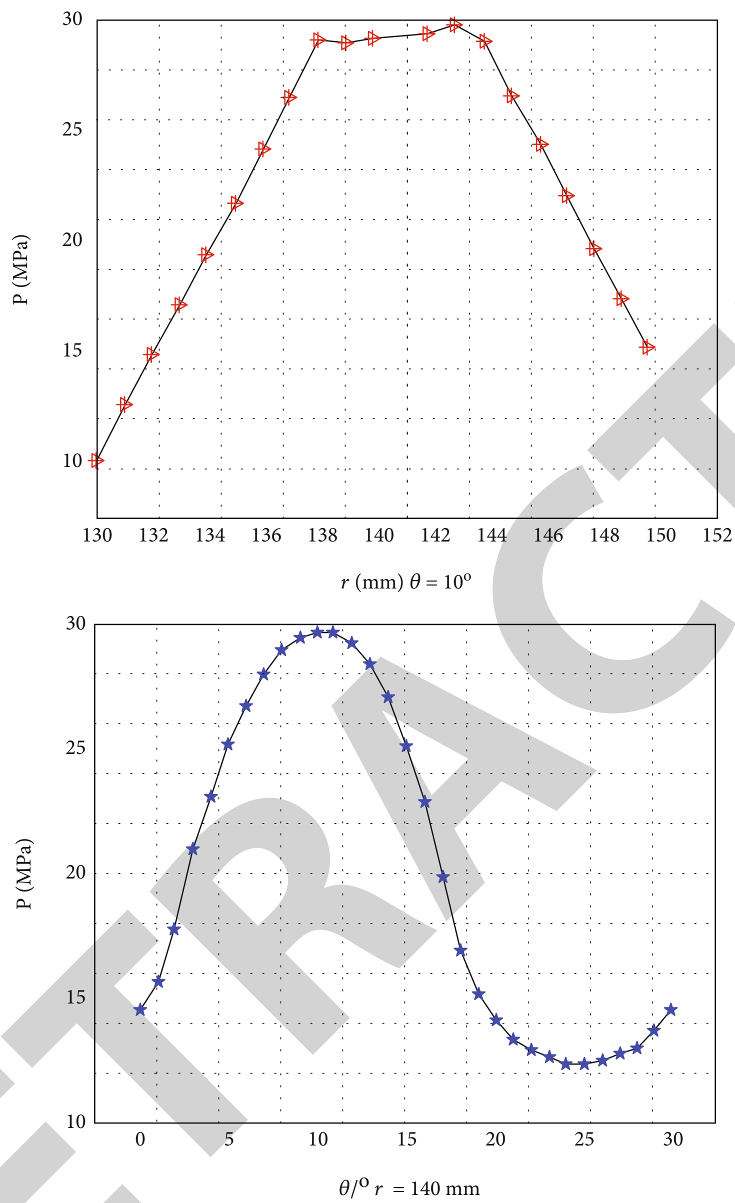

FIGURE 3: Radial and circumferential pressure distribution of fluid film.

the inner diameter direction and the circumferential direction of the calculation area, the number of grids in the circumferential direction is 120, and the final number of grids is 100×120.

3.4.3. Fluid Film Pressure Distribution. Figure 3 shows the radial and circumferential distribution curves of pressure. When studying the radial distribution law, take the circumferential direction $\theta = 10^\circ$, it has the ability to be seen from the figure that along the direction of radius reduction, the pressure first increases, then passes through a stable stage, and finally gradually decreases. The section with radius $r = 130 \sim 138$ mm is a sealed dam, the section with radius $r = 138 \sim 144$ mm is a groove area, and the section with radius $r = 144 \sim 150$ mm is a sealed weir. Due to the dynamic pressure effect, the pressure value in the groove area is larger than that in the dam area and weir area. Therefore, taking the groove area as the center, the pressure value towards the dam area and weir area gradually decreases, and finally, the pressure value reaches the boundary value of the seal ring pressure. When studying the circumferential distribution law, take $r = 140$ mm. It has the ability to be seen from the figure that with the increase of the circumferential degree, the pressure value first increases and then decreases and finally gradually increases to and $\theta =$ Value at 0°. When $r = 140$ mm, $\theta =$ the range of $0 \sim 15^\circ$ is the groove area, and the direction of angle increase is the direction of fluid inflow, so the pressure value gradually increases, $\theta =$ the section at $15 \sim 25^\circ$ is weir area, and the pressure value gradually decreases, $\theta =$ the interval between 25° and 30° is the groove area of another calculation area, so the pressure value gradually increases. At the same time, due to the boundary conditions set in the circumferential direction, the pressure value will reach $\theta =$ value at 0°.

