Experimental Study on the Performance of the Gas Centrifugal Seal with Continuous Injection and Discharge of Sealing Fluid

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Abstract. The centrifugal seal has the advantages of simple structure, long service life and the cost is far lower than the dry gas seal. But under high-speed gas conditions, the temperature of the sealing fluid rises too high and low sealing pressure. This paper studies the gas centrifugal seal with continuous injection and discharge of sealing fluid, establishes a numerical model, simulates the gas-liquid two-phase flow field and temperature field in the sealing chamber. Revealing the effects of the speed and the sealing fluid flux on the sealing performance by comparing the changes in the sealing fluid pressure, temperature, and sealing capacity under different operating conditions. And the reliability of its application to gas mechanical seals is verified through experiments. The results show that at high speeds, the pumping compression effect of the sealing fluid is enhanced, and the sealing capability is improved. Within a certain range, increasing the flux of the sealing fluid can effectively reduce the temperature and prevent the sealing fluid from vaporizing, the continuous injection and discharge structure has a significant effect.

1. Introduction
Centrifugal seal is a device that pumps leaked liquid to high pressure area or forms a liquid barrier to prevent the leakage of high pressure gas through the rotation of rotating parts [1], so as to prevent the leakage of liquid or gas from high pressure to low pressure. As one of the traditional fluid dynamic seals, centrifugal seals have the characteristics of simple structure, no friction and wear of the main seals, and long life compared with commonly used dry gas seal [2-4] and labyrinth seal [5-7]. They have achieved good results in sealing liquids, have good sealing capabilities, and have been widely used in various seals similar to centrifugal pumps. However, when sealing the gas, due to the high-speed rotation of the impeller, the sealing fluid is driven to generate a large amount of stirring heat. This part of the heat cannot be taken away in time, which causes the temperature of the sealing fluid to rise rapidly, or even vaporize, leading to seal failure [8], which severely limits the further development of centrifugal seals in the field of gas seals. The centrifugal seal with continuous injection and discharge of sealing fluid can effectively remove the stirring heat, overcome the problem of the temperature rise of the sealing fluid, improve the sealing ability, and solve the disadvantages of the centrifugal seal application under high-speed gas conditions. Research on it has great significance for promoting the application of the centrifugal seal in the field of gas sealing and energy saving.
The research on centrifugal seals is mainly based on the engineering modification of seals on high-speed rotating equipment such as pump shafts and fans. In recent years, with the development of computational fluid dynamics and computer numerical simulation software, domestic research on the analysis mechanism of internal flow in centrifugal seals has gradually increased. Young C et al. [9] introduced the basic principle of the VOF method in CFD and its advantages in two-phase flow analysis, and used FLUENT software to analyze the two-phase flow field of the centrifugal seal in the steam turbine. Denecke J [10] then studied the geometric model of centrifugal seals in steam turbines, and compared the positions of oil and gas boundaries for centrifugal seals in steam turbines by theoretical calculations and numerical simulation analysis. At the same time, a test system was established and the above-mentioned tests were verified by experiments. But the author did not analyze the heat balance of the centrifugal seal in detail. Jianping Yuan et al. [11] numerically simulated the flow field inside the centrifugal pump with the auxiliary impeller seal, and used a steady-state model to analyze the sealing mechanism and internal flow conditions. The results show that the auxiliary impeller seal has a slight effect on the flow field in the main impeller and the volute. The slight effect, which shows that the centrifugal pump with the auxiliary impeller seal has a higher pressure head but a lower efficiency, and this change becomes more obvious as the flux increases. Long Cai [12] performed numerical simulations on four different sizes of back blades and analyzed the influence of the sealing gap on the sealing pressure. It was found that as the sealing gap decreases, the seal pressure of back blades with different sizes showed an increasing trend. Shujuan Li [13] used a combination of numerical simulation and orthogonal test to analyze the influence of the structural parameters of the auxiliary impeller centrifugal seal on the sealing performance and pump performance, and verified the feasibility of the orthogonal test in analyzing the sealing structure by experiments.

Based on this, this paper uses Fluent software to study the gas-liquid two-phase flow field and temperature field of the centrifugal seal with continuous injection and discharge of sealing fluid, so as to analyze the influence of operating parameters on the sealing performance parameters. Through research, it has been shown that the gas centrifugal seal with continuous injection and discharge of sealing fluid can effectively solve the problems of small sealing pressure and fast temperature rise of the centrifugal seal under high-speed gas conditions.

