Performance Analysis and Optimization of Internal Spiral Groove Dry Gas Seal Based on Fluid-solid-thermal Coupling Method

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ABSTRACT:  Internal spiral groove dry gas seal (ISGDGS) are used to seal special positions of rotating machinery. There is an urgent need to address the problems caused by contacting of rotating and static rings during high-speed rotation of ISGDGS. The main aim of this study is to investigate the influence of spiral groove structural parameters on ISGDGS sealing performance. Fluid-solid-thermal coupling method was used to analysis the relationship of sealing performance (opening force, leakage, friction power, and gas film stiffness) and spiral-groove structural parameters. The test analysis method is used to study the sealing performance under four different structural parameters, and the obtained trends are the same as the analysis results. According to the analysis results, the value range of structural parameters for optimal sealing performance is proposed This thesis has provided a deeper insight into the influence of the structural parameters of end face on the sealing performance, and provides a basis for its optimal design.

KEY WORDS:  Internal Spiral Groove, Fluid-solid-thermal Coupling, Sealing Performance

NOMENCLATURE

\( a \)  Helix angle
\( r_i \)  Sealing face inner radius
\( r_o \)  Sealing face outer radius
\( r \)  Spiral groove root radius
\( r_g \)  Spiral groove outer radius
\( h_g \)  depth of groove
\( h_o \)  End flow field thickness (film thickness)
\( T \)  Temperature distribution
\( N_g \)  Number of grooves
\( R_w \)  Ratio of groove to weir
\( R_d \)  Ratio of groove to dam
\( r \)  Radial position
\( \theta \)  Angular position
INTRODUCTION

Internal spiral groove dry gas seal (ISGDGS) are used to seal special positions of rotating machinery due to its advantages on pumping out medium and boosting pressure. ISGDGS are non-contact gas seals used in high-speed rotating gas machinery. If the end face parameters are not selected properly, the rotating and static rings cannot be opened at high speed. That scenario will lead to serious consequences, such as burning the seals which means that the sealing ring is malfunctioning [1]. ISGDGS are different from the traditional dry gas seals. There are grooves on the high-pressure inner diameter side, which can ensure the high-pressure medium to be pumped out with a small leakage rate avoiding medium pollution.

In actual working conditions, the state of the sealing end face is affected by structure, fluid and temperature. Recently, researchers have shown an increased interest in coupling analysis which is a research method that comprehensively considers the effects of various aspects. Researchers studied the deformation of the rotating and static rings of dry gas seals [2]. Hu Q [3] established a three-dimensional heat transfer model, and studied the effects of the spindle speed on the structure-thermal coupling deformation. Luo X [4] coupled the end face waviness and taper deformation to perform fluid calculation, and studied the influence of the groove shape on the non-contact sealing performance. Huang W F [5] coupled a multi-physics field to establish a fluid-solid-thermal simulation model to study the mechanical properties of upstream pumping, and revealed the weakening effect of temperature rise and deformation on upstream pumping capacity. Wang J H [6] established a thermo-mechanical coupling model to analyze the friction characteristics of friction pairs. Bai Y G [7] used the Gauss-Seidel block and iterative coupling method to analyze the influence of flow and heat transfer in the internal cavity of the sealed structure and external heat transfer on the seal. Chen H L [8] performed a fluid-solid-thermal coupling analysis on the seal end face, and obtained that the fluid film thickness change on the inner and outer diameter sides was 16%. Some researchers have established micro-gap flow field models based on micro-gap fluid flow characteristics [9-10], two-phase flow particle dispersion mechanism [11], and spiral groove end face lubrication state [12]. Li G Q [13] summarized common seal types such as dry gas seals. Wang R J [14] studied the characteristics of double-end mechanical seals and proposed common applications for double-end mechanical seals. Li F Q [15] summarized the dry gas sealing technology. The scholars have analyzed the dry gas sealing performance of end grooves such as herringbone grooves [16], spiral grooves [17,18], T-shaped grooves [19], double spiral grooves [20-22], and tree grooves [23] and obtain the influence of various structural parameters and working conditions on sealing performance.

