Effects of seal face width on the contact characteristics between the end faces for contacting mechanical seals

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Abstract. In order to research and master the effects of seal face width on the contact characteristics between the end faces for contacting mechanical seals, based on the contact fractal model and average film thickness fractal model of the seal face, the effects of seal face width on the contact characteristics between the end faces for contacting mechanical seals was analyzed by simulation calculation. The calculation results show that with the increasing seal face width, the film pressure factor of seal faces increased linearly, the average film thickness increased continuously, but the increase amplitude decrease d gradually, the asperity bearing area ratio decreased rapidly at first and then changed slightly, the elastic contact area ratio increased approximately linearly, the elastic-plastic contact area ratio and the plastic contact area ratio decreased approximately linearly. There exist elastic, elastic-plastic and plastic deformed asperities at the same time between the seal faces. Among them, the proportion of elastic contact area was the largest and that of plastic contact area was the smallest.

1. Introduction
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[1]. The contact between the end faces of contacting mechanical seal is the contact between rough surfaces with different roughness, and the actual contact only occurs on the asperities with higher friction surface. Studying the contact characteristics between the end faces for mechanical seals plays an important role in analyzing their sealing, friction, wear and heat transfer properties. Based on the fractal theory, Wei et al. [2-4] studied the deformation property of end face asperity of contacting mechanical seals and the contact characterizations of end faces in mechanical seals running-in. Cheng et al.[5] and Li et al.[6] studied the effects of seal face width on mechanical seal performance by experiments. Mastering the effects of seal face width on the contact characteristics between the end faces for contacting mechanical seals has important significance both in the design of seal face structure. However, the research on the effects of seal face width on the contact characteristics between the end faces for contacting mechanical seals has not been paid much attention.

Research results show that the contour curves of the end faces for mechanical seals have isotropic fractal characteristics with no relevance to scale [7]. In this paper, based on the fractal models of contact and average film thickness of the end faces for mechanical seals established by Wei et al.[3, 4, 8, 9], the effects of seal face width on the contact characteristics between the end faces for contacting mechanical seals were analyzed by simulation calculation.
2. Calculation model

2.1 Contact fractal model of the end faces for mechanical seals

Usually, a mechanical seal friction pair is composed of a rotating ring and a stationary ring, and they are made of a hard material and a soft material, respectively. Wei et al. [8] equated the contact between the hard ring and the soft ring end faces of mechanical seals as the contact between the rigid ideal smooth plane and the rough surface. Taking into account the elastic deformation, plastic deformation and elastic-plastic deformation of the contact asperities of the seal faces, a contact fractal model of the end faces for mechanical seals was established. Their expressions are:

\[ p_c = EG^{D-1} \eta_r(D) b_m^{D/2} \left[ g_2(D)b_m^{D/2} - a_i^{n(D-2)/2} \right] + 1.1K_f \sigma_2, g_2(D)b_m^{D/2}a_i^{n(D-2)/2} - \\
K(D)g_2(D)b_m^{D/2} \left\{ \frac{3}{(a_i^v - a_{pt}^v)^2} \left[ f(3) - 2f(2)a_{pt}^v + f(1)a_{pt}^v \right] - \\
\frac{2}{(a_i^v - a_{pt}^v)^3} f(4) - 3f(3)a_{pt}^v + 3f(2)a_{pt}^v - f(1)a_{pt}^v \right\} \]

\[ (1 < D < 2, \ D \neq 1.5) \]  \hspace{1cm} (1)

\[ p_c = \frac{\pi^{3/4}}{6^{3/4}} \psi^{16} E \left( \frac{b_m}{3 \psi^{3/4} a_v} \right) + 1.1 \times 3^{3/4} K_f \sigma_2, \psi^{16} b_m^{3/4} a_v^{3/4} - \\
\frac{3^{3/4}}{4} \psi^{16} b_m^{3/4} K(D) \left\{ \frac{3}{(a_i^v - a_{pt}^v)^2} \left[ f(3) - 2f(2)a_{pt}^v + f(1)a_{pt}^v \right] - \\
\frac{2}{(a_i^v - a_{pt}^v)^3} f(4) - 3f(3)a_{pt}^v + 3f(2)a_{pt}^v - f(1)a_{pt}^v \right\} \]

\[ (D = 1.5) \]  \hspace{1cm} (2)

