The thermal and mechanical deformation study of up-stream pumping mechanical seal

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Abstract. Taking the viscosity-temperature relationship of the fluid film into consideration, a 3-D numerical model was established by ANSYS software which can simulate the heat transfer between the upstream pumping mechanical seal stationary and rotational rings and the fluid film between them as well as simulate the thermal deformation, structure deformation and the coupling deformation of them. According to the calculation result, thermal deformation causes the seal face expansion and the maximum thermal deformation appears at the inside of the seal ring. Pressure results in a mechanical deformation, the maximum deformation occurs at the top of the spiral groove and the overall trend is inward the mating face, opposite to the thermal deformation. The coupling deformation indicate that the thermal deformation can be partly counteracted by pressure deformation. Using this model, the relationship between deformation and shaft speed and the sealing liquid pressure was studied. It's found that the shaft speed will both enhance the thermal and structure deformation and the fluid pressure will enhance the structure deformation but has little to do with the thermal deformation. By changing the sealing material, it's found that material with low thermal expansion coefficient and low elastic modulus will suffer less thermal-pressure deformation.

1. Introduction
In 1984 John Crane mechanical seals company posted a new concept about mechanical seal----up-stream pumping: open grooves in the sealing surface, and a small amount downstream leakage fluid will be pumped back. Subsequently, the company developed and produced the 8000 series of spiral groove upstream pumping mechanical seals, marking the upstream pumping mechanical seal officially put into use. Theoretically, compared with ordinary mechanical seal, upstream pumping mechanical seal with a non-contact, zero leakage, low maintenance cost and reliable operation, etc., however in fact, the research on sealing-lubricating mechanism of upstream mechanical seal and the design and application of it, etc. still need further studies.

Upstream pumping mechanical seal micro-gap lubrication problem has attracted attention of many scholars and researchers, and has made important progress. In the early time, the theory of narrow groove has been adopted [2,3]. It is assumed that the number of the spiral groove is infinite, and the geometric discontinuity and complex boundary conditions have been ignored, then the problem has been simplified to a one-dimensional model and been solved. Later, analytic methods have been used. The problem was simplified to a two-dimensional model. And the performance of mechanical seal could be predicted through solving Reynolds equation[4,5,6]. However, for the mechanical seal with...
complex structure such as spiral groove, analytical methods obviously was not in conformity with the practical engineering. Recently, the numerical simulation method has been adopted. The numerical simulation has been carried on the spiral groove three-dimensional micro clearance which could effectively predict the pressure distribution in flow field inside and the flow regularity[7,8].

The mechanical seal internal flow field research has been effectively promoted. However, the thermal characteristics of the mechanical seal and the thermal - mechanical deformation has become a key point in studying the sealing mechanism and performance, especially the research about the mechanical seal with surface texturing. And it is found that the thermal deformation can make the flat sealing surface deform along the radial. When the sealing surface produced dry friction, the deformation was more serious, which made the inner sealing surface more wear. So we must try to reduce the thermal deformation[9]. Jianfeng Zhou et al. established the heat transfer model of mechanical seal ring of static pressure type, and the ratio was determined that the friction heat between rotating ring and static ring accounting for the total heat. Then the coupling process between the friction heat of liquid film and the thermal deformation of the sealing surface were studied. And the rotating speed w of rotating ring, and the relationship between the film thickness \( t \) of the inside diameter and the radial angle \( \beta \) of the sealing ring surface after deformation[10,11] were determined. Cicone er al. studied the non-adiabatic heat characteristics of the liquid film between mechanical sealing surfaces, and on the basis of this, he obtained accurate liquid film opening force, the liquid film stiffness and friction torque. But it was assumed that the viscosity of the liquid film is constant, finally it had a larger gap between the leak he obtained and the real value[12]. Qiu yifan et al. considered the viscosity-temperature relationship of sealing medium, and solved the temperature distribution of the circular ring with spiral groove and the pressure distribution of liquid film. But they didn't further solve the thermal and mechanical deformation[13].

Based on the existing studies, this dissertation simulate the spiral groove upstream pumping mechanical face seal by combine the finite volume method(FVM) and finite element method(FEM) together, acquire the temperature and pressure field, and further imitate the thermal deformation, stress distortion and the coupling deformation. All this will provide some reference to the studying and design of upstream pumping mechanical face seal.