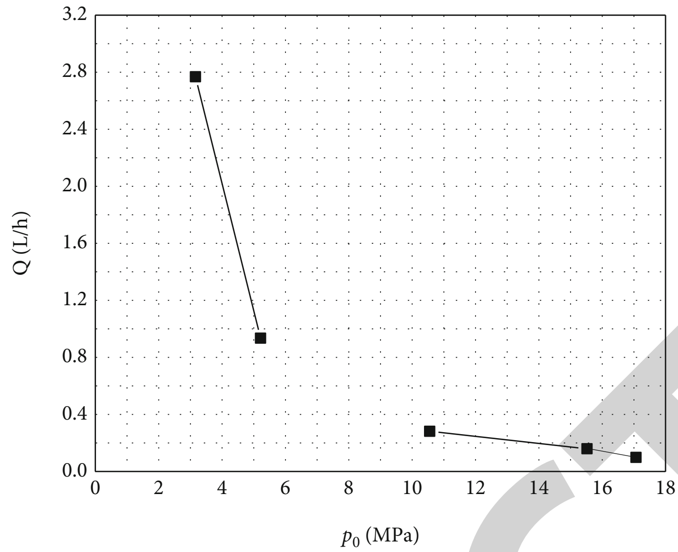

Figure 4: Variation law of leakage with force.

4. Simulation Experiment Analysis of Shaft Seal Friction of RCP Based on Sensor

At present, the theoretical and practical research on hydrostatic mechanical seals for RCP has become mature, and the theoretical research and related experimental research on shaft seal friction based on sensors and computer simulation need to be further improved. Using sensor and computer simulation technology, the reusability and scalability of the simulation model are mainly considered. Problems in the field of industrial applications are generally complex, and many types of decision variables coexist, such as continuous and discrete decision variables. There are quantitative and qualitative problems in discrete decision variables, and the size of different types of problems in the same field is also different. Therefore, we should pay attention to the research on the ability of simulation optimization algorithm to solve the problem range. Especially in China, the relevant research on the RCP seal for sensors is in its infancy, and the relevant experimental research is less [24, 25]. There is a great difference between the theoretical research and experimental research of mechanical seals, because the theoretical analysis is often obtained through a large number of simplifications, for example, ignoring the possible errors in the actual installation, the vibration of mechanical devices, the inertia effect, and other factors. Therefore, experimental research is a research method that can better reflect the real situation of mechanical seal operation.

This chapter uses sensor data and computer simulation data to conduct experimental research on the friction performance of the shaft seal of the RCP, analyzes the change law of the sealing performance with the operating parameters, and analyzes the change of the seal face morphology before and after operation, so as to lay a test foundation for optimizing the mechanical seal of this structure in the future.

4.1. Experimental Steps

Step 1. Install the mechanical seal correctly, check the installed equipment accordingly, and check the static pressure of the seal cavity.

Step 2. Start the motor, adjust the speed, and observe whether the motor operates normally. Then, it is necessary to record whether the operation is normal under no-load state.

Step 3. Conduct static pressure test, start the three-stage plunger pump, and add water to the sealing cavity. When the cavity is filled with water, the pressure of the liquid in the sealing cavity is changed by adjusting the bypass valve. Under different pressures, the pressure sensor records the leakage rate of the end face. After changing the pressure every time, wait about 5 minutes before recording the data. Use a measuring cylinder to measure the leakage rate, each time for 6 minutes, a total of 3 times, and finally take the average value. After the static pressure test, restore the pressure to the initial value and prepare for the operation test.