Among them, \( E = \left(1 - \nu_1^2 + 1 - \nu_2^2 \right)^{-1} \), \( G^* = \frac{G}{\sqrt{A_n}} \), \( b_m = \frac{A_n}{A_v} \), \( a_i^v = \pi G^2 \left( \frac{20E}{11K_f \sigma_2} \right)^{2(D-1)} \),

\[ a_{pt}^v = \frac{G^2 \left( \frac{\pi^{3-D} E}{900 \times 4^D \sigma_2} \right)^{3(D-1)}}{A_n} \], \( K_f = 1 - 0.228 f_c \), \( K(D) = 1.1K_f \sigma_2^y \), \( 2G^{D-1} a_i^{n(D-2)/2} \),

\[ g_1(D) = \frac{2^{(7-3D)/2} \pi^{1(D-1)/2} D^{D/2}}{3(3 - 2D)} \psi^{2-D/D^2} a_v, \ g_2(D) = \left[ \frac{2 - D}{D^{(1-D)/2}} \right]^{(3-2D)/2} \psi^{2-D/D^2} a_v, \ g_3(D) = \left( \frac{D}{2D - D} \right)^{(D-D)/2} \psi^{2-D/D^2} a_v, \]

\[ g_4(D) = \frac{D}{2} \left( \frac{2 - D}{D} \right)^{D/2} \psi^{2-D/D^2} a_v, \]

where \( p_c \) is face pressure of mechanical seal; \( E \) is comprehensive elastic modulus, \( E_1, E_2 \) are elastic modulus of hard ring and soft ring, respectively; \( \nu_1, \nu_2 \) are poisson’s ratios of hard ring and soft ring, respectively; \( \sigma_2 \) is compressive yield strength of the soft ring; \( G^* \) is dimensionless characteristic length scale; \( G \) is profile characteristic length scale; \( D \) is profile fractal dimension; \( A_i \) is seal band area; \( r_1, r_2 \) are inner radius and outer radius of the seal face, respectively; \( b_m \) is asperity bearing area ratio of the seal faces; \( A_r \) is real contact area of the seal faces; \( a_{pt}^v \) is dimensionless critical elastic deformation micro-contact area; \( a_{pt}^v \) is dimensionless critical elastic deformation micro-contact area; \( a_i^v \) is expansion coefficient in fractal region; \( K_f \) is friction correction coefficient; \( f_c \) is contact friction coefficient of asperities; \( f(n) \) is intermediate function, and \( n \) is an integral number between 1~4.

According to the axial force balance condition of mechanical seal, it can be obtained:
\[ p_r = p_s + (B - K_m)p \]

where \( p_s \) is spring pressure; \( B \) is balance factor, and the value of it depends on the structure of mechanical seal; \( K_m \) is the film pressure factor; \( p \) is sealed fluid pressure.

When the end faces of contacting mechanical seals are in mixed film friction state, the film pressure factor can be calculated according to the following formula:[8].

\[ K_m = \frac{3 \rho \omega^2 (r_2^2 - r_1^2) [1 - \ln (r_2/r_1)]}{40 p \ln (r_2/r_1)} + \frac{r_2^2 [2 \ln (r_2/r_1) - 1] + r_1^2}{2(r_2^2 - r_1^2) \ln (r_2/r_1)} \]

where \( \rho \) is the density of fluid film; \( \omega \) is angular velocity.

2.2 Fractal model of asperity contact area ratio in different deformation state of the seal faces

The contact characteristics of the seal faces can be characterized by the proportion of the elastic contact area, the elastic-plastic contact area and the plastic contact area of asperities in the real contact area of the seal faces. The expressions of elastic contact area ratio \( T_{re} \), elastic-plastic contact area ratio \( T_{rep} \) and plastic contact area ratio \( T_{rp} \) of the end faces for mechanical seals are respectively [3]

\[ T_{re} = \frac{A_{re}}{A_r} = 1 - \left[ \frac{\pi D}{82 - D} \frac{G^{*2}}{b_m} \left( \frac{20E}{11K_f \sigma_{2y}} \right)^{\frac{2}{D-1}} \right]^{2-D \frac{2}{4}} \]

\[ T_{rep} = \frac{A_{rep}}{A_r} = \left[ \frac{\pi D}{82 - D} \frac{G^{*2}}{b_m} \left( \frac{20E}{11K_f \sigma_{2y}} \right)^{\frac{2}{D-1}} - \left[ \frac{D}{2} \frac{G^{*2}}{b_m} \left( \frac{\pi^{3-D} E^2}{900 \times 4^D \sigma_{2y}^2} \right)^{\frac{1}{D-1}} \right]^{\frac{2-D}{4}} \right]^{2-D \frac{2}{4}} \]

\[ T_{rp} = \frac{A_{rp}}{A_r} = \left[ \frac{D}{2} \frac{G^{*2}}{b_m} \left( \frac{\pi^{3-D} E^2}{900 \times 4^D \sigma_{2y}^2} \right)^{\frac{1}{D-1}} \right]^{\frac{2-D}{4}} \]

where \( A_{re} \), \( A_{rep} \) and \( A_{rp} \) are elastic contact area, elastic-plastic contact area, and plastic contact area of asperities of the seal faces, respectively.