2. Geometry Model and Boundary Conditions
Taking the complexity of the study object, the feasibility of the modeling and the scientificity of the result into consideration, the following basic assumptions are made:

1) The performance of the sealing rings' materials does not change with temperature.
2) Ignores the impact of surface roughness.
3) Do not consider sealing ring radiation heat transfer, ignoring the heat generated by the seal ring stir and all heat is generated by the fluid film inner viscous dissipation.
4) Assumes the rings take away all the viscous dissipative energy.
5) The temperature distribution within the mechanical seal ring does not change with time, that it is a steady-state CFD model.

The 104-50 mechanical face seal was chosen and the matting rings profile parameters are shown in Figure 1. The spiral groove is set at the rotational ring end face as shown in Figure 2, and the parameters of it are shown in Table 1. Where, \( \gamma = (r_e-r_o)/(r_i-r_o), \quad \beta = l_{wg}/l_{wt}, \quad l_{wg} \) is the length of groove, and \( l_{wg} \) is the length of groove plus seal dam.

The calculation model is built in the ANSYS Workbench platform, the fluid film is fluid domain and it's calculated by Fluent; the mating rings is structure domain, the heat transfer is calculated by Steady-State Thermal module, and the deformation is calculated by Steady Structure module.

As shown in Figure 3, the seal rings' temperature and stress fields are mainly produced by the fluid film. \( Q_1 \) is the micro gap liquid viscous heat and it is respectively transfered to the static ring and rotational ring. \( Q_1 \) is the heat that rotational ring obtained and mainly passed to the sealing liquid by wall A-D; \( Q_2 \) is the heat that passed to the sealing liquid by stationary ring wall I-L; \( Q_3 \) and \( Q_4 \) is the
heat passed to the air respectively by rotational ring and static ring. Due to heat is generated by the fluid film as well as the sealing end face's pressure is gained by the fluid film, then the 3-D micro gap need to be calculated first.

![Figure 1](image1)

**Figure 1.** geometry parameters of the a) rotational and b) stationary rings: unit:mm

![Figure 2](image2)

**Figure 2.** Rotational face geometry of upstream pumping mechanical seal

![Table 1](image3)

| Parameter                        | Value |
|----------------------------------|-------|
| Inner Radius $r_i$ (mm)          | 25    |
| Outer Radius $r_o$ (mm)          | 32    |
| Spiral Angle $\alpha$ (°)        | 18    |
| Groove-to-dam Ratio $\gamma$     | 0.5   |
| Groove-to-land Ratio $\beta$     | 0.7   |
| Number of Grooves $N_g$          | 12    |
| Groove depth $h_c$ (μm)          | 6     |

![Table 1. The geometric parameters of the spiral grooves](image4)

**Table 1.** The geometric parameters of the spiral grooves

As the figure 4 represents, there are two parts of the fluid film, the groove liquid which is filled in spiral groove and the gap liquid which is between seal faces. The groove liquid is set to rotation as the given rotation speed and the gap liquid is set to stationary. The end face of rotational ring, the end face and side faces of groove are set to rotational wall boundary. The end face of stationary ring is set to stationary wall boundary. The pressure of input equals the pressure of medium and the pressure of output equals atmospheric pressure. The fluid is contacted with the rotational wall and stationary wall,
the heat transfer between them is set to conversation heat transfer boundary condition. \( \alpha_{fs} \) and \( \alpha_{fr} \) is the heat transfer coefficient between the fluid film and the rotational and stationary ring, and they can be calculated by following formula[14,15]:

\[
\alpha_{fr} = \alpha_{fs} = \alpha' = 0.664 k_f pr^{0.33} \left( \frac{u_f}{v L_e} \right)^{0.5}
\]

(1)

Where

\[
L_e = \pi (r_o + r_i), \quad u_f = \frac{(r_o + r_i)}{4}, \quad pr = \frac{c_{pr} \mu_f}{k_f}
\]

(2)

In calculating the solid domain, the use of boundary conditions set as follows: wall A ~ D and I~ L contact with the sealing liquid, high-pressure side, the pressure is medium pressure \( P_1 \); wall E ~ H and O ~ S for the low-pressure side in contact with air \( P_0 \); wall surface of the stationary ring in contact with the wall surface N ~ O, is set to a fixed constraint fix support; wall A ~ H in contact with the wall surface of the spring seat, when stabilized, the radial displacement of the spring seat as 0, so the set displacement constraints displacement, so that the axial displacement is 0. Movement loop operation, the energy obtained from the liquid film to seal liquid cooling and air cooling are set to each wall convective heat, each wall convection coefficient is calculated by the following equation [14-15]:

1) Rotational Wall A~D:

\[
\alpha_{mr} = 0.135 \lambda \left( 0.5 \text{Re}_c^2 + \text{Re}_u^2 \right)^{0.33} / D_r
\]

(3)

where: \( \text{Re}_c = \omega D_r^2 / v \), reflect the impact of the media rotary stirring Reynolds; \( \text{Re}_u = v D_r / v_c \), reflect the impact of the media lateral flow around Reynolds; \( \lambda \) is the thermal conductivity of the sealed medium, [W/m*K]; \( P_r = \mu_c \cdot c_{pr} / k_i \) is Plante constant, \( D_r \) is the equivalent diameter of the outer peripheral ring, [m]; \( v \) is the axial velocity of surrounding medium of the rotational ring, [m/s]; \( \omega \) is the shaft speed, [rad/s].

2) Stationary Wall I~L:

\[
\alpha_{ms} = 0.023 \lambda \cdot \text{Re}^{0.8} \cdot \text{Pr}^{0.4} / S_s
\]

(4)

where \( S_s \) is gap between the static ring and the seal chamber, [m].

3) Wall E–H and O–S:

Since the sealing ring inside the boundary is contacted with air, it can be seen from the above formula 4, relative to the ring, the convective heat transfer coefficient of static ring outer diameter of the inner boundary of the convective heat transfer coefficient is negligible, generally during the ANSYS simulation, its convective heat transfer coefficient only need to properly set a reasonable value.

3. Calculation and analysis

3.1. Features of Thermal Deformation

In this part, the case that the shaft speed is 1000rpm and the medium pressure(gauge pressure) is set as an example for calculation and analysis. When Calculating, the film thickness is assumed to be stable and set as 3 \( \mu \) m, the choice of dynamic and static ring material parameters as shown in Table 2; the sealing liquid material parameters as shown in Table 3. \( \mu_f \) variation as a function of temperature follows the exponential law as follows: \( \mu_f = \mu_0 e^{-\gamma (T - T_0)} \), is implemented in Fluent by UDF
programming; Table 4 shows the exact convection heat transfer coefficient in this case.

**Table 2. Parameters of the sealing rings**

| Parameter                  | Rotational ring | Stationary ring |
|----------------------------|-----------------|-----------------|
| material                   | sic             | Grp*            |
| Density \( \rho_f \) (kg/m\(^3\)) | 3150            | 1810            |
| Specific Heat \( c_{ps} \) (J/kg\(\cdot\)K) | 710             | 880             |
| Thermal Conductivity \( k_s \) (W/m\(\cdot\)K) | 150             | 45              |
| Coefficient of Thermal Expansion \( \alpha \) (10^-6 k\(^{-1}\)) | 4.3             | 6.2             |
| Poisson's Ratio \( \gamma \) | 0.27            | 0.26            |
| Young's modulus \( E \) (GPa) | 380             | 25              |

**Table 3. Parameters of the sealing liquid**

| Parameter                  | Value |
|----------------------------|-------|
| \( T_\infty \) (K)        | 300   |
| Density \( \rho_f \) (kg/m\(^3\)) | 880   |
| Specific Heat \( c_{pf} \) (J/kg\(\cdot\)K) | 1670  |
| Thermal Conductivity \( k_f \) (W/m\(\cdot\)K) | 0.15  |
| Viscosity \( \mu_f \) (Pa\(\cdot\)s) | 0.0035 |
| viscosity temperature coefficient \( \gamma \) | 0.3   |

**Table 4. The convective heat transfer coefficient of the rings**

| Boundary  | A–B | B–C | C–D | E–H | H–A | I–J | J–K | K–L | L–S |
|-----------|-----|-----|-----|-----|-----|-----|-----|-----|-----|
| convective coefficient \[ w/m^2 \cdot k \] | 2091.3 | 1956.9 | 1824.7 | 4 | 1956.9 | 1016.1 | 1271.6 | 1629.7 | 4 |
| Temperature [k] | 300 |

