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Experimental Verification on Friction Factor Fractal Model of the End Faces for Mechanical Seals

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Abstract. In order to verify the correction of the friction factor fractal model of the end faces for mechanical seals, some tests for mechanical seal were conducted on the self-designed mechanical seal testing device. Experimental sealed medium was 15°C water. A series of experimental data of friction factor were obtained under different spring pressures and sealed fluid pressures as well as at various rotating speeds. Compared the theoretical calculation data with those of experimental, the results show that the friction factor variation of theoretical was the same as those of experimental with the change of the working parameters, the relative error decreased with the increase of rotating speed and sealed fluid pressure, and the maximum relative error was less than 5.5% when the rotating speed reached the normal working speed of 2900 rpm.

1. Introduction

Friction factor of end faces is an important parameter to characterize the working performance between the end faces for mechanical seals. Wei, et al characterized surface morphology by using fractal parameters with scale-independence, described the effect of real rough surface on liquid film viscous shear stress by introducing contact factor, established the coupled model of friction factor and average temperature of the end faces for contacting mechanical seals, and analyzed the effects of operating parameters and surface topography fractal parameters on friction factor [1]. But it was not experimentally verified. In this paper, experimental verification on friction factor fractal model of the end faces for contacting mechanical seals was conducted.

2. Friction Factor Fractal Model of the End Face

The fractal expression of friction factor of the end face for contacting mechanical seal is [1]

$$f=\frac{\pi^{(8-D)/2}\left(r_2^3-r_1^3\right)\left(\dfrac{D}{2-D}\right)^{(2-D)/2}\psi^{(2-D)^2/4}\mu_m n\phi_c}{45(\pi-2)2^{(3-D)}\left(r_2^2-r_1^2\right)G^{(D-1)}A_n^{(2-D)/2}\left(1-b_m\right)^{(4-D)/2}(p_s+Kp)}+f_c\left(1-\frac{K_m p}{p_s+Kp}\right) \quad (1)$$

where r_1, r_2 are inner radius and outer radius of the end faces respectively; D is profile fractal dimension of the soft ring; G is profile characteristic length scale of the soft ring; ψ is expansion coefficient in fractal region; μ_m is dynamic viscosity of liquid film; n is rotating speed; A_n is nominal contact area; b_m is asperity bearing area ratio of seal face, and it can be seen in Ref.[2]; ϕ_c is contact factor, and reflects the effect of the rough surface on liquid film viscous shear stress; p_s is spring pressure; K is load factor, and the value of it depend on the geometrical structure ; p is sealed fluid pressure; f_c is contact friction factor of asperities of the seal face; and K_m is film pressure factor.

Contact factor ϕ_c can be expressed as [3]

$$\phi_c = \begin{cases} \exp(-0.6912 + 0.782\lambda - 0.304\lambda^2 + 0.0401\lambda^3) & \lambda \le 3 \\ 1 & \lambda > 3 \end{cases} \tag{2}$$

Where λ is the ratio of film thickness.

The fractal expression of the ratio of film thickness between the end faces of friction pair for mechanical seals is [1]

$$\lambda = \frac{(\pi-2)2^{(4-D)}}{\pi^{(6-D)/2}}\left(\frac{2-D}{D}\right)^{(2-D)/2}\left[(2-D)\ln 1.5\right]^{1/2} lr^{(D-2)}\psi^{-(2-D)^2/4}A_n^{(2-D)/2}\left(1-b_m\right)^{(4-D)/2} \tag{3}$$

Where lr is the sampling length.

Substituting Eq. (3) into Eq. (2), the contact factor ϕ_c, of mechanical seal under the conditions of certain operating parameters and profile fractal parameters can be obtained.

It can be known from Eq. (1) that liquid film dynamic viscosity μ_m has a great effect on friction factor f, and μ_m can be determined by average temperature t_m. The relationship between dynamic viscosity and temperature of water is [4]

$$\mu_m = 0.001\exp\left[-0.0175(t_m - 20)\right] \tag{4}$$

WEI, et al [5] simplified mechanical seal ring as equal cross-sectional equivalent cylinder, average temperature calculation formula between end faces for contacting mechanical seal was deduced

$$t_m = \frac{f p_g v_m A_n}{\left[m_r\lambda_r A_{cr}\tanh(m_r L_r) + m_s\lambda_s A_{cs}\tanh(m_s L_s)\right]} + t_f \tag{5}$$

where v_m is the average linear velocity of the seal face; m_r is the coefficient of heat transfer of the rotating ring; λ_r is the heat conductivity of the rotating ring; A_{cr} is the axial cross-sectional area of the equivalent cylinder of the rotating ring; L_r is the length of the equivalent cylinder of the rotating ring; m_s is the coefficient of heat transfer of the stationary ring; λ_s is the heat conductivity of the stationary ring; A_{cs} is the axial cross-sectional area of the equivalent cylinder of the stationary ring; L_s is the length of the equivalent cylinder of the stationary ring; and t_f is the average temperature of the fluid in the annular seal space.