2.3 Average film thickness fractal model of the end faces for mechanical seals

The clearance between the end faces of contacting mechanical seals is composed of surface roughness and waviness. Fluid film with local discontinuous can be formed between the end faces of rotating and stationary rings of mechanical seals in mixed film friction state[10]. Fluid film thickness of the seal faces reflects the contact state between the end faces of rotating and stationary rings, and is an important parameter affecting the working performance of the mechanical seal. Wei et al.[9] based on the contact fractal model of end faces for mechanical seals, the average film thickness fractal model between the end faces for contacting mechanical seals was established by solving the volume of micro-voids on the end face. The expression is:

\[ h_0 = \frac{(\pi - 2) 2^{(2-D)/2} D^{(2-D)/2}}{\pi^{(6-D)/2}} \left( \frac{2 - D}{D} \right)^{(2-D)/2} \psi^{-(2-D)/2} G^{D-1} A_h^{(2-D)/2} (1 - b_m)^{(4-D)/2} \]

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 $t_f = 20\,^\circ C$. 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·K⁻¹) | Elastic modulus (MPa) | Poisson’s ratio | Compressive yield strength (MPa) |
|----------------|-------------------|--------------------------------|-----------------------|-----------------|----------------------------------|
| Rotating ring  | Hard alloy YG8    | 80                             | $6\times 10^3$        | 0.24            | –                               |
| Seating ring   | 301               | 26.8                           | $2.23\times 10^5$     | 0.29            | –                               |
| Stationary ring| Carbon graphite M106K | 15                         | $1.6\times 10^4$      | 0.20            | 200                             |

Table 2. Structural parameters of seal faces.

| Inner diameter \( d_1 \) (mm) | Outer diameter \( d_2 \) (mm) | Seal band area \( A_n \) (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 film pressure factor and average film thickness of the seal faces
The relationship curves of film pressure factor $K_m$ and average film thickness $h_0$ with seal face width $b$ under different sealed fluid pressures $p$ are obtained by simulation calculation, as shown in Figure 1 and Figure 2, respectively. As can be seen from Figure 1 that the film pressure factor of the seal faces increases linearly with the increase of seal face width and decreases with the increase of seal fluid pressure. As can be seen from Figure 2, the average film thickness of the seal faces increases with the increase of seal face width and decreases with the increase of seal fluid pressure. The increase amplitude of average film thickness decreases gradually with the increase of seal face width.

![Figure 1. Relationship between $K_m$ and $b$.](image1)

![Figure 2. Relationship between $h_0$ and $b$.](image2)
3.2 Effects of seal face width on asperity bearing area ratio and contact area ratio of the seal faces

The relationship curves of asperity bearing area ratio $b_m$, elastic contact area ratio $T_e$, elastic-plastic contact area ratio $T_{ep}$ and plastic contact area ratio $T_p$ with seal face width $b$ under different sealed fluid pressures $p$ are obtained by simulation calculation, as shown in Figure 3~Figure 6, respectively. As can be seen from Figure 3, the asperity bearing area ratio of the seal faces decreases rapidly at first and then changes slightly with the increase of seal face width and increases with the increase of seal fluid pressure. The higher the sealed fluid pressure, the greater the effect degrees of the seal face width on the asperity bearing area ratio of the seal faces. From Figure 4 to Figure 6, it can be seen that with the increase of seal face width, the elastic contact area ratio of the seal faces increases approximately linearly, the elastic-plastic contact area ratio and the plastic contact area ratio of the seal faces decrease approximately linearly. With the increase of sealed fluid pressure, the elastic contact area ratio increases, the elastic-plastic contact area ratio and plastic contact area ratio decrease. The smaller the sealed fluid pressure, the greater the effect degrees of the seal face width on the elastic contact area ratio, elastic-plastic contact area ratio and plastic contact area ratio. Comparing the size of the elastic contact area ratio, elastic-plastic contact area ratio and plastic contact area ratio, it can be seen that there exist elastic, elastic-plastic and plastic deformed asperities at the same time between the seal faces. Among them, the elastic contact area accounts for the largest proportion, more than 80%, and the plastic contact area accounts for the smallest proportion, less than 5.5%.

Figure 3. Relationship between $b_m$ and $b$.

Figure 4. Relationship between $T_e$ and $b$.

Figure 5. Relationship between $T_{ep}$ and $b$.

Figure 6. Relationship between $T_p$ and $b$. 
4. Conclusions
(1) With the increase of the seal face width, the film pressure factor of seal faces increases linearly, while the average film thickness of seal faces increases continuously, but the increase amplitude decreases gradually.

(2) With the increase of seal face width, the asperity bearing area ratio of the seal faces decreases rapidly at first and then changes slightly, the elastic contact area ratio of the seal faces increases approximately linearly, the elastic-plastic contact area ratio and the plastic contact area ratio of the seal faces decrease approximately linearly.

(3) There exist elastic, elastic-plastic and plastic deformed asperities at the same time between the seal faces. Among them, the elastic contact area accounts for the largest proportion, more than 80%, and the plastic contact area accounts for the smallest proportion, less than 5.5%.

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