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Simulation Calculation of Temperature of the End Face for Mechanical Seals Based on Fractal Theory

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Abstract. In order to study the effects of operating parameters and surface topography on temperature of the end faces for mechanical seals, an average temperature calculation model of end faces for mechanical seals was established based on the contact fractal model and the friction factor fractal model, adopting fractal parameters to characterize surface topography, and simplifying the seal ring as equal cross-sectional equivalent cylinder. Influencing factors of average temperature were analyzed by simulation calculation for friction pair faces matched by cemented carbide YG8-carbon graphite M106K of partial balance mechanical seal. Theoretical calculation results show that the average temperature of seal face increased linearly with the increase of spring pressure or sealed fluid pressure, and increased approximately linearly with the increase of rotating speed. The average temperature increased with the increase of fractal dimension and with the decrease of characteristic length scale of the soft ring. The change of average temperature was smaller when the fractal dimension was smaller and the characteristic length scale was larger, while the average temperature increased rapidly when the fractal dimension was larger and the characteristic length scale was smaller.

1. Introduction

Temperature of seal faces is a key parameter to effect the working performances of the end faces for mechanical seals. Under the normal working conditions of mechanical seals, because of mutual attaching and relative sliding of seal rings, the generated friction heat could cause temperature rise of seal rings, especially of the seal faces. Most of the seal rings are narrow. The deformation of the seal face generated by thermal load has little effect on the working performances of the mechanical seals. Therefore, the average temperature of the seal faces can be substituted for the actual complex temperature field, which can satisfy the demand for engineering accuracy\cite{1, 2}.

Research results show that the profile curves of the end faces for mechanical seals before and after wearing have isotropic fractal characteristics with no relevance to scale\cite{3}. In this paper, some fractal parameters with scale-independence were adopted to characterize surface morphology of the end faces for mechanical seals. Simplifying the seal ring as equal cross-sectional equivalent cylinder, on the basis of the contact fractal model and the friction factor fractal model established by Wei, et al\cite{4,5}, an average temperature calculation model of end faces for contacting mechanical seals was established, and the effects of working parameters and fractal parameters of surface topography on average temperature of the end faces for mechanical seal were analyzed by simulation calculation according to established model.
2. Establishing model

2.1 Heat balance of friction pair for mechanical seals

Under normal working conditions, the temperature of mechanical seal friction pairs depends mainly on the friction heat $Q_f$ of the friction pair comprised of the rotating ring and stationary ring. The generated friction heat is transmitted to the sealed fluid and then taken away. The heat balance equation can be expressed as[1]

$$Q_f = Q_r + Q_s$$  

(1)

where $Q_r$ is heat of fluid in the annular seal space transmitted through rotating ring, $Q_s$ is heat of fluid in the annular seal space transmitted through stationary ring.

For contacting mechanical seals, the friction heat of the seal faces can be calculated by[6,7]

$$Q_f = f p g v_m A_n$$  

(2)

where $f$ is friction factor of the seal face; $p_g$ is specific load of the seal face, $p_g = p_s + K p_s$; $p_s$ is spring pressure; $v_m$ is average linear velocity of the seal face; $A_n$ is seal band area, $A_n = \pi (r_2^2 - r_1^2)$; $r_1$, $r_2$ are inner radius and outer radius of the seal face respectively.

The heat transfer process, in which the friction heat is transmitted from the outer circumference of the seal rings to the fluid in annular seal space, can be regarded as a convective heat transfer process of a hollow cylindrical fin (equivalent cylinder) heated only on one side. The heat transmitted to the fluid in the annular seal space through the rotating and stationary rings are as follows, respectively[1,8]

$$Q_r = m_r \lambda r A_{cr} (t_m - t_f) \tanh(m_r L_r)$$  

(3)

$$Q_s = m_s \lambda s A_{cs} (t_m - t_f) \tanh(m_s L_s)$$  

(4)

