Main Pump Seal's Characteristics Affected by Cone Angle and Clearance

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Abstract

Aimed at the face film characteristics of Main Pump Seal (MPS), related theoretical calculation formulas were deduced based on the Reynolds equation. A case with a certain NPPS is carried out, it is take the leakage as the basic constraint conditions, the calculation focusing on leakage, stiffness and opening force, the end cone angle and clearance change been the main changed parameters. The results show that: (1) Leakage is direct ratio with clearance of three powers;(2) Overall amount of leakage increases with the end cone angle synchronously, and in the smaller cone angle changes significantly while larger slightly; (3) film stiffness increased with clearance (cone angle) first and then decreased, and the maximum value is obtained in a certain clearance (cone angle). Finally, the preferred clearance is preferably in the range of 6~7μm, and the cone angle preferably value is 0.4~2.5'. It is provide a reliable theoretical support for MPS design and key parameter optimization, and conducive to any research related