It can be seen from Eqs. (1), (4) and (5) that there exists a mutual coupling relationship between the friction factor f and the average temperature t_m. Hence, a trial method is adopted to obtain the friction factor accurately, and the calculation process is shown in Fig.1.

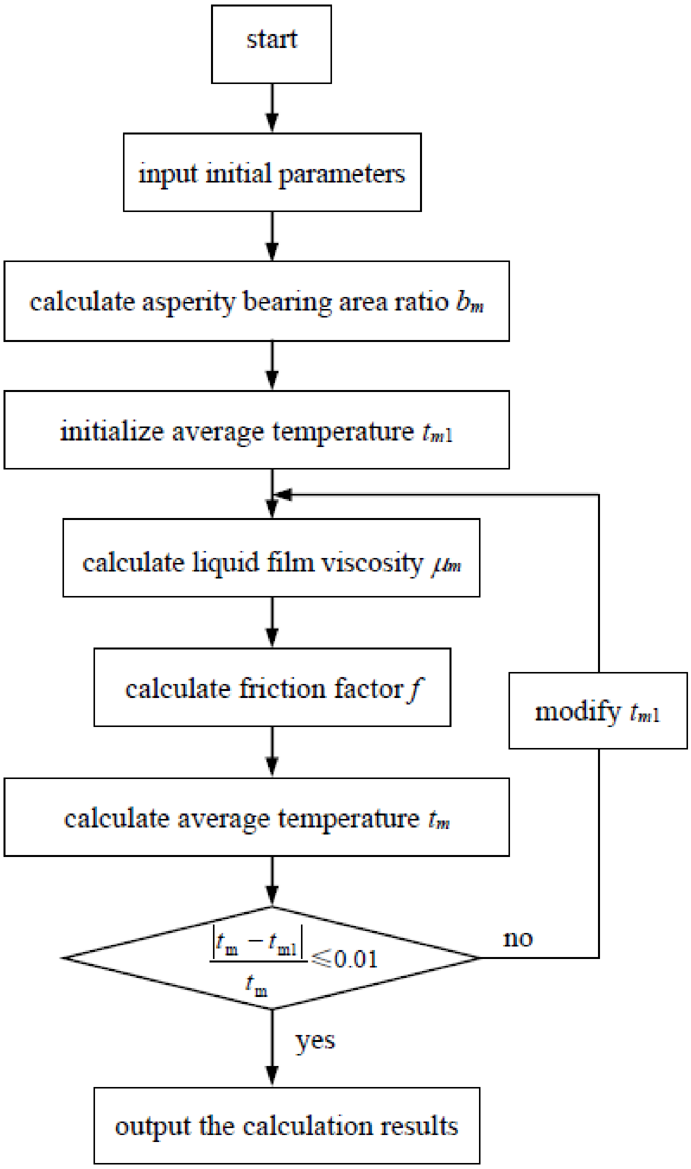


Figure 1. Flowchart for calculating

3. Experiments of Friction Factor of the End Faces

Experiments were conducted on the self-designed mechanical seal testing device [6], the surface profile of soft ring was measured by AF-LI stylus profilometer.

3.1. Specimens and Experimental Parameters

Specimens are B104a-70 mechanical seals. The rotating ring is inlaying narration structure. The stationary ring is monolithic structure. The seal face parameters are inner diameter d_1=69 mm, outer diameter d_2=78 mm, nominal contact area A_n=1039 mm^2, and load factor K=0.895. Performance parameters of seal rings are listed in Table 1.

The experimental sealed medium is water with a flow rate $q_x=5\times10^{-4}$ m^3·s^{-1} and the average temperature t_f= 15°C. The friction factors were measured under different spring pressures and medium pressures, and at various rotating speeds. The time of each experiment was 3min.