where $m_r$ is coefficient of heat transfer of the rotating ring, $m_r = \frac{h_r C_r}{\lambda r A_{cr}}$; $m_s$ is coefficient of heat transfer of the stationary ring, $m_s = \frac{h_s C_s}{\lambda s A_{cs}}$; $\lambda r$ is heat conductivity of the rotating ring; $\lambda s$ is heat conductivity of the stationary ring; $A_{cr}$ is axial cross-sectional area of the equivalent cylinder of the rotating ring; $A_{cs}$ is axial cross-sectional area of the equivalent cylinder of the stationary ring; $L_r$ is length of the equivalent cylinder of the rotating ring; $L_s$ is length of the equivalent cylinder of the stationary ring; $h_r$ is surface heat transfer coefficients between outside of the rotating ring and the fluid in the annular seal space; $h_s$ is surface heat transfer coefficients between outside of the stationary ring and the fluid in the annular seal space; $C_r$ is outer circumference of the equivalent cylinder of the rotating ring; $C_s$ is outer circumference of the equivalent cylinder of the stationary ring; $t_m$ is average temperature of the seal face, and $t_f$ is average temperature of the fluid in annular seal space.

where $h_r$ and $h_s$ can be calculated by Eqs. (5) and (6), respectively[1,9]

$$h_r = 0.135 \frac{\lambda r}{D_r} [0.5 Re_c^2 + Re_c] Pr^{0.33}$$  

(5)

$$h_s = 0.135 \frac{\lambda s}{D_s} \epsilon_c Re_c^{0.66} Pr^{0.33}$$  

(6)

where $\lambda f$ is heat conductivity of the sealed fluid; $D_r$ is outer diameter of the rotating ring; $Re_c$ is reynolds number related to the rotary stirring effect of the sealed fluid, $Re_c = \frac{\rho D_r^2}{\mu} \omega$; $\rho$ is density of the
sealed fluid; $\omega$ is angular velocity of the rotating ring; $\mu_f$ is dynamic viscosity of the sealed fluid; $Re_a$ is Reynolds number related to the cross-flow effect of the sealed fluid, $Re_a = \frac{\nu_f D_1}{\mu}$; $\nu_f$ is axial average flow rate at the outside of the rotating ring; $Pr$ is Prandtl number; $D_s$ is outer diameter of the stationary ring; $\varepsilon_l$ is correction coefficient of the rotary stirring effect, $\varepsilon_l$ is usually 0.2; $Re_{as}$ is Reynolds number of the cross-flow effect of the sealed fluid near the stationary ring, $Re_{as} = \frac{\nu_h D_1}{\mu}$; $\nu_h$ is axial average flow rate at the outside of the stationary ring.

2.2 Calculation model of the average temperature of the end faces for mechanical seals

The average temperature of the end face for mechanical seals can be obtained by Eqs. (1)∼(4).

$$t_m = \frac{f \rho \nu_f A_s}{[m \lambda_1 A_s \text{tanh}(m L_s) + m \lambda_2 A_s \text{tanh}(m L_s)]} + t_f$$   \hspace{1cm} (7)

Usually, a mechanical seal friction pair is comprised of a rotating ring and a stationary ring, and they are made of hard and soft materials, respectively. Wei, et al simplified the contact between hard ring and soft ring to the contact between an ideal smooth surface and a rough surface, the friction factor fractal model of the end face for mechanical seals was established, and the expression is \[ f = \frac{\pi^{(4-D)/2} [\lambda^2 - \lambda]^{3/2} \mu_f n \phi_c}{45(\pi - 2)^{2(1-D)/5} (\lambda^2 - \lambda) [p_s + Kp]} \times \left( \frac{D}{2 - D} \right)^{2(1-D)/4} \psi^{2-D} p_m^{1/4} \left( 1 - \frac{K_m p}{p_s + Kp} \right) \] where $D$ 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; $\mu_m$ is dynamic viscosity of liquid film; $n$ is rotating speed; $\phi_c$ is contact factor; $b_m$ is asperity bearing area ratio of seal face, and it can be seen in Ref.[4]; $f_c$ is contact friction factor of asperities of the seal face, and $K_m$ is film pressure factor.