Previous researches into dry gas seals mainly have focused on coupling method and the end face groove shape. Far too little attention has been paid to the analysis of the ISGDGS’ end face fluid
considering the heat and end face deformation. This study sets out to explore the influence of the structural parameters of the spiral grooves on the sealing performance using fluid-solid-thermal coupling analysis method. The effects of spiral angle, number of grooves, ratio of weir, ratio of groove to dam, and depth of groove on sealing performance such as opening force, leakage, friction power, and film stiffness were revealed. The findings shed new light on the optimal design of ISGDGS.

1 Structure and principle
Fig. 1 is a schematic diagram of ISGDGS’ structure, which is mainly composed of a rotating ring and a static ring assembly. High pressure gas flows through the seal from right to left. There are micron-sized dynamic pressure groove on the rotating ring. The static ring component is composed of a static ring, a static ring seat, an O-ring, a wave spring, spring pad and snap ring.

![Figure 1. Seal structure.](image)

1- rotating ring; 2- static ring; 3- static ring seat; 4- O-ring; 5- wave spring; 6- spring pad; 7- Snap ring

The gas film structure is shown in Fig. 2. The main leak location is at the contact surface of the rotating ring. When it is stationary, the static ring is tightly fitted with the rotating ring to seal the gas on both sides. At high-speed rotation, as the Figure 3 shows, high pressure gas generates dynamic pressure effect, as the flow channel becomes narrower in the groove. When the dynamic pressure effect is large enough, the end face divides, and a micron-scale air film with a certain rigidity will be formed between the end face. The rigid film is the reason for the sealing effect.

![Figure 2. End face spiral groove structure.](image)

Dynamic pressure effects during rotation is related to the end face structural parameters. The structural parameters of the flow field analysis of the seal are shown in Tab. 1.

![Figure 3. ISGDGS schematic diagram.](image)
Table 1. Structural parameters of the flow field of the seal.

| Structural parameters                      | Value |
|-------------------------------------------|-------|
| Sealing face inner radius $r_i$/mm        | 28.5  |
| Sealing face outer radius $r_o$/mm        | 34.5  |
| Spiral groove root radius $r$/mm          | 27.5  |
| Spiral groove outer radius $r_g$/mm       | 32.5  |
| Spiral groove curve equation $R = r e^{\theta \tan 15^\circ}$ |       |
| depth of groove $h_g$/μm                  | 5     |
| End flow field thickness (film thickness) $h_o$/μm | 3     |
| Number of spiral grooves $N_g$            | 12    |

2 Analysis model

2.1 Gas Film Governing Equation and Solving Conditions

The basic equation describing the pressure distribution of fluid film between sealing faces of the dynamic pressure seal is the pressure control equation (Reynolds equation):

$$
\frac{\partial}{r \partial \theta} \left( \frac{\rho h_o^3}{\eta} \frac{\partial p}{\partial \theta} \right) + \frac{\partial}{r \partial \theta} \left( \frac{r \rho h_o^3}{\eta} \frac{\partial p}{\partial r} \right) = 6 \omega \frac{\partial (p h_o)}{\partial \theta}
$$

(1)

The solving method of governing equation (1) is mainly based on the variational equation of Galerkin method to discretize pressure governing equation and using the finite element method to solve total equation in the calculation domain so as to obtain the pressure value of each node of fluid film after dispersion $\rho$.

(1) Assumptions

In view of the actual characteristics of fluid film on the end face of the dynamic pressure seal, the following reasonable basic assumptions need to be made for fluid film between the seal faces:

① The fluid flows between the gap of end faces is laminar flow.

② The fluid film is thin enough, and dimensions between thickness direction of the working film and structural direction are thousands of times different. It can be assumed that the fluid density and pressure remain unchanged along the working film thickness direction;

③ Fluid does not cause relative sliding when it comes into contact with the rotating and static rings;

④ The speed gradient only considers $\partial U/\partial Z$ and $\partial V/\partial Z$ ignoring other speed gradients, since the working film thickness is micron level, but the speed changes greatly.

