Simulation calculation of the effects of seal face width on mechanical seal performance

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Abstract. In order to research and master the effects of seal face width on mechanical seal performance, based on the fractal models of average temperature, friction coefficient and leakage rate of seal faces, the influencing factors of seal face width on mechanical seal performance were analyzed by simulation calculation. The calculation results show that the average temperature of seal faces increased approximately linearly, the friction coefficient and leakage rate of seal faces decreased with the increasing seal face width. The decrease amplitudes of friction coefficient and leakage rate became smaller with the increasing seal face width. The effect degrees of seal face width on friction coefficient and leakage rate increased with the increasing sealed fluid pressure. The change of seal face width has little effect on leakage rate when the sealed fluid pressure was low.

1. Introduction

Seal face width is one of the key structural parameters of mechanical seals, which has an important impact on the temperature, friction and leakage characteristics of the seal faces[1]. Cheng et al.[2] measured the temperature of seal face with two widths. The results show that the seal face width is an important factor affecting the temperature rise of seal faces. The narrow face width can significantly reduce the temperature of seal faces. Li et al.[3] experiments show that reducing the seal face width can effectively reduce the temperature rise of the annular seal space under the condition of insufficient mechanical oil. At present, the seal face width of contacting mechanical seals at home and abroad tends to narrow. However, too narrow seal face will increase the contact pressure, wear and leakage of the seal faces[1]. Therefore, it is necessary to choose a reasonable seal face width by comprehensive consideration.

Research results show that the contour curves of the end faces for mechanical seals have isotropic fractal characteristics with no relevance to scale[4]. In this paper, some fractal parameters with scale-independence were adopted to characterize surface morphology of the end faces for mechanical seals. Based on the fractal models of average temperature, friction coefficient and leakage rate of the end faces for mechanical seals established by Wei et al.[5-7], the effects of seal face width on mechanical seal performance were analyzed by simulation calculation.

2. Calculation model

2.1 Calculation models of average temperature and friction coefficient of seal faces
Usually, the rotating ring and stationary ring of mechanical seals are paired with a hard and a soft material. Wei et al. [5-6] equated the contact between hard ring and soft ring as the contact between rigid ideal smooth plane and rough surface, the calculation models of average temperature and friction coefficient of the end faces for mechanical seals were established. Their expressions are respectively

\[ t_m = \left[ \frac{f(p_r + Bp)v_mA_n}{m_rA_n \text{tanh}(m_rL_r) + m_rA_n \text{tanh}(m_rL_s)} \right]^{\frac{1}{f}} \tag{1} \]

\[ f = \frac{\pi^{(8-D)/2}(r_1^2 - r_2^2)}{45(\pi - 2)2^{(2-D)/2}} \left[ \frac{D}{2-D} \right]^{(2-D)/2} \psi^{12-Dn/4} \mu_n \phi_c \] \[ + f_c \left(1 - \frac{K_m}{p_s + Bp} \right) \] \tag{2}

where \( t_m \) is average temperature of the seal faces; \( f \) is friction coefficient of the seal faces; \( p_r \) is spring pressure; \( B \) is balance factor, and the value of it depends on the structure of mechanical seal; \( p \) is sealed fluid pressure; \( v_m \) is average linear velocity of the seal face; \( A_n \) is seal band area, and \( A_n = \pi(r_1^2 - r_2^2) \); \( r_1, r_2 \) are inner radius and outer radius of the seal face, respectively; \( m_r \) is coefficient of heat transfer of the rotating ring; \( \lambda_r \) is equivalent heat conductivity of the mosaic rotating ring material; \( A_{cs} \) is axial cross-sectional area of the equivalent cylinder of the rotating ring; \( L_m \) is length of the equivalent cylinder of the rotating ring; \( m_s \) is coefficient of heat transfer of the stationary ring; \( \lambda_s \) is heat conductivity of the integral stationary ring material; \( A_{cs} \) is axial cross-sectional area of the equivalent cylinder of the stationary ring; \( L_s \) is length of the equivalent cylinder of the stationary ring; \( t_f \) is average temperature of the fluid in annular seal space; \( D \) is profile fractal dimension of the soft ring; \( G \) is profile characteristic length scale of the soft ring; \( \psi \) is expansion coefficient in fractal region; \( \mu_{af} \) is dynamic viscosity of liquid film; \( n \) is rotating speed; \( \phi \) is contact factor; \( b_m \) is asperity bearing area ratio of seal faces, and its specific calculation method is referred to reference [8]; \( f_c \) is contact friction coefficient of asperities of the seal faces; \( K_m \) is film pressure factor.