The dynamic viscosity of liquid film $\mu_m$ can be determined by the average temperature of the seal face $t_m$. The relationship between dynamic viscosity of water and temperature can be expressed by the following formula

$$\mu_m = 0.001 \exp[-0.0175(t_m - 20)]$$   \hspace{1cm} (9)

Contact factor $\phi_c$ can be expressed as \[ \phi_c = \begin{cases} \exp(-0.6912 + 0.782\lambda - 0.304\lambda^2 + 0.0401\lambda^3) & \lambda \leq 3 \\ 1 & \lambda > 3 \end{cases} \] \hspace{1cm} (10)

where $\lambda$ 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\[ \lambda = \left( \frac{\pi - 2}{\pi^{(4-D)/2}} \left( \frac{2 - D}{D} \right)^{2(1-D)/4} \left( 2 - D \right) \ln 1.5 \right)^{2/3} \times \left( \frac{D}{2 - D} \right)^{2(1-D)/4} [\lambda^2 - \lambda]^{3/2} \] \hspace{1cm} (11)

where $l^r$ is the sampling length.
3. Simulation calculation of average temperature of the seal face

It can be seen from Eqs. (7), (8) and (9) 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 average temperature accurately, and the calculation process is shown in Figure 1.

3.1 Effects of operating parameters on average temperature of the seal face

3.1.1 Effect of spring pressure and sealed fluid pressure on average temperature

The Relationship between average temperature $t_m$ and spring pressure $p_s$ with different sealed fluid pressure $p$ obtained under the conditions of rotating speed $n=2900$ rpm, fractal dimension $D=1.56$ and characteristic length scale $G=9.8\times10^{-7}$m is shown in Figure 2. As can be seen from Figure 2 that the average temperature of seal face increase linearly with the increase of spring pressure or sealed fluid pressure.
Table 1. Main parameters of seal rings

| Material        | Rotating ring | Stationary ring | Carbide YG8 Carbon graphite M106K |
|-----------------|---------------|----------------|----------------------------------|
| Structural parameter |               |                |                                  |
| inner radius of seal face, \(r_1\)/mm |               |                | 34.5                             |
| outer radius of seal face, \(r_2\)/mm |               |                | 39                               |
| Load factor, \(K\) |               |                | 0.895                            |
| Performance parameter |               |                |                                  |
| Elastic ratio of rotating ring, \(E_1\)/MPa |               |                | \(6\times10^5\)                  |
| Elastic ratio of Stationary ring, \(E_2\)/MPa |               |                | \(1.6\times10^4\)                |
| Poisson's ratio of rotating ring, \(\nu_1\) |               |                | 0.24                             |
| Poisson's ratio of Stationary ring, \(\nu_2\) |               |                | 0.20                             |
| Compressive yield strength of Stationary ring, \(\sigma_{sy}\)/MPa |               |                | 200                              |

Table 2. Relevant parameters of equivalent cylinder of seal rings

| Equivalent cylinder of rotating ring | Heat conductivity, \(\lambda_r\)/W\(\cdot\)m\(^{-1}\)\(\cdot\)°C\(^{-1}\) | Axial cross-sectional area, \(A_{cr}\)/m\(^2\) | Length, \(L_r\)/m | Outer diameter, \(D_r\)/m | Outer circumference, \(C_r\)/m |
|-------------------------------------|-------------------------------------------------|---------------------------------|-----------------|-----------------|-----------------|
|                                     | 35.9                                             | \(3.2\times10^{-3}\)           | 0.019           | 0.094           | 0.2953          |

| Equivalent cylinder of stationary ring | Heat conductivity, \(\lambda_s\)/W\(\cdot\)m\(^{-1}\)\(\cdot\)°C\(^{-1}\) | Axial cross-sectional area, \(A_{cs}\)/m\(^2\) | Length, \(L_s\)/m | Outer diameter, \(D_s\)/m | Outer circumference, \(C_s\)/m |
|---------------------------------------|-------------------------------------------------|---------------------------------|-----------------|-----------------|-----------------|
|                                       | 15                                               | \(1.039\times10^{-6}\)         | 0.0035          | 0.078           | 0.245           |

Figure 2. Relationship between \(t_m\) and \(p_s\).