**Figure 5. Temperature distribution on the seal face**

Due to the liquid film is thickness, temperature distribution along the radial liquid film is almost consistent, so the temperature field distribution is mainly distribution along the direction of the seal face. Figure 5 shows the temperature distribution along static ring end face of upstream spiral groove.
mechanical seal The figure shows that in the inner diameter of the sealing ring, the temperature is lower. This is because here is backflow area, and the temperature close to the reflux temperature, artificially set to 300 k. And then into the spiral groove area, temperature increases, when arrives at sealing weir, the temperature reaches the maximum value. This is due to the friction of buffer fluid in the spiral groove, the temperature of the buffer fluid increases from the inner diameter to the outer diameter. While in sealing weir area, relative linear velocity is the largest, and the thickness of liquid film is relatively thinner, so the temperature reached the maximum in the sealing weir area.

Figure 6 shows the temperature profile of the stationary ring. The figure shows that the highest temperature occurs at the outer side of the seal end face, the temperature is gradually reduced from the seal end face along the axial direction, to the lowest one on the back of the sealing ring. As the interior of the seal ring touches the air, and the exterior of the ring touches the liquid, the heat transfer rate of the interior is lower than the exterior.

Figure 7 shows the thermal deformation figure of sealing ring. When calculating the thermal deformation, the sealing medium pressure and liquid film pressure were ignored, but the fixed constraint on lateral side of static ring and displacement constraint on lateral side of rotating seal ring were retained, and then thermal deformation were figured out. The figure shows that the thermal deformation embodies in expansion with heat from the side face of sealing ring to seal face, and the maximum thermal deformation appeared in the inside of seal face. After deformation the inner side of the sealing end face is higher than the outside, the dry friction can easily occur on the end face.

Figure 6. Temperature distribution of stationary seal ring

Figure 7. Temperature Deformation of stationary seal ring along axial direction

3.2. Features of Mechanical Deformation

Figure 8 shows the static pressure cloud picture of upstream pumping mechanical seal face. It depicts that the largest pressure appears on top of the spiral groove, higher than the medium pressure, and this is result of the pumping effect of spiral groove. Due to the existence of the spiral groove, leakage liquid on low pressure side will be pumped to high pressure side, causing the rise of pressure on the top of spiral groove. With minimal pressure on the inside of spiral groove, partial vacuum appears, and it is easy to produce cavitation. The figure shows that the pressure distribution on pumping mechanical seal face of spiral groove is uneven. The maximum pressure is very high (17 times of medium pressure), and the minimum pressure is very low. This big difference will obviously lead to deformation of the seal face.

Figure 9 shows deformation of seal ring end face. It depicts that the mechanical seal face is all compressive deformation, and pressure deformation and thermal deformation are in the opposite direction. The maximum deformation appears on the top of the spiral groove because of the highest pressure there. Figure shows the largest directional force deformation (Z Axis) is 0.48717 μm, and the
maximum directional thermal deformation (Z Axis) is 1.4219 μm. In the same working condition, the force deformation is much smaller than thermal deformation.

**Figure 8.** Pressure distribution on the seal face

**Figure 9.** Pressure Deformation of stationary seal ring along axial direction

### 3.3. Features of Thermal-Mechanical Coupling Deformation

Figure 10 is a deformation state after coupling of thermal and mechanical deformations. By contrast to Figure 8, coupling deformation and thermal deformation share the same tendency, but the amount of coupling deformation is decreased, it is because of a large thermal deformation, and the opposite direction of the mechanical and thermal deformation, which has an offset function. Therefore, as the final deformation result, the maximum deformation occurs in the inside of the end face of the seal ring, and appears to be an expansion deformation. In addition, the deformation decreases gradually outwardly.

### 3.4. Effect of operating conditions on the deformation

#### 3.4.1. Effect of the sealing shaft speed on the deformation

Figure 11 shows a relationship of opening force, end face temperature and deformation amount with the shaft speed in upstream pumping
mechanical sealing. According to the figure, with the constant of medium pressure, the film's opening force and end surface temperatures are increasing with the increase of shaft speed in case of film thickness remaining constant. Accordingly, the thermal deformation and mechanical deformation are also increasing with the speed increasing. Besides, it can be seen from the figure that the thermal - mechanical coupling deformation is between the thermal deformation and the mechanical deformation, which illustrates thermal deformation can be partially offset by mechanical deformation at different rotating speeds.