⑤ The temperature and viscosity of fluid during the flow process are constant, mainly because the pressure analyzed in this paper is not so high, and the temperature is normal.

⑥ This paper analyzes the steady-state performance, assuming that the fluid film is stable and will not rupture.

(2) Boundary conditions

Solving the pressure control equation must satisfy the following boundary conditions:

① When $r = r_i$, then $p = p_i$; when $r = r_o$, then $p = p_o$.

② The wall surfaces on both sides of the seal in the circumferential direction are periodic boundary conditions:

$$
p(r, \theta) = p(r, \frac{2\pi}{N_g} + \theta)
$$

(2)

2.2 Performance parameter solving equation
After obtaining the pressure distribution of the fluid film on the end surface, use the following equations to obtain performance parameters such as the opening force $F$, the film stiffness $s$, the leakage $q$, and the friction power $P_n$.

1. **Opening force $F$**: The fluid opening force is calculated by integrating the pressure on the area. The opening force calculation formula is shown in (3).

   \[ F = \int_{0}^{2\pi} \int_{r_i}^{r_o} prdrd\theta \]  

2. **Film stiffness $s$**: The fluid film stiffness characterizes the ability of the sealing end face to resist interference. The film stiffness $s$ calculation formula is shown in (4).

   \[ s = -\frac{\partial F}{\partial z} \mid_{h_0} \]  

3. **Leakage $q$**: The radial mass flow through the film thickness $h_0$ and width $rd\theta$ is:

   \[ q_r = \int_{h_0}^{h} \rho u, dz \mid_{r_i}^{r_o} \int_{\theta_i}^{\theta_o} \left( \int_{h_0}^{h} \frac{\rho U}{2\mu} \frac{dp}{dr} \left( z^2 - hz \right) dz \right) r d\theta = -\left( \int_{h_0}^{h} \frac{\rho h^3}{12\mu} \frac{dp}{dr} \right) r d\theta \]  

   Integrate it along the circumference of the leakage surface to get the radial leakage:

   \[ q = \int_{0}^{2\pi} q_r d\theta = -\int_{0}^{2\pi} \frac{\rho h^3}{12\mu} \frac{dp}{dr} r d\theta \]  

4. **Friction power consumption $P_n$**: The friction of the fluid on the end face of the seal mainly overcomes the friction torque $m_0$ generated by the shear force of the fluid viscosity. Obtain the friction torque $m_0$ as:

   \[ m_0 = \int_{0}^{2\pi} \int_{r_i}^{r_o} \tau r^2 drd\theta = \int_{0}^{2\pi} \int_{r_i}^{r_o} \left( \frac{\partial p}{r d\theta} \frac{h}{2} + \mu \frac{\omega r}{h} \right) r^2 drd\theta \]  

   Further solving friction power:

   \[ P_n = m_0 \omega \]  

### 2.3 Fluid-solid-Thermal Analysis Method

Fig. 4 shows the process of fluid-solid-thermal analysis. Under the actual operating conditions, the end surface is deformed by the influence of centrifugal force, pressure and temperature, which causes the thickness of fluid film on the end surface to change, and then affect fluid film pressure distribution and sealing performance. The fluid-solid-thermal coupling method couples the calculation results of fluid, solid and heat, getting the actual fluid film pressure distribution. The pressure distribution contributes in calculating the fluid sealing performance. Since the pressure change of the fluid film has little effect on the temperature, for the sake of calculation, only the influence of temperature on the unidirectional of the fluid film is considered. The specific calculation steps are as follows:

1. Divide the mesh according to the input parameters and calculate the deformation components $\delta_{ij}$ and $\delta_{id}$ of the seal ring;
2. Correct the initial thickness of the fluid film to obtain the initial thickness correction value $h_i$ for the fluid domain, where $h_i = h_0 + \delta_{ij} + \delta_{id}$.
3. The thermal deformations $\delta_{ij}$ and $\delta_{id}$ are obtained through thermal calculation.
4. Obtaining fluid pressure distribution $p_1$ through calculation of the fluid part
5. The pressure $p_1$ is the boundary condition for the new round of solid deformation calculation, and the deformation components $\delta_{ij}$ and $\delta_{id}$ are obtained. The new fluid film thickness is $h_i = h_0 + \delta_{ij} + \delta_{id} + \delta_{ij} + \delta_{id}$. Repeated iterative calculations. When the pressure difference meets the set accuracy ($\varepsilon$), the pressure distribution of the fluid film and the deformation of the seal ring reach a steady state.
2.4 Verification calculation

With this method, the rotating ring end face deformation, the static ring end face deformation and end face fluid film pressure distribution are obtained showing in Fig. 5, Fig. 6, and Fig. 7.