2. Working principle
The structure of the continuous injection and discharge centrifugal seal is shown in Figure 1. When the gas device is running, the rotating shaft drives the centrifugal impeller to rotate at high speed to promote the flux of the sealing fluid on both sides of the impeller. The sealing fluid with a higher pressure is injected into the sealing chamber from the inlet, flows through the centrifugal impeller through the water groove E on the high pressure side and flows out from the water groove F on the low pressure side. Due to the pressure difference between the two sides of the gas, the fluid in the gap between the impeller and the sealing chamber forms a rotary siphon because of rotation, thereby achieving a non-contact radial seal. Finally, the sealing fluid flowing from F will be recovered from the sealing fluid outlet to the liquid source in the control system.

When the main engine is stopped or running at low speed, a pneumatically controlled sliding valve type parking seal plays a role. The parking seal is an end face mechanical seal, and the axial movement of the spool valve is achieved through a spring and a control gas. At low speeds, the spring force closes the end face of the slide valve and the end face of the impeller to achieve the parking seal. When running at high speed, the spool valve is pushed open by introducing control gas to achieve centrifugal sealing.
In order to calculate the sealing ability of the performance of the centrifugal seal with continuous injection and discharge of sealing fluid, the sealing pressure formula of the seal is derived based on the momentum equation below. Take a fluid element at a radius R, from the momentum equation:

\[ \frac{dp}{dR} = \rho R \omega^2 \]  

(1-1)

Assuming the fluid is incompressible, integrate equation (1-1) from A to B and from D to C:

\[ p_A - p_B = \frac{\rho}{2} \omega_1^2 (R_A^2 - R_B^2) \]  

(1-2)

\[ p_D - p_C = \frac{\rho}{2} \omega_2^2 (R_D^2 - R_C^2) \]  

(1-3)

Among them, A and D respectively indicate the positions of the gas-liquid interface on the high-pressure medium side and the low-pressure atmospheric side. pA is the medium pressure, pD is the atmospheric pressure. B and C are the inner walls of the sealing chamber, and pB = pC. \( \omega_1 \) is the angular velocity of the liquid ring in the chamber 1, \( \omega_2 \) is the angular velocity of the liquid ring in the chamber 2. And \( \rho \) is the density of the blocking liquid. Generally, the difference between \( \omega_1 \) and \( \omega_2 \) can be ignored [14]. When \( \omega_1 = \omega_2 \), according to (1-2), (1-3) can get:

\[ \Delta p = p_A - p_D = \frac{\rho}{2} \omega^2 (R_A^2 - R_D^2) \]  

(1-4)

Under normal circumstances, the speed and the density of the sealing fluid are known. It can be seen that determining the position of the gas-liquid interface is not only a measure of whether the seal has failed, but also a key factor in determining the sealing pressure value.

In the above derivation process, the assumption that the liquid ring angular velocity and the impeller angular velocity on both sides are equal is actually used. In fact, there is a certain difference between the liquid ring angular velocity and the impeller angle due to the viscosity of the liquid. In order to consider the impact of introducing this assumption, Introducing a opposite pressure coefficient k [15], the equation becomes:

\[ \Delta p' = k \Delta p = k \frac{\rho}{2} \omega^2 (R_A^2 - R_D^2) \]  

(1-5)

In general, due to the viscosity of the liquid, there is a speed difference between the rotating part and the stationary part. The actual value of the sealing pressure is smaller than the theoretical value. Therefore, some scholars also refer to the opposite pressure coefficient k as the pressure reduction coefficient [16]. In fact, the opposite pressure coefficient k reflects the influence of the annular vortex
inside the sealing chamber on the average angular velocity of the liquid ring. The larger the opposite pressure coefficient, the higher the degree that the impeller drive the liquid, the better the sealing effect.

Through the above analysis, the key performance parameters of the centrifugal seal with continuous injection and discharge of sealing fluid can be determined as the position of the air-liquid interface on the high and low pressure side, the temperature of the sealing fluid, the pressure of the medium, and the opposite pressure coefficient.