Step 4. Start the motor, slowly increase the speed to 1485 rpm after running in at low speed, and then operate the mechanical seal for a period of time. After the mechanical seal operates stably, carry out the next test. Adjust the pressure passing through the seal chamber and record the friction torque and leakage rate of the mechanical seal. After changing the pressure every time, wait about 0.5 hours before recording the data. It takes a long time for the mechanical seal to operate stably due to dynamic pressure. Use a measuring cylinder for measurement, each measurement time is 6 minutes, a total of 3 measurements, and finally take the average value.

Step 5. Conduct the life test, stabilize the pressure at 15.5 MPa, rotate at 1485 rpm, and conduct the 100 h long-

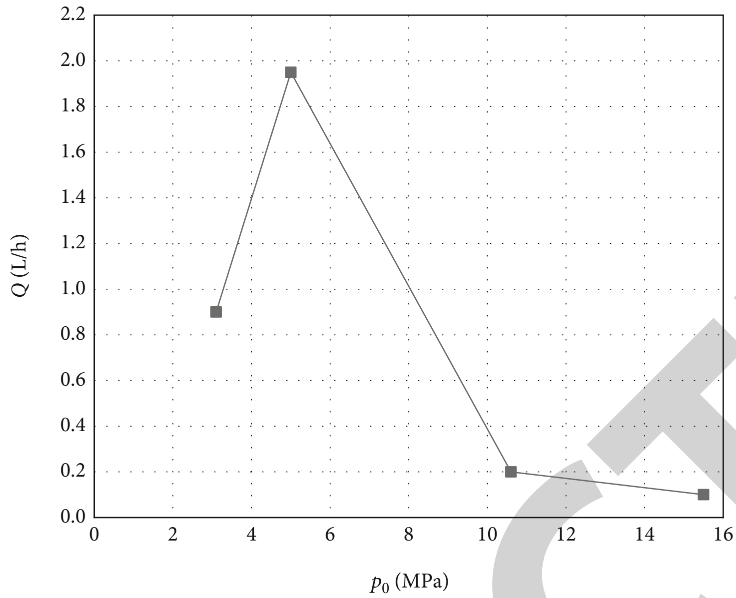

Figure 5: Variation law of seal leakage with pressure.

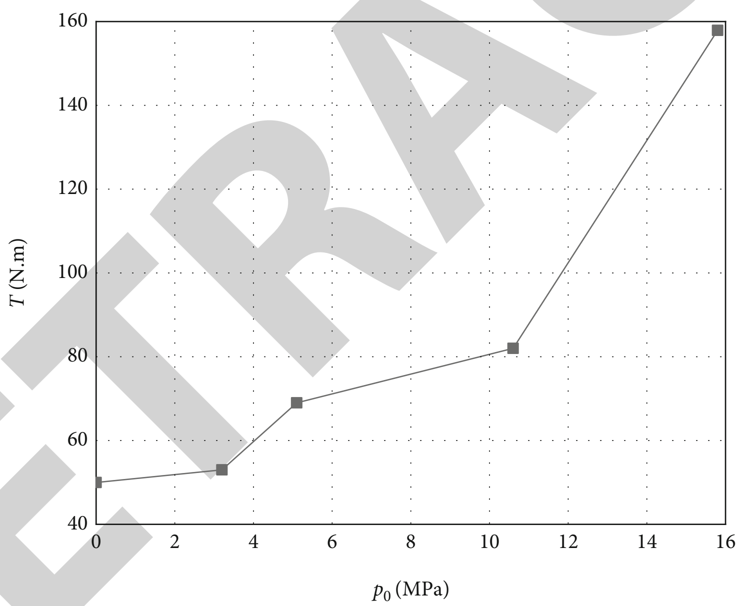

Figure 6: Variation law of friction torque with pressure.

term operation test. During the test, the data shall be recorded every 2 h.

Step 6. After the 100 h operation test, study the influence of the change of operating parameters on the sealing performance by adjusting the pressure and speed, and record the relevant test data. The measurement method is the same as that in Step 4. In addition, it is necessary to compare and analyze the recorded data with the data before the operation test.