It can be seen from the Fig.5 and Fig.6 that the outside of the end face is higher than the inside. It
indicates that the end face gap is convergent. Convergence gap is conducive to the formation of fluid film, which is beneficial to the stability of the seal and the reduction of the leakage. Due to the existence of the dynamic pressure groove, the deformation of some parts of the end face changes. The figure also shows that the relative deformations of the rotating ring and static change are 1nm and 3nm, respectively, which has a certain impact on the micron-scale fluid film, demonstrating the necessity of fluid-structure-thermal coupling analysis. It can be known from the Fig. 7 that the seal can produce a considerable dynamic pressure effect under working conditions.

3 Test verification

3.1 Test device
The ISGDGS test bench is shown in Fig. 8, and the measuring device is shown in Fig. 9.

![Figure 8. The ISGDGS test bench.](image1)

![Figure 9. high-precision flow meter.](image2)

High pressure gas enters from the N1 port and exits from the N2 port. N3 is a leak measurement port opened on the side of the air atmosphere. N3 is connected to an electronic flow meter to measure the leakage of the seal. The test uses a high-precision flow meter to measure the leakage of the seal, and its measurement accuracy can reach 0.001 L/min.

3.2 Verification analysis
The static ring is softer than the rotating ring, so the wear marks caused by the friction are more obvious on the static ring. The micro-morphology of the static ring end surface before and after operation is shown in Fig. 10(a) and (b) respectively.

![Figure 10. Micro-morphology of static ring.](image3)

(a)  (b)

The flood white point in the picture is the metal antimony immersed in graphite. As can be seen from Fig. 10(a), the end face of the static ring is flat and free of scratches before operation. Fig. 10(b) shows the end face of the static ring after operation. There are regular scratches in the direction of the
lower left corner to the upper right corner after the operation. Using the electronic scale with an accuracy of 0.1g, the static ring mass difference before and after operation is 0.0g. There is almost no wear on the end face at high speeds, which means that the ISGDGS design is feasible.

The test data and calculation data are shown in Fig. 11. What is striking in this figure is the growth of Leakage. The leakage increases with the increase of the rotation speed. The larger the pressure difference, the larger the leakage. The test leak and the calculated leak have the same trend with the rotation speed. But the numerical calculation assumes that the fluid film between the end faces is a stable gas film, the sealing dynamic pressure effect is weak at low speeds, the end face is not opened, and the fluid film is unstable. Therefore, at 2000 r·min⁻¹, the test leakage of 0 L·min⁻¹ is inconsistent with the calculated data. In the case of the seal opening, the deviation between the test and calculation data is very small so the seal analysis calculation is of reference significance.

Figure 11. Test and calculation of leakage under different pressures.

4 Structural parameter analysis results
The deformation of the rotating and static rings and the distribution of the film pressure will affect the sealing performance. The sealing performance is mainly achieved by adjusting the end face parameters such as the helix angle, the number of grooves, the ratio of groove to weir, the ratio of groove to dam, and the depth of groove. The analysis structural parameters are shown in Tab 1. Set the analysis external pressure to 0.15 MPa, internal pressure to 0 MPa, and the initial temperature of the medium to 20 °C.