3.1.2 Effect of rotating speed on average temperature

The Relationship between average temperature \(t_m\) and rotating speed \(n\) with different spring pressure \(p_s\) obtained under the conditions of sealed fluid pressure \(p=1.0\)MPa, fractal dimension \(D=1.56\) and characteristic length scale \(G=9.8\times10^{-9}\)m is shown in Figure 3. As can be seen from Figure 3 that the average temperature of seal face increase approximately linearly with the increase of rotating speed.
3.2 Effect of surface topography fractal parameters on average temperature of the seal face

The Relationship between average temperature $t_m$ and fractal dimension $D$ with different characteristic length scale $G$ obtained under the conditions of spring pressure $p_s=0.2\text{MPa}$, sealed fluid pressure $p=1.0\text{MPa}$ and rotating speed $n=2900\text{rpm}$ is shown in Figure 4. As can be seen from Figure 4 that the average temperature of seal face increase with the increase of fractal dimension and with the decrease of characteristic length scale of the soft ring. The change of average temperature was smaller when the fractal dimension was smaller and the characteristic length scale was larger, while the average temperature increase rapidly when the fractal dimension was larger and the characteristic length scale was smaller.

4. Conclusions

(1) A Calculation model of the average temperature of the end face for mechanical seals was proposed, which expresses the relationship among the average temperature and operating parameters, structural parameters, surface topography fractal parameters, material characteristic parameters and friction factor between the seal faces. The model provides a foundation for studying the friction, wear and leakage rate between the end faces for mechanical seals, and has important references for predicting the working performances between the seal faces in actually running and designing of the end face for mechanical seals.
(2) Operating parameters have a great effect on average temperature of the end faces for mechanical seal. The average temperature of seal face increase linearly with the increase of spring pressure or sealed fluid pressure, and increase approximately linearly with the increase of rotating speed.

(3) The surface topography of mechanical seal has also been an important factor affecting the average temperature of seal face. The average temperature of seal face increase with the increase of fractal dimension and with the decrease of characteristic length scale of the soft ring. The change of average temperature was smaller when the fractal dimension was smaller and the characteristic length scale was larger, while the average temperature increase rapidly when the fractal dimension was larger and the characteristic length scale was smaller.

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References
[1] X.D.Peng, Y.B.Xie, Y.Q.Gu, Chemical Engineering & Machnery, 23, 333 (1996)
[2] Z.G.Luan, M.M.Khonsari, Journal of Engineering Tribology, 221, 717 (2007)
[3] L.Wei, B.Q.Gu, P.G.Zhang, CIESC Journal, 63, 3202 (2012)
[4] L.Wei, B.Q.Gu, Q.H.Liu, CIESC Journal. 64, 1723 (2013)
[5] L.Wei, P.G.Zhang, G.F.Fang, Applied Mechanics and Materials, 687-691, 142 (2014)
[6] J.D.Summers-Smith, Mechanical seal practice for improved performance (1992)
[7] Y.Q.Gu, Practical technology of mechanical seals (2001)
[8] B.Q.Gu, J.F.Zhou, Y.Chen, Science in China Series E: Technological Sciences, 51, 611 (2008)
[9] G.P.Yan, Z.L.Liu, Chinese Journal of Computational Mechanics, 29, 551 (2012)
[10] C.W.Wu, L.Q.Zheng, ASME Journal of Tribology, 111, 188 (1989)