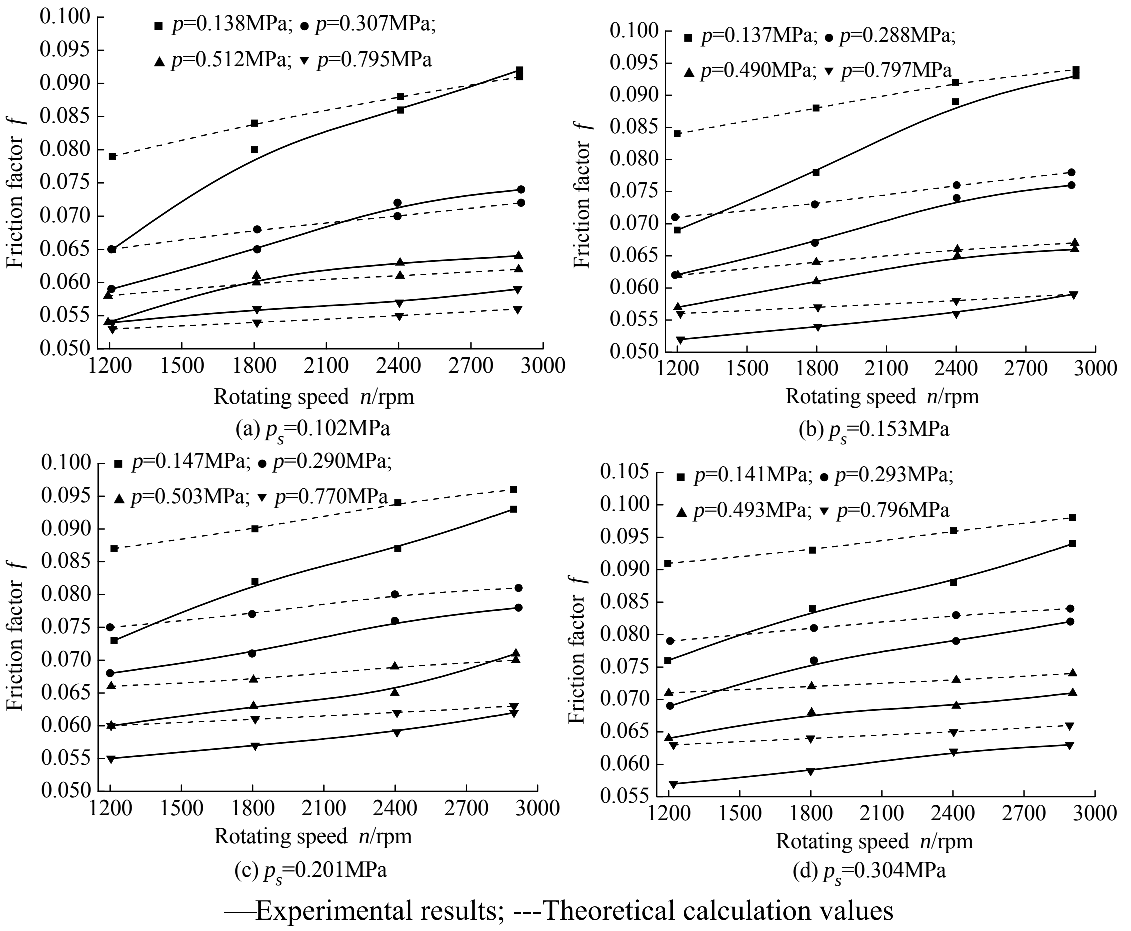
Table 1. Performance parameters of seal rings

Seal ring		Material	Heat conductivity /[W· (m °C)$^{-1}$]	Elastic modulus /MPa	Poisson's ratio	Compressive yield strength/ MPa
Rotating ring	Seal face	Hard alloy YG8	80	6×10^5	0.24	—
	Seating ring	301	26.8	2.23×10^5	0.29	—
Stationary ring		Carbon graphite M106K	15	1.6×10^4	0.20	200

3.2. Experimental Method

The surface profile of the soft ring was measured, and the fractal parameters were calculated before the tests. Mechanical seal specimen was installed on the testing device, the operating parameters were adjusted, and the experiment was conducted. The friction factor was measured in operating. After the test was finished, the tested mechanical seal specimen was taken out and cleaned. The surface profile of the soft ring was measured, and the fractal parameters were calculated. Steps above were repeated.

3.3. Contrastive Analysis between Theoretical and Experimental Friction Factor



—Experimental results; ---Theoretical calculation values

Figure 2. Comparison between theoretical calculation values and experimental results of friction factor

Fig. 2 illustrates the comparison between theoretical results and experimental data of the friction factor. In Fig.2, the experimental results are the average values of the testing date after eliminating several singular values, and the theoretical friction factors are obtained on the assumption that the fractal parameters of the end faces are the same under the same spring pressures, which are the average values of the data measured before and after each test.

It can be seen from Fig.2 that with the changes of spring pressure, sealed fluid pressure and rotating speed, the theoretically calculated friction factor presents nearly the same change regularity as the experimental result. When both the rotating speed and sealed fluid pressure are low, the differences between the theoretical values and experimental data of friction factor are relatively large, and the maximum relative error is 21.7%. However, the differences between the theoretical values and experimental data of friction factor are low when the rotating speeds are higher. The maximum relative error is 9.1% when the rotating speed is 2400 rpm, and the maximum relative error was less than 5.5% when the rotating speed reaches the normal working speed of 2900 rpm.

4. Conclusion

Compared the theoretical calculation data with those of experimental, the results show that the friction factor variation of theoretical is the same as those of experimental with the change of the working parameters, the relative error decreases with the increase of rotating speed and sealed fluid pressure, and the maximum relative error is less than 5.5% when the rotating speed reached the normal working speed of 2900 rpm.

The errors between the theoretical and experimental results result mainly from the assumptions made for the established model and the liquid film hydrodynamic effect enhanced by the taper caused from the thermal deformation and the manufacturing errors of the end faces. It may be necessary to improve the proposed model by considering the coupling effect of the deformation, temperature and friction characteristic of the seal faces.

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