Step 7. Disassemble the sealing device, check the friction and wear condition of the seal ring end face, make corresponding records, and analyze and explain the phenomena in the previous test through the changes of the seal ring end face morphology.

4.2. Test Results and Discussion

4.2.1. Static Pressure Test. After the assembly of the test device, the static pressure test shall be carried out on the mechanical seal. Figure 4 shows the change curve of the leakage rate of the mechanical seal with the increase of the sealing pressure under the static state. It can be seen that the leakage rate decreases with the increase of pressure. The reason is that with the increase of the pressure of the sealing medium, the sealing face fits tighter, so the sealing leakage rate continues to decrease.

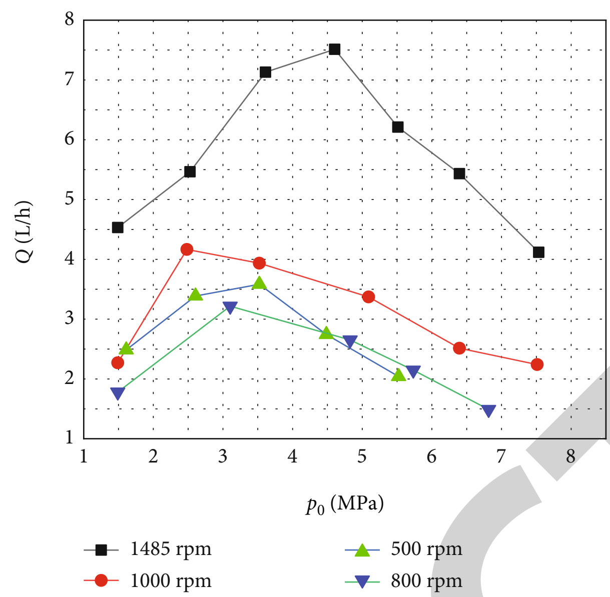

FIGURE 7: The variation of seal leakage rate with pressure at different rotating speeds.

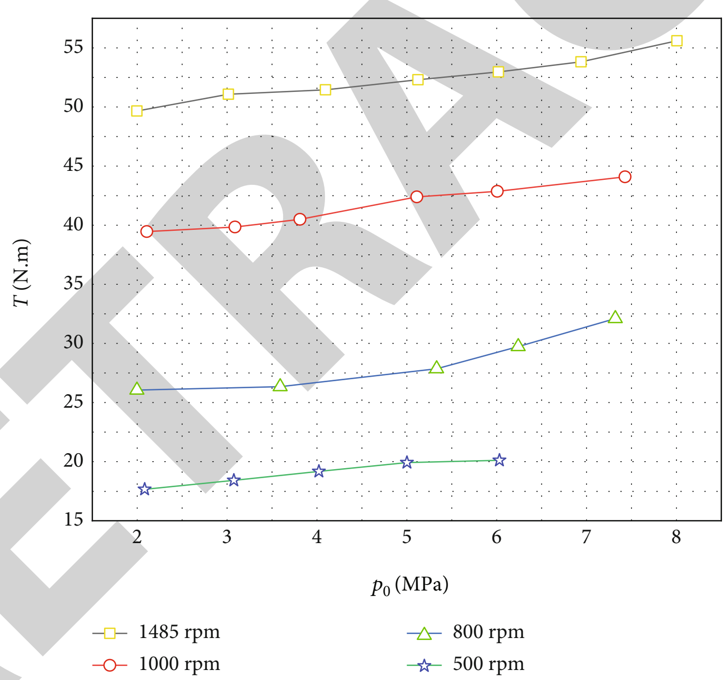

FIGURE 8: Variation of torque with pressure at different speeds.

4.2.2. Operation Test before Long-Term Running In. Figures 5 and 6 show the leakage rate and friction torque curves of mechanical seals under different sealing pressures. It can be seen from Figure 5 that the leakage rate between seal faces first increases and then decreases with the increase of sealing pressure. Under high pressure, the leakage rate between seal faces is almost zero. This may be because with the increase of pressure, the seal face deforms and has a weak hydrodynamic effect, which increases the leakage; when the pressure exceeds a certain value, the negative taper deformation of the end face may be large, resulting in the contact between the two end faces, thus reducing the leakage rate. The results shown in Figure 6 show that the friction torque of the seal face increases with the increase of pressure. This may also be caused by the tighter contact between the two seal faces when the pressure rises.