**Figure 11. Shaft Speed and Deformation Relationship**

3.4.2. Effect of the medium pressure on the deformation. Figure 12 shows a relationship of opening force, end face temperature and deformation amount with the medium pressure in upstream pumping mechanical sealing. According to the figure, when the rotational speed remains at 1000rpm, medium pressure increases, the opening force is increased gradually, while the temperature of the end face increases slowly. Therefore the corresponding changing amount of thermal deformation is small, and changing amount of mechanical deformation is large. However, the thermal deformation is greater than the mechanical deformation, which will increase the offset of thermal deformation to further reduce the amount of deformation.

3.5. Effect of the sealing material on the deformation

According to section 3.4, the deformation amount of the sealing ring is enlarged along with the increase of medium pressure and rotating speed, so the study will be performed in the working condition of the speed below3000rpm and the medium pressure of 0.8Mpa, to determine the effects of sealing materials on deformation amount. Table.5 shows the physical parameters of other three common sealing materials. Table.6 shows the largest deformation of sealing ring of different materials. According to Table.5 and Table.6, thermal deformation is related to the material expansion coefficient, while the mechanical deformation is related to elastic modulus E. The small thermal expansion coefficient of the seal ring produces small thermal deformation; while small elastic modulus produces a large mechanical deformation. In the working condition of large parameters, the material with small thermal expansion coefficient and low elastic modulus is selected ,to help reduce the thermal deformation, meanwhile, it will increase the offsetting effect of the mechanical deformation on the thermal deformation. Eventually coupling deformation reaches a minimum value. Table.6 Alumina ceramic material shows a verification of this conclusion.
Table 5. Three additional common mechanical seal materials

| Parameter | \( \rho \) (kg/m³) | \( E \) (Gpa) | \( \alpha \) (10⁻⁶/K) | \( \lambda \) | \( \gamma \) |
|-----------|----------------|-------------|-----------------|----------|-------|
| Ywn8      | 14700          | 600         | 5.1             | 100      | 0.25  |
| Copper alloy | 8600      | 93         | 17.2            | 120      | 0.34  |
| Alumina ceramic | 3400         | 200        | 5               | 12       | 0.22  |

Table 6. The max deformation of each material when rpm is 3000 and the media pressure is 0.8MPa

| Rotational Ring Material | Stationary Ring Material | Max Axial Thermal Deformation (µm) | Max Axial Mechanical Deformation (µm) | Max Axial Coupling Deformation (µm) |
|-------------------------|--------------------------|------------------------------------|--------------------------------------|------------------------------------|
| SIC                     |                          | 2.2879                             | -0.069729                            | 2.2326                             |
| Grp*                    |                          | 2.8564                             | -1.0716                              | 2.0212                             |
| Ywn8                    |                          | 2.5939                             | -0.045442                            | 2.5577                             |
| Copper alloy            |                          | 9.1747                             | -0.251042                            | 8.9844                             |
| Alumina ceramic         |                          | 1.8742                             | -0.14142                             | 1.7594                             |

4. Conclusion
1) This paper established a Thermal-Mechanical Coupled model of spiral groove upstream pumping mechanical seal, and performed a numerical simulation of upstream pumping mechanical seal 3-D micro gap and sealing ring. The temperature and stress fields of the upstream pumping mechanical seal are finally obtained, along with the resulting thermal, mechanical and coupling deformation;
2) Simulation results of heat, power and coupling deformation show that the upstream pumping mechanical seal surfaces expand outward with the thermal deformation, but the mechanical
deformation compresses the surface to the inner side. The thermal deformation changes rapidly, but can be partially offset by mechanical deformation;

3) Effect of operating conditions on the deformation illustrates that the increase of shaft speed comes along with the increase of thermal deformation and mechanical deformation. Medium pressure exerts greater impact on mechanical deformation than thermal deformation. The increase of the sealing medium pressure is companied by the offsetting increase of thermal deformation by mechanical deformation at the same speed, causing the total amount of deformation to reduce;

4) Within the scope of this study, while under fast shaft speed and high sealing medium pressure condition, the deformation is mainly caused by thermal. In order to decrease the deformation, the seal material with small coefficient of thermal expansion could be selected, meanwhile, smaller elastic modulus material can be chosen to increase the mechanical deformation, which will enhance the offsets that the mechanical deformation brings to thermal deformation.

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