3. Numerical Simulation
The centrifugal seal with continuous injection and discharge of sealing fluid simultaneously achieves the purpose of preventing the leakage of the sealing medium and cooling the sealing chamber by continuously injecting and discharging the sealing fluid, but it also makes the fluid in the sealing chamber in a state of high-speed and high-pressure gas-liquid two-phase flow. In order to simplify the calculation, the following assumptions are made [17]:
(1) The sealing fluid used in the calculation is incompressible fluid by default, and the sealing medium and the atmosphere are air of constant density by default.
(2) During the sealing process, the sealing fluid and the sealing medium will not undergo chemical reaction and physical phase change by default.

3.1. Geometric Model

![Figure 2: Fundamental diagram of the centrifugal seal with continuous injection and discharge of sealing fluid](image)

![Figure 3: Calculation model](image)

In order to reduce the calculation time and improve the calculation accuracy, the ring-shaped water groove on the cavities on both sides of the impeller are simplified into the inlet and outlet of the sealing fluid when performing the modeling.

Figure 3 is the calculation model of the centrifugal sealed flow channel. From left to right are the extended section of the low-pressure side air inlet, the extended section of the sealing fluid outlet, the axial gap 2, the radial gap, the axial gap 1, and the high-pressure side sealing medium. There are seven sections including the extension section of the inlet and the extension section of the sealing fluid inlet. Table 1 shows the main structural dimensions of the calculation domain.

| The main geometric parameters | Value |
|-------------------------------|-------|
| Diameter of impeller D/mm     | 180   |
| Axial clearance a1/mm         | 1.5   |
| Axial clearance a2/mm         | 1.5   |
| Radial clearance b/mm         | 1.5   |
| Radial distance of sealing fluid inlet c/mm | 30 |
| Depth of annular water groove d/mm | 2   |
3.2. Boundary condition and solution

In the calculation process, the gas phase mainly refers to air and sealed medium, and the liquid phase mainly refers to sealing fluid. The gas phase medium is selected as standard air, and the liquid phase medium is selected as water. The remaining operating parameters and boundary conditions are shown in Table 2. The physical properties of the sealing fluid and sealing medium are shown in Table 3.

| parameters                  | Boundary conditions | Value            |
|-----------------------------|---------------------|------------------|
| Low-pressure side air inlet | Pressure-inlet      | $P_{in}=0$MPa (G) |
| High-pressure side seal medium inlet | Pressure-inlet | $P=0.15$MPa (G) |
| Sealing fluid inlet         | Flux-inlet          | $V_{in}=1.544$m/s |
|                             |                     | $m_{in}=4$kg/s   |
| Sealing fluid outlet        | Pressure-outlet     | $P_{out}=0$MPa (G) |
| Sealed chamber              | Wall                | 0r/min           |
| Centrifugal impeller        | Rotation boundary   | $n=4000$r/min   |

Since the default liquid phase is an incompressible fluid, the speed inlet in the sealing fluid inlet is equal to the mass inlet. Considering the annular area of the sealing fluid inlet, the calculated sealing fluid inlet flux is 4kg/s. In addition to the boundary conditions given in the table, the VOF model also needs to determine whether the medium at each inlet and outlet is a gas phase or a liquid phase. In FLUENT, the volume fraction of the second phase is used to represent this concept. In order to accelerate convergence, this paper uses the gas phase medium as the first phase and the liquid phase medium as the second phase. Therefore, the volume fraction of the low-pressure side air inlet and the high-pressure side seal medium inlet is set to 0, and the volume fraction of the sealing fluid inlet and outlet is 1.

| Type           | Density kg/m$^3$ | Viscosity kg/m$^3$·s (Pa·s) | Thermal Conductivity w/m·K | Specific heat J/kg·K | Boiling point °C |
|----------------|------------------|-----------------------------|---------------------------|----------------------|-----------------|
| Water          | 1000             | 0.001003                    | 0.5989                    | 4183                 | 100             |
| Air/sealed medium | 1.225         | $1.7894 \times 10^{-5}$    | 0.0242                    | 1006.43              |                 |

The solid boundary conditions in FLUENT mainly include the constant temperature boundary, constant heat flux density boundary, and convective heat transfer boundary. When the centrifugal impeller rotates at a fixed speed, on the one hand, the end surface of the impeller is used as the constant heat flux density boundary and the stirring heat flux density is given. On the other hand, the circumferential surface of the impeller is taken as the convective heat transfer boundary and the convective heat transfer coefficient is given. The wall surface of the sealing chamber is taken as the constant temperature boundary and its temperature is given.

