Effect of taper parameters on mechanical seal performance of taper/pores composite mechanical seal

X P Cheng, Q Liu, Y L Zhang, J Wei
Institute of Applied Physics, Jiangxi Academy of Sciences, 330029, Nanchang, China
E-mail:654268480@qq.com

Abstract. A new combination fluid seal with the cone/pores is given, and the 3D fluid/solid coupling theory model is constructed, considering the interaction between the liquid pressure field and the stress deformation of the seal rings, led to seal clearance unceasingly change. Through the corresponding numerical calculation, film pressure distribution and end face deformation situation are obtained between the seal end faces, the influence rule of taper on sealing performance under the high and low pressure conditions is analyzed. The results show that the dynamic pressure effect caused by rhomboid pores can produce circumferential and radial wavy deformation of the seal end faces, while the static pressure effect also changes with the taper change of the end faces. For the low pressure and medium speed equipment, the conical end face which the taper is 0-2.4 should be chosen priority. For higher pressure and rotational speed device, the convergence end face seal which the taper is 0-2.4 should first be chosen.

1. Introduction
In 1980 Etsion studied the performance parameters of taper end face seal in detail and first presented the feasibility of taper end face seal theoretically\(^\text{[1,2]}\). Subsequently, scholars from various countries have carried out researches on the relatively axial, radial, angular syndication and end faces deformation of mechanical seals during operating\(^\text{[3,4]}\). By controlling the end faces clearance, the design of the seal structure and working conditions, the relative deflection of the end faces is reduced and the end faces deformation is neutralized to prevent the contact between the two end faces\(^\text{[5,6]}\). Metcalfe\(^\text{[7]}\) adopts the influence coefficient method and considers the constraints and forces between the components of the sealing system to attribute the seal end faces deformation to the comprehensive influence of various parameters, and calculates the deformation value of each point of the sealing rings. Mayer\(^\text{[8]}\) believed that force and temperature were the factors affecting the deformation and calculated the deformation of the mechanical seal end faces by using the ring theory. Salani\(^\text{[9]}\) analyzed the seal end faces deformation of different structures and obtained experimental verification. And it is found that the experimental values are in good agreement with the calculated ones. Metcalfe\(^\text{[10]}\) classified the deformation of the seal end faces into the comprehensive influence of various parameters by considering the constraints between components and the stress of the seal structure, and calculated the deformation of end faces. Blasisk\(^\text{[11]}\) used displacement sensor and piezoelectric brake device to measure the end faces deformation of non-contact fluid mechanical seal, and learned that angular velocity and ring material have similar effects on the end faces temperature rise. Peng Xudong\(^\text{[12]}\) systematically explored the influence of seal rings deformation on performance under various structures and constraints. Meng Xiangkai\(^\text{[13]}\) calculated the end faces deformation
behavior of mechanical seals under different pressures by adopting the integral contact finite element method and considering the influence of auxiliary sealing rings.

2. Model

2.1. Geometric model

Fig.1 shows the schematic model of the local end faces geometric structure of cone-diamond pores composite mechanical seal. There is a certain taper on the static torus, among them: \( \varphi \)-cone angle; \( r_m \)-starting radius of the cone; \( r_i \), \( r_o \)-inner and outer diameter of end face Seal ring; \( h_0 \)-basis of film thickness; the taper is defined as \( \phi=100 \times (r_o-r_m) \times \tan \varphi / h_0 \) [14]. The rhombic pores in the rotary ring are distributed symmetrically in the center along the circumferential direction, then the rhombic pores are equally spaced in the radial direction, And the diamond pores is equal depth, their depth is \( h_{1} \). Diamond cut faces are directional pores, two structure parameters are used to indicate its geometric characteristics, and is defined the direction factor \( \xi \) for:

\[
\xi = b/a
\]  

Among them: \( a \)-semi-major axis of diamond pores symmetry axis, \( b \)-semi-minor axis.