The specific calculation methods of the related parameters in equations (1) and (2) are referred to references [5,6].

The dynamic viscosity of liquid film can be determined by the average temperature of the seal face. The relationship between dynamic viscosity and temperature of liquid film can be derived from the following viscosity-temperature equation:

\[ \mu_m = \mu_0 \exp\left[-\alpha(t_m - t_0)\right] \tag{3} \]

where \( \mu_m \) is dynamic viscosity of liquid film at temperature \( t_m \); \( \mu_0 \) is dynamic viscosity of liquid film at reference temperature \( t_0 \); \( \alpha \) is viscosity-temperature coefficient, when the sealed fluid is water, take \( \alpha=0.0175 \).

It can be seen from equations (1), (2) and (3) that there exists a mutual coupling relationship between the average temperature and friction coefficient of the seal faces. Hence, a trial calculation method is adopted to obtain the average temperature and friction coefficient accurately, and the specific calculation process is referred to reference [5].

2.2 Calculation model of leakage rate of seal faces

Wei et al. [7] introduced the pressure flow rate factor to reflect the effect of actual rough surface on leakage passage, fractal expression of pressure flow rate factor was derived, and leakage fractal model of the end faces for mechanical seals based on average film thickness and pressure flow rate factor was established. The expression was

\[ q = \frac{(\pi - 2) \vartheta G_m}{48 \pi \mu_m} \left[ \frac{2 \pi}{\sqrt{\pi}} \left( \frac{2-D}{D} \right)^{(2-D)/2} \psi^{12-Dn/4} G^{D-1} A_n^{(2-D)/2} (1-b_m)^{(4-D)/2} \right]^{3} \left[ \frac{p - \rho_m L_m n}{3000} \right] \] \tag{4}
where $\varphi_r$ is pressure flow rate factor; $r_m$ is average radius of the seal face; $b$ is seal face width; $\rho_m$ is density of fluid film at temperature $t_m$.

The pressure flow rate factor can be calculated by the following formula

$$\varphi_r = 10 - 0.9\exp(-0.56\lambda) \quad (\lambda > 0.5)$$  \hspace{1cm} (5)

where $\lambda$ is the ratio of film thickness.

The fractal expression of the ratio of film thickness between the end faces for mechanical seals is

$$\lambda = \frac{(\pi - 2)^{2-D}}{\pi^{(6-D)/2}} \left( \frac{2 - D}{D} \right)^{(2-D)/2} \left[ (2 - D)\ln1.5 \right]^{1/2} l_f \gamma^{-D} \psi^{(2-D)/3} \Lambda_m \left( 2 - D \right)^{1/2} \left( 1 - b_m \right)^{4-D/2}$$  \hspace{1cm} (6)

where $l_f$ is the sampling length for measuring the contour of the seal face.

### 3. Simulation calculation and result analysis

Take mechanical seal with inward leakage for example. The rotating ring is mosaic structure, stationary ring is integral structure. The performance parameters of seal rings are shown in Table 1, and the structural parameters of seal faces are shown in Table 2. The sealed fluid is water with the average temperature $t_f = 20 ^\circ\text{C}$. When calculating, spring pressure $p_s = 0.15\text{MPa}$, rotating speed $n = 2900 \text{rpm}$, fractal dimension $D = 1.636$, characteristic length scale $G = 5.7 \times 10^{-9} \text{m}$, contact friction coefficient of asperities $f_c = 0.1$ body are taken.