3.3. Calculation results and analysis

3.3.1. Phase distribution and pressure and temperature distribution. When the gas centrifugal seal works, under the double action of external pressure and centrifugal force of impeller, the sealing fluid is continuously injected into the sealing chamber from the inlet and discharged from the outlet. On the other hand, under the action of centrifugal force of rotating impeller, it is thrown to the inner wall of the chamber, so after the flow is stable, a rotating siphon will be formed in the gap between the impeller and the inner wall of the chamber.
Figure 4 is a gas-liquid distribution cloud diagram of the half section of the sealing chamber at \( z = 0 \), where red represents the liquid phase and blue represents the gas phase. Figure 5 is a cloud diagram of the total pressure distribution of the half section of the sealing chamber at \( z = 0 \). From the figure, it can be seen that the high pressure area is close to the sealing medium side, and the low pressure area is close to the atmospheric side. Moreover, the pressure of the sealing fluid at the sealing fluid inlet is significantly higher than the pressure in other positions in the sealing chamber.

![Figure 4. Two-phase distribution of the half section of the sealing chamber at Z=0](image)

![Figure 5. Pressure distribution of the half section of the sealing chamber at Z=0](image)

Figure 6 shows the temperature distribution in a sealing chamber. It can be seen from the figure that the temperature is higher in the place full of medium and close to the impeller rotation boundary, which is caused by the stirring heat generated by the impeller rotation. And the temperature distribution in the other places full of sealing fluid is more average, which indicates that this method can effectively take away the stirring heat generated by the impeller rotation and reduce the temperature in the sealing chamber.

![Figure 6. Temperature distribution of the half section of the sealing chamber at Z=0](image)

3.3.2. Impact of rotational speed on performance. (1) Effect of rotational speed on pressure and opposite pressure coefficient of blocking fluid.  

The change of the impeller speed will affect the centrifugal force generated by the rotation of the sealing fluid, which will cause the change of the air-liquid interface on both sides of the impeller, and will affect the compression of the liquid in the chamber, which will change the pressure field in the seal. Therefore, it is necessary to study the effect of rotational speed on the sealing performance under the condition that the flux of the sealing fluid is unchanged.

Figure 7 shows the change law of the sealing fluid pressure \( P_{in} \) with the impeller rotation \( n \) when the medium pressure is \( 0.15 \text{MPa} \) and the flux of the sealing fluid is \( 4 \text{ kg/s} \). It can be seen that when the flux of the sealing fluid in and out of the sealing chamber is constant, as the impeller speed \( n \) increases, the required sealing fluid pressure \( P_{in} \) also gradually increases. The pumping compression effect of the sealing fluid is enhanced, which in turn causes an increase in pressure near the sealing fluid inlet. Therefore, in order to ensure a certain sealing fluid flow, a higher sealing fluid pressure is required.
Figure 8 is a curve of the opposite pressure coefficient $k$ with the rotation speed $n$. It can be seen from the figure that when the flux is constant, the opposite pressure coefficient gradually increases with the increase of the rotation speed. This shows that the higher the rotation speed, the greater the impeller's driving degree on the sealing fluid, and the smaller the impact of the annular vortex in the sealing chamber on the sealing performance.

![Figure 7. Effects of speed on the pressure of the sealing fluid](image1)

![Figure 8. Effects of speed on the opposite pressure coefficient $k$](image2)

(2) Effect of rotational speed on the temperature of the sealing fluid

Figure 9 shows the change in the temperature of the sealing fluid $T$ in the sealing chamber with the impeller speed $n$ when the medium pressure is 0.15MPa and the flux of the sealing fluid is 4 kg/s. The increase in temperature $T$ is relatively slow, which means that after the increase of the rotational speed, although the stirring heat generated by the impeller rotation will increase, the continuous injection and discharge of sealing fluid can remove the heat in the chamber in time, thereby effectively reducing the temperature rise of the fluid.

![Figure 9. Effects of speed on the temperature of the sealing fluid in the chamber](image3)

(3) Effect of rotational speed on sealing ability

When the pressure of the medium changes, the centrifugal seal can automatically adjust the position of the air-liquid interface on both sides of the impeller to adapt to the seal of different pressure differences, which makes the sealing device have a certain pressure sealing ability at each speed.