![Fig.1 Schematic diagram of diamond pores distribution and conical surface structure](image)

2.2. Mathematical model

The fluid pressure between the end faces is constant along the direction of film thickness by assuming. The sealing fluid is Newtonian fluid whose viscosity remains constant, then the governing equation used to describe the liquid film pressure on the end faces can be expressed by two-dimensional Reynolds Equation:

\[
\frac{\partial}{\partial x} \left( \frac{h^4}{6 \mu} \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial y} \left( \frac{h^4}{6 \mu} \frac{\partial p}{\partial y} \right) = U \frac{\partial h}{\partial x} + V \frac{\partial h}{\partial y}
\]  

Among them: \( U, V \)-average linear velocity of end face \( x \) and \( y \), \( h \)-the thickness of liquid film, \( p \)-liquid film pressure, \( \mu \)-viscosity coefficient of sealing medium. The end surface liquid film thickness equation for:

\[
h = \begin{cases} 
  h_n - h_i + h_j & \text{No cone area outside the pores} \\
  h_n - h_i + h_j + h_e & \text{Cone area outside the pores} \\
  h_n + h_i - h_j + h_t & \text{No cone area inside the pores} \\
  h_n + h_i - h_j + h_t + h_c & \text{Cone area inside the pores}
\end{cases}
\]  

Among them: \( h_n \)-the amount of deformation at a point on the rotary ring; \( h_i \)-the amount of deformation at a point on the stationary ring; \( h_t \)-the cone height at a point on the cone; When the mechanical seal operates stably, the axial force of the rotary ring assembly is balanced, then:

Closing force \( F_c \equiv \text{opening force } F_o \)  

Among them:

\[
F_c = \pi (r_o^2 - r_i^2) \left[ p_{\varphi} + B(p_o - p_l) \right]
\]
In the formula: $p_{eq}$—spring pressure, $B$—seal end face balance ratio, $A_s$—the cross-sectional area of a single diamond pore, $p_i$—inner and outer diameter pressure of sealing ring, $p_o$—environmental pressure or cavitation pressure, $N$—period number of rhomboid pores.

Formula (2) is discretized by the finite difference method, then combined with formula (3)-(6) is solved to establish a three-dimensional mathematical analysis model of fluid-solid coupling for hydrodynamic mechanical seals. The deformation of the seal rings is calculated by commercial finite element software programming, and the $p$ distribution is solved again with film thickness deformed, and the iterative solution is carried out repeatedly until the convergence criterion meets the program setting, then the iteration stops. At fast, seal performance parameters such as liquid film stiffness, opening force and leakage rate are calculated from the final iterated pressure. See the references for detailed algorithms. 

3. Calculation results and discussion

The influence of the taper of the seal end face on seal performance of the cone-diamond pores combination seal under the condition of high and low pressure was studied. Structural parameters: $r_1=127.25\text{mm}$, $r_2=141.75\text{mm}$, $a=2\text{mm}$, $h_1=1.25\mu\text{m}$, Number of diamond pores in single row $n=4$, $\xi=0.67$. Elastic modulus of rotary and static rings: $E_1=617\text{GPa}$, $E_2=23\text{GPa}$, Poisson's ratio of rotary and static rings: $\nu_1=0.28$, $\nu_2=0.25$, the thickness of rotary and static sealing rings: $h_{b1}=10\text{mm}$, $h_{b2}=15\text{mm}$, $B=0.85$, $p_{eq}=0.1$, $N=150$. Operating parameters: $p_s=p_i=0.1\text{MPa}$, low pressure $p_e=0.3\text{MPa}$, high pressure $p_h=6.6\text{MPa}$, $\mu=0.001\text{Pa.s}$.

When calculating and studying the influence of certain parameters on seal performance parameters, the other parameters remain unchanged except for explanation. On this basis, the change rule of sealing performance under high and low pressure conditions and different taper conditions would be analyzed.