4.1 Helix Angle
The effect of the helix angle on the spiral groove dynamic pressure sealing performance is shown in Fig. 12. What stands out in this figure is the rapid increase in opening force with the increase of the helix angle. After the helix angle is 15-20°, the increasing trend slows down. The leakage increases with the increase of the helix angle, and the increase rate gradually slows down. This phenomenon demonstrates that within the scope of the study, the increase of the helix angle reduces the resistance of the liquid to enter the spiral groove, the dynamic pressure effect is enhanced, and the leakage and the opening force are large near 22-24 °. Friction power is not greatly affected by the helix angle, and the trend of friction power at high speeds is different from lower rotation speeds. This is mainly because friction power is related to the end surface state and deformation, and the end surface deformation is relatively large at high speed. Increasing helix angle leads to an increase in fluid film stiffness. After turning on, the rotation speed will not affect the process of fluid entering the spiral groove, and the influence of the spiral angle on the sealing performance under different rotation speeds is the same. Increasing the helix angle helps to improve the opening performance, but at the same time the increase in the leakage rate is detrimental to the normal operation of the seal. A smaller helix angle reduces the opening force, which in turn requires a greater rotational speed to balance the closing force, and the seal requires a greater rotational speed to open and ensure the rigidity of the gas film.
Therefore, considering the influence of the helix angle on the sealing performance, it is reasonable to choose a helix angle of 15°~20°.

![Figure 12](image1.png)

**Figure 12.** Effect of helix angle on dynamic pressure sealing performance.

### 4.2 Number of Grooves

The effect of the number of grooves on the sealing performance is shown in Fig. 13.

![Figure 13](image2.png)

**Figure 13.** Effect of the number of grooves on the dynamic pressure sealing performance.

Figure 13(a) reveals that the opening force increases with the increase of the number of grooves, and finally stabilizes. After the number of grooves is 16 to 20, the opening force reaches the maximum. This is because when the number of grooves is small, the dynamic pressure effect of the fluid entering the spiral groove is not obvious. Increased number of grooves mean increased leak paths so the greater the number of slots, the greater the leakage. The leakage reaches a maximum when the number of grooves is about 16. Due to the small overall leakage at low speeds, the variation of the leakage with the number of grooves is not obvious. The increase in the number of grooves makes the state of the end face worse and the frictional heat generation increases. So the friction power increases, there are more number of grooves. The increase of the number of grooves increases the rigidity of the air film, and the increase trend slows down when the number of grooves is 16-20. When the number of grooves exceeds 16, the opening force, leakage rate, stiffness, and friction power tend to stabilize. In order to ensure enough opening force and a small leakage, it is better to take 12-16 grooves.

### 4.3 Ratio of Groove to Weir

The influence of the ratio of groove to weir on the sealing performance is shown in Fig. 14. What can be clearly seen in Fig. 14 is the opening force and leakage rate have a maximum value within the ratio of groove to weir of 0.6 to 0.75. This is because a larger ratio is easier to enter the grooves area than a fluid, resulting in a dynamic pressure effect and a large leakage. When the ratio is too large, the flow channel is overdone, and the dynamic pressure effect is not obvious. The larger ratio, the larger
the average gap between the end faces of the seal will be, which will cause the shear force between the end faces to decrease and the power consumption to decrease. When the ratio is 0.5, the groove area and the weir area is the same, the film pressure distribution is the most uniform, and the stiffness of the liquid film reaches the maximum, so the optimal range of stiffness is 0.4-0.6. Comprehensive analysis of the sealing performance, the ratio of groove to weir should be taken between 0.5-0.6.

4.4 Ratio of Groove to Dam

The effect of the ratio of groove to dam on the sealing performance is shown in Fig. 15. As can be seen from the Fig. 15, there is a maximum opening force when the ratio is 0.5 to 0.7. The leakage increases with the increase of the ratio. This is because a small ratio is not conducive to the fluid entering the trough. A large ratio makes the dam area smaller, the fluid resistance and the dynamic pressure effect decreased, resulting in leakage increased. When the ratio of groove to dam increases, the difference in leakage between high and low rotation speeds becomes larger. This is because the gas is more likely to leak under the large ratio and the taper deformation of the end face. When the ratio is too large, the bearing capacity and throttling capacity of the film decrease, and the stiffness reaches the maximum between 0.55 and 0.75. These findings suggest that the best overall sealing performance is obtained when the ratio of groove to dam is between 0.65 and 0.75.