In order to achieve continuous injection and discharge of sealing fluid while effectively sealing, investigate the sealing ability at different speeds when the flux of the sealing fluid is 4kg/s, and the changes of the corresponding pressure of sealing fluid and average temperature.
Figure 10. Effects of speed on the range of the sealing pressure and correlation parameters

3.3.3. Impact of sealing fluid flux on performance. The gas centrifugal sealing with continuous injection-discharge of sealing fluid requires constant renewal of the sealing fluid. Changes in the sealing fluid flux will cause changes in the blocking fluid pressure and temperature, which will affect the flow field and temperature field in the entire sealing chamber. It is necessary to study the influence of sealing fluid flux on sealing performance.

(1) Effect of sealing fluid flux on sealing fluid pressure and opposite pressure coefficient

Figure 11 shows the change rule of sealing fluid pressure pin with flux min when medium pressure is 0.15Mpa and rotating speed is 4000r/min. It can be seen that with the increase of the flux, the pressure of the sealing fluid increases gradually, which is due to the fact that the degree of compression of the impeller to the sealing fluid is basically unchanged when the rotating speed is fixed.

Figure 12 is a curve showing the change of the opposite pressure coefficient k with the flux of the sealing fluid, and it can be seen from the figure that the opposite pressure coefficient does not change basically with the flux of the sealing fluid. This shows that increasing or decreasing the flux of sealing fluid will not affect the annular vortex in the sealing chamber.
Figure 11. Effects of the flux of the sealing fluid on the pressure of the sealing fluid

(2) Effect of sealing fluid flux on sealing fluid temperature

Figure 13 shows the change law of the temperature $T$ of the sealing fluid with the flux $m_{\text{in}}$ when the medium pressure is 0.15MPa and the rotation speed is 4000 r/min. It can be seen from the figure that the temperature of the seal fluid decreases gradually with the increase of the flux. Because the stirring heat generated by the impeller is constant when the speed is constant, and the increase of the flux not only enhances the convection heat transfer between the seal fluid and the chamber, but also increases the heat taken away by the sealing fluid, thus reducing the temperature of the chamber.

Figure 13. Effects of the flux of the sealing fluid on the temperature of the sealing fluid in the chamber

(3) Effect of sealing fluid flux on sealing ability

In order to analyze the effect of the sealing fluid flux on sealing ability at a fixed speed, investigating the sealing ability of the gas centrifugal seal at a speed of 4000r/min and a sealing fluid flux of 4, 4.2, 4.4kg/s. Further study the change curve of the sealing fluid pressure, opposite pressure coefficient and sealing fluid temperature under different medium pressure.

Figure 14. Effects of mass flow rate on the range of the sealing pressure and correlation parameters
It can also be seen from figure 14 that the sealing ability of the gas centrifugal seal does not change substantially when the flux of the sealing fluid changes.

Figure 14 (a) shows that the pressure of the sealing fluid under different flux is basically not affected by the change in the pressure of the medium, and the pressure of the sealing fluid increases as the flux of the sealing fluid increases.

Figure 14 (b) shows the change curve of the temperature of the sealing fluid with the pressure of the medium under different flux. It can be seen from the figure that when the pressure of the medium increases, the temperature of the sealing fluid rises. The reason is that the increase of the pressure of the medium causes the gas-liquid interface on the high pressure side to rise, thereby reducing the cooling area of the sealing fluid. At the same time, the annular vortex flow intensify in the chamber, which generates more heat.

4. Test verification

The centrifugal seal test system is shown in Figure 15. Because the test needs to be performed at the shaft end of the test stand, the prototype uses a double-end structure as shown in Figure 16 (a), and the actual prototype of the test is shown in Figure 16 (b).

![Figure 15. Testing system of the centrifugal seal with continuous injection and elimination of the sealing fluid](image)

![Figure 16. Experimental Prototype of the centrifugal seal](image)

(a) structural drawing  
(b) Prototype

1-Left flinging ring of water;  2-Drive pin;  3-Left centrifugal impeller;  4-Left Drain;  5-Left baffle;  6-Left slide valve;  7-spring;  8-Left end cap;  9-Right end cap;  10-Intermediate chamber;  11- Right gland;  12-Right slide valve;  13-Right centrifugal impeller;  14-Right flinging ring of water;  15-Right baffle;  16- Drive sleeve
In order to study the influence of the rotational speed on the sealing performance separately, in the test, the intake pressure was controlled by a high-precision automatic regulating valve, and the excessive pressure was discharged through the relief valve to stabilize the sealing fluid pressure. At the same time, the speed is adjusted by the inverter, so that the impact of the speed on the performance of the continuous injection and discharge centrifugal seal can be studied separately under the premise that the pressure of the sealing fluid is stable.