### Table 1. Performance parameters of seal rings.

| Seal ring | Material | Heat conductivity (W·m⁻¹·℃⁻¹) | Elastic modulus (MPa) | Poisson’s ratio | Compressive yield strength (MPa) |
|-----------|----------|--------------------------------|----------------------|----------------|---------------------------------|
| Rotating ring | Seal face | Hard alloy YG8 | 80 | 6×10⁵ | 0.24 | — |
| Seating ring | 301 | 26.8 | 2.23×10⁵ | 0.29 | — |
| Stationary ring | Carbon graphite M106K | 15 | 1.6×10⁴ | 0.20 | 200 |

### Table 2. Structural parameters of seal faces.

| Inner diameter $d_1$ (mm) | Outer diameter $d_2$ (mm) | Seal band area $A_0$ (mm²) | Seal face width $b$ (mm) | Balance factor B |
|---------------------------|---------------------------|---------------------------|--------------------------|-----------------|
| 1 | 72 | 76 | 465 | 2 | 1.48 |
| 2 | 71 | 77 | 697 | 3 | 1.16 |
| 3 | 70 | 78 | 930 | 4 | 1.00 |
| 4 | 69 | 79 | 1162 | 5 | 0.91 |
| 5 | 68 | 80 | 1395 | 6 | 0.84 |
| 6 | 67 | 81 | 1627 | 7 | 0.80 |

### 3.1 Effects of seal face width on average temperature of the seal faces

The relationship between the average temperature $t_m$ and the seal face width $b$ under different sealed fluid pressures $p$ is obtained by simulation calculation as shown in Figure 1. As can be seen from Figure 1 that the average temperature of the seal faces increase approximately linearly with the increase of seal face width, and the higher the sealed fluid pressure, the better the linearity of the relationship curves between average temperature and seal face width.
3.2 Effects of seal face width on friction coefficient of the seal face
The relationship between the friction coefficient \( f \) and the seal face width \( b \) under different sealed fluid pressures \( p \) is obtained by simulation calculation as shown in Figure 2. As can be seen from Figure 2, the friction coefficient of the seal faces decreases with the increase of seal face width, and the decrease amplitude of friction coefficient becomes smaller with the increase of seal face width. The effect degrees of seal face width on friction coefficient of the seal faces increase with the increase of sealed fluid pressure.

![Figure 2. Relationship between \( f \) and \( b \).](image1)

3.3 Effects of seal face width on leakage rate of the seal face
The relationship between the leakage rate \( q \) and the seal face width \( b \) under different sealed fluid pressures \( p \) is obtained by simulation calculation as shown in Figure 3. As can be seen from Figure 3, when the sealed fluid pressure is low, the seal face width has little effect on the leakage rate of the seal faces, and the leakage rate decreases slightly with the increase of seal face width. When the sealed fluid pressure is high, the leakage rate of the seal faces decreases with the increase of seal face width, and the decrease amplitude of leakage rate becomes smaller with the increase of seal face width.

![Figure 3. Relationship between \( q \) and \( b \).](image2)

4. Conclusions
(1) Seal face width is one of the key structural parameters of mechanical seals, which has an important impact on the working performance of mechanical seals.

(2) The average temperature of the seal faces increases approximately linearly with the increase of seal face width, and the higher the sealed fluid pressure, the better the linearity of the relationship curves between average temperature and seal face width.
(3) The friction coefficient and leakage rate of the seal faces decrease with the increase of seal face width, and the decrease amplitudes of friction coefficient and leakage rate become smaller with the increase of seal face width. The effect degrees of seal face width on friction coefficient and leakage rate increase with the increase of sealed fluid pressure. The change of seal face width has little effect on leakage rate when the sealed fluid pressure is low.

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