4.5 Depth of Groove

The effect of groove of depth on sealing performance is shown in Fig. 16. When the groove of depth is less than 12μm, the opening force and leakage rate increase rapidly with the increase of the groove of depth, but when the groove of depth exceeds 12μm, the opening force and leakage rate both slowly decrease. This is because the resistance of the fluid in and out of the groove area is greater due to the large groove of depth, and the dynamic pressure effect decreases. When the groove depth
excessive increase, pressure in the groove area decrease so that the pressure difference between groove areas and dam areas decrease causing the leakage reduced. Power consumption decreases with increasing groove of depth. When the groove of depth is 5-10μm, the gas film can produce a certain dynamic pressure effect and a better bearing capacity. At the same time, the thickness of the gas film is relatively small, and the stiffness reaches the maximum. In summary, when the groove of depth is about 7-10μm, it has better opening force, leakage rate and stiffness.

![Figure 16](a)  ![Figure 16](b)

**Figure 16.** Effect of the groove of depth on the dynamic pressure sealing performance.

### 4.6 Sealing performance comparison test analysis

According to the results of numerical analysis, four structures of T1~T4 were selected for experimental analysis. The specific parameters of T1~T4 are shown in Tab 2.

| structure type | $r_o$/mm | $r_i$/mm | $N_g$ | $h_g$/μm | $R_d$ | $\alpha$/° |
|----------------|----------|----------|-------|-----------|-------|------------|
| T1             | 34.5     | 28.5     | 12    | 5         | 0.7   | 15         |
| T2             | 34.5     | 28.5     | 8     | 5         | 0.7   | 15         |
| T3             | 34.5     | 28.5     | 12    | 10        | 0.7   | 15         |
| T4             | 34.5     | 28.5     | 12    | 5         | 0.72  | 15         |

The test was carried out under three conditions of 0.05, 0.1MPa, and 0.15MPa, and the test data is shown in Figure 17(a), (b)and(c).

![Figure 17](a)  ![Figure 17](b)  ![Figure 17](c)

**Figure 17.** Test data.

The end face is easy to wear at low speed, so the 2000rpm working condition passed quickly in the test. Observe that there is no significant change in the leakage of the four types of seals under the three pressure conditions during this process. In order to observe the trend, temporarily replace the seal leakage at 2000 rpm with 0 leakage.
Under the three conditions of 0.05, 0.1MPa, and 0.15MPa, the leakage of the inner spiral groove dry gas seal increases with the increase of speed. This shows that the increase in rotation speed increases the film thickness and the leakage.

In the speed range of 0–8000rpm, the leakage of T1, T3, T4 three structural seals suddenly increases between 2000–6000rpm. The sealing of structure T2 suddenly increases in leakage between 6000 and 8000 rpm. The sudden increase in leakage means that the dynamic pressure effect generated by the end fluid pushes away the dynamic and static rings. This means that the end face opens when the leakage increases suddenly.

It can be seen from the leakage change trend that the order of opening performance from good to bad is T3, T4, T1, T2. It can be seen that the large depth of groove the opening performance is better. The reduction of the number of grooves has an obvious effect on the opening performance of the seal. The structure with a large ratio groove to dam has a larger leakage at a high speed, and a structure with a small number of grooves has a smaller leakage. It can be seen from the figure that when the structural parameter is T1, the seal opening speed is lower and the leakage is the smallest, which is similar to the numerical analysis result.

5 Conclusions
The study was designed to verify the feasibility of ISGDGS and analyze the impact of end face parameters on its performance. The findings suggest an important role for end face parameters in promoting ISGDGS’ performance. When the helix angle is 15-20°, the number of grooves is 12-16, the ratio of groove to weir is 0.5-0.6, the ratio of groove to dam is 0.65-0.75, and the depth of groove is 7-10 um, the opening force and air film stiffness are relatively large, the leakage and friction power are small, and the seal with the best comprehensive performance is obtained. The correctness of the analysis model established based on fluid-solid-thermal coupling has been verified by the experimental results, laying the groundwork for future research into the optimal design of ISGDGS.

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