3.1. Influence of taper on sealing performance at low pressure ($=0.3\text{MPa}$)

Under the working condition of low pressure, $\omega=800\text{r/min}$, and different tapers, figure 2 shows that changes of film thickness caused by deformation of end faces during stable operation. The results show that the liquid flow in the pores area increases due to the presence of rhomboid pores on the end face, and the strong dynamic pressure effect is produced due to the block and extrusion of the pores boundary, so the pressure field of the liquid film between the end faces has regular and alternating trough and peak area. In the circumferential direction, the high pressure region appears on the side blocked by the boundary of the rhomboid pores; on the contrary, the trough appears in the divergent region of the liquid film in the rhomboid pores. In the radial direction, the highest pressure peak occurs near the outside diameter when the $\phi$ is 0, then the pressure peak decreases in turn to the inside diameter, while the highest pressure peak occurs near the inside diameter side when the $\phi$ is not zero, the pressure peak decreases in turn to the outer diameter side. At the same time, the peak pressure varies with the tapers different, and the highest pressure peak has the following rule: $p_{b(\phi=0)}>p_{b(\phi=2.4)}>p_{b(\phi=4)}$, so the change of the maximum pressure difference ($\Delta p(\phi=0)$) between the end faces is the largest when $\phi$ is zero, while $\Delta p(\phi=2.4)$ and $\Delta p(\phi=4)$ is progressively smaller. The range of high and low pressure areas is also different when the $\phi$ is different, the low pressure area of the liquid film between the end faces is very large, while the high pressure area is very small when $\phi$ is 4. Therefore, the flow pressure field can easily cause the unbalanced operation of the seal end faces. By contrast, the range of high and low pressure of the liquid film pressure field is relatively balanced when $\phi$ is zero, so the two sealing end faces in this state will be very stable in the process of operation, not easy to contact, it is difficult to wear, And mechanical seal fluid film stability in the three when $\phi$ is 2.4.

It can also be seen that the hydrostatic pressure action gradually enhanced with the change of taper along the radial direction from the outside diameter to the inside diameter, the dynamic pressure effect is corresponding change, resulting in the sealing ring surface compression deformation to form uneven surface, the film thickness changes accordingly, no longer is the same thickness, with a certain
waveform and taper. With the increase of taper, the film thickness on the inside diameter side gradually decreases, while that on the outside diameter side gradually increases. In the radial direction, the film thickness increases from inside diameter side to outside diameter side, and the order of maximum and small film thickness is as follows: $h_{\text{max}}(\phi=4)>h_{\text{max}}(\phi=2.4)>h_{\text{max}}(\phi=0)$, $h_{\text{min}}(\phi=0)>h_{\text{min}}(\phi=2.4)>h_{\text{min}}(\phi=4)$. The reasons are as follows: with the increase of the surface taper, the scope which is influenced by the static effect of liquid film between end faces began to grow and increase, and the range which is influenced by the dynamic pressure effect gradually become smaller and less. To maintain the balance of axial force, mechanical seal will automatically adjust the overall film thickness between end faces to control the membrane pressure in the operation of the sealing device, so if the film thickness increases, membrane pressure will decrease accordingly.

![Fig. 2](image_url) Film pressure and film thickness under different taper conditions

Fig. 3 shows the influence rule of taper on sealing performance when $p_o$ is 0.3MPa, and $\omega$ is 800r/min. The left vertical axis corresponds to the gray, black and yellow bar chart, and the right vertical axis corresponds to the colored bar chart. The results show that: the opening forces are not very different when $\phi$ is 0, 2.4, 4 and 6.4, while the liquid film stiffness varies greatly, sort for: $K_z(\phi=0)>K_z(\phi=2.4)>K_z(\phi=4)$, $K_z(\phi=6.4)$, this indicates that the liquid film stability between the sealing end faces is greatly affected by the uniformity and strength of the liquid film pressure fluctuation. The more balanced the liquid film pressure is, the greater the strength is, the better the corresponding liquid film stability is, and the greater the liquid film stiffness is. The more stable the sealing surface is in the process of operation, the more difficult it is to contact, and the longer the life cycle is. It can also be seen that the law of minimum film thickness and leakage rate in the color histogram on the right: $h_{\text{min}}(\phi=0)>h_{\text{min}}(\phi=6.4)>h_{\text{min}}(\phi=4)>h_{\text{min}}(\phi=2.4)$ and $Q(\phi=6.4)>Q(\phi=0)>Q(\phi=4)>Q(\phi=2.4)$, that is, the minimum film
thickness of the seal face basically determines the change of its leakage rate, the smaller the minimum film thickness is, the better the sealing is, but the smaller the minimum film thickness is, the sealing face is more easy to contact wear, the life will be shortened.