4.2.3. Operation Test after Long-Term Running In. Figure 7 shows the change of leakage rate with pressure at different speeds after 100 hours of end face running in. It can be seen that the leakage rate first increases and then decreases with the increase of pressure at different speeds. It can be seen from Figures 5 and 7 that the leakage rate between seal faces is much higher after the mechanical seal has been operated for 100 h than that before operation, and the change law of the leakage rate with the increase of pressure has also changed. This may be because the outer edges of the two seal faces are seriously worn after 100 h operation of the mechanical seal under high pressure, resulting in the formation of a taper with hydrostatic effect at the outer diameter of the two seal faces. Therefore, when the pressure is small, the deformation of the end faces is small, and the liquid film between the seal faces is convergent, so the leakage rate increases; with the further increase of pressure, the deformation of the end face increases, and the liquid film on the sealing end face changes from convergence to divergence. At this time, increase the sealing pressure, and the leakage rate decreases.

When the pressure is fixed, Figure 7 also shows the change law of leakage rate with speed. Thus, when the pressure is less than 5 MPa, the leakage rate of the end surface decreases first and then increases significantly with the increase of speed. This is mainly because when the speed and pressure are low, there is less deformation between seal faces, and the speed has less influence on the liquid film on the end. When the rotating speed rises to 1000 rpm, the rotating speed has a great impact on the liquid film, so the leakage rate increases significantly. When the pressure and speed are large, the leakage rate between the end faces increases with the increase of speed. This is mainly because when the pressure increases, the hydrostatic pressure effect of the liquid between the seal faces increases, and with the increase of speed, it is conducive to improve the hydrostatic pressure and dynamic pressure effect of the lubricating liquid film, so the leakage rate increases with the increase of speed.

Figure 8 shows the variation curve of torque with pressure at different speeds. It can be seen that at different speeds, with the increase of pressure, the friction torque between the end faces gradually increases, which is not obvious. However, under the fixed seal pressure, the friction torque increases significantly with the increase of rotating speed.

5. Conclusions

In this paper, the friction of shaft seal of RCP is studied. The computer simulation technology is used to simulate and analyze the relevant data of friction obtained by the sensor. Considering the inlaid structure of the seal ring, the fluid opaque coupling version of the mechanical seal is established, the solution blueprint of the mathematical version is given, the influence law of the functioning parameters on the performance of the mechanical seal is studied, the influence of the geometric parameters of the seal ring and the structural parameters of the linear groove on the sealing performance are analyzed, and the optimized seal ring structure is obtained. The existing sealing products are tested and studied, the influence of operating parameters on sealing performance is studied, the causes of end face wear are analyzed, and the corresponding improvement measures are put forward. Through the above research, the essential conclusions are as follows.

Below the influence of the straight-line deep groove, the seal face produces circumferential waviness and radial taper. With the enhancement of seal pressure, the wavy deformation and taper deformation of seal face increase. The thickness of lubricant film between seal faces increases with the enhanced seal pressure, and the friction coefficient decreases with the increase of seal pressure. Because the influence of heat is not considered in the numerical calculation model, with the increase of rotating speed, the amplitude deformation between seal faces decreases, but the minimum film thickness increases, and the leakage rate does not alter remarkably with the increase of rotating speed.

In this paper, the research on the sealing face is based on the fact that the sealing rings are parallel to each other, but the actual work will inevitably produce heat, which will cause the deformation of the sealing ring and lead to the nonparallel sealing gap. At the same time, this paper does not consider the influence of temperature and surface roughness of seal ring on the performance parameters of mechanical seal, which should be improved in the future research.

Data Availability

The labeled data set used to support the findings of this study is available from the corresponding author upon request.

Conflicts of Interest

The authors declare that there are no conflicts of interest.

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