![Figure 17. Effects of speed on the maximum of the sealing pressure](image)

During the test, in order to obtain the maximum sealing pressure at each speed as much as possible, it is necessary to gradually increase the pressure of the sealing fluid and the medium until the zero leakage after stabilization is achieved. Recording the medium pressure at this time is the maximum sealing pressure at this speed. Figure 17 shows the relationship between the speed and the maximum sealing pressure. It can be seen from the figure that the maximum sealing pressure increases as the speed increases. This is because the pumping effect of the centrifugal impeller on the sealing fluid is increased when the speed is increased, resulting in the total pressure of the sealing fluid in the chamber is increased. And the medium with higher pressure can be sealed when the fluid level on both sides is not changed much. This test verifies the correctness of the increase in the pressure-sealing ability as the increase of the rotational speed proposed in 3.3.2 of this paper.

![Figure 18. Effects of speed on the temperature of the sealing fluid at the exit](image)

The temperature of the sealing fluid flowing out of the sealing chamber was then measured using a temperature sensor. Figure 18 is the curve of the temperature of the outlet of the sealing fluid when the medium pressure at each speed is the maximum. It can be seen from the figure that the temperature at the outlet of the sealing fluid rises slowly with the increase of the rotation speed in the range of 3500r/min-4300r/min, which is the same as the pattern shown in Figure 9. Then gradually increase the sealing speed to 6000r/min, and found that the temperature of the sealing fluid at the outlet gradually increased, which indicates that when the speed is around 4000r/min, the sealing fluid continuously injected and discharged can take away the stirring heat during the impeller rotation. Then, with the increase of the speed and the accumulation of heat during the test, the temperature of the sealing fluid
that at the outlet increased. This test verifies the feasibility of reducing the temperature rise by continuously injecting the sealing fluid.

According to the working parameters shown in Table 2, the rotational speed of the rotating shaft was fixed at 4000r/min, and the pressure in the surge tank was stabilized at about 0.165 MPa for testing. Then increase the medium pressure, record the medium pressure at this time and the flux and temperature of the sealing fluid at the corresponding outlet.

![Figure 19. Effects of the pressure of the sealing medium on the mass flow rate of the sealing fluid at the exit](image)

It can be seen from Figure 19 that under the same working conditions, the measured medium pressure is lower than the simulation result. The main reason for this phenomenon is the impact of the pressure of the medium and the pressure of the sealing fluid during the sealing operation. A small amount of axial movement will occur, causing the axial sealing gap on both sides to change, which will cause a reduction in sealing pressure.

Figure 19 is the change curve of the flux of sealing fluid at the outlet with the pressure of the medium. It can be seen from the Figure 19 that the flux is stable at about 4 kg/s when the pressure in the surge tank is 0.165 MPa. This verifies that the correspondence between the mass boundary and velocity boundary of the sealing fluid inlet set in the simulation calculation and the sealing fluid pressure calculated are correct.

5. Conclusion

Aiming at the problems of centrifugal seals in the field of gas seals, this paper studies the centrifugal seals with continuous injection and discharge of sealing fluid and draws the following conclusions:

1. The continuous injection and discharge of the sealing fluid structure can effectively remove the stirring heat generated during the rotation of the seal, which solves the problems of low sealing pressure and fast temperature rise when the centrifugal seal is applied to the gas centrifuge machine;
2. The higher the rotation speed and the greater the flux of the sealing fluid, the greater the sealing ability, but at the same time, the temperature of the sealing fluid in the chamber will rise higher, which will cause the vaporization of the sealing fluid. According to the analysis results, within the range of rated speed and sealing fluid flux, the effect of centrifugal seal on centrifugal machinery is significant;
3. The correctness of the numerical simulation calculation was verified by experiments, and it was pointed out that there was a certain fluctuation in the flux of the sealing fluid limited by the test conditions, and the measured temperature was lower than that in the simulation due to the double-end seal structure used in the test.

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