To sum up, considering the stability of liquid film and sealing reliability, the low pressure and medium speed equipment, convergence taper face which the taper is 2.4 should be chosen, the leakage rate is lesser, the opening force and liquid film stiffness are better.

Fig. 3 Comparison of sealing performance parameters with different tapers

3.2. Influence of taper on sealing performance at high pressure (=6.6 MPa)

Fig. 4 The film thickness and film pressure diagram under the different tapers

It can be seen that the changing rule of the pressure field and the liquid film thickness between end faces under the working condition of high pressure, high speed (ω=4000r/min) and different tapers in
Fig. 4. It can be seen from the figure that the static pressure effects of the liquid film on the end faces play an extremely important role under the conditions of high speed and high pressure, and the peak pressure varies greatly with the difference of taper. And the sorting size of the highest pressure peak is $p_{\text{high}}(\phi=0)>p_{\text{high}}(\phi=2.4)>p_{\text{high}}(\phi=4)$, the variation range of pressure difference ($\Delta P(\phi=0)$) between the end faces is the largest, then $\Delta P(\phi=2.4)$ and $\Delta P(\phi=4)$ go down. The local scope of ultrahigh pressure liquid membrane area is far less than the range of low pressure area when the taper is zero, and the pressure difference is very big, the difference between high and low pressure area is smaller when the taper is 2.4 and 4, so the stability of liquid film is better. In addition, with the increase of taper, the influence range of internal and external static pressure difference on the whole seal end face is larger, the action range of static pressure effect is much larger than that of dynamic pressure effect.

It can also be seen that the film thickness gradually decreases from the outside diameter to the inside diameter, forming a linear radial convergence cone, and the size of the taper increases with the increase of the taper. From the perspective of the overall film thickness, the larger the taper is, the larger the film thickness is, while the maximum and minimum film thickness also increase with the increase of the taper. The reason is as follows: with the increase of the taper, the influence range of the static pressure effect on the liquid film between the seal end faces starts to increase and strengthen, while the influence range of the dynamic pressure effect gradually decreases and weakens. In order to maintain the balance of axial forces under the action of fixed closing force, the sealing device will automatically adjust the overall film thickness between end faces to control the film pressure of end faces during operation.

Fig. 5 Influence of taper on sealing performance

The influence law of taper on sealing performance parameters under high pressure and high speed ($\omega=4000$ r/min) conditions in Fig. 5. When the taper is zero, 2.4, 4 and 6.4, the opening forces in the gray and yellow bar on the left are not very different by the figure, but the liquid film stiffness sequence is $K_{\text{st}}(\phi=2.4)>K_{\text{st}}(\phi=4)>K_{\text{st}}(\phi=6.4)>K_{\text{st}}(\phi=0)$. The order of the leakage rate and minimum film thickness in the color histogram on the right are $h_{\text{min}}(\phi=6.4)>h_{\text{min}}(\phi=4)>h_{\text{min}}(\phi=2.4)>h_{\text{min}}(\phi=0)$ and $Q(\phi=6.4)>Q(\phi=4)>Q(\phi=2.4)>Q(\phi=0)$. Thus, mechanical seal which the taper is 2.4 has the best stability and good sealing, Mechanical seal of which the tapers are 0 and 2.4. The liquid film stability and sealing are common, the effect of $\phi=6.4$ is the worst, so it can be selected that the appropriate end face seal according to the different requirements of the equipment.

4. Conclusion

1) Dynamic pressure effect caused by the diamond pores can make faces circumferential and radial wave distortion under the conditions given, and with the change of taper, the static pressure effect also changes accordingly in the area of end faces. The membrane stability is greatly affected by the uniformity and strength of the membrane pressure fluctuation. The more balanced the pressure changes, the greater the strength is, the better the stability of the corresponding liquid film is, and the greater the liquid film stiffness is. The operation is the smoother, the harder the contact is and the longer the life cycle is.

2) For low pressure and medium speed equipment, in order to improve the stability of liquid film and seal reliability, the convergence taper ($\phi=0-2.4$) seal face should be chosen. For devices with higher pressure and speed, the convergence taper seal face should be chosen as 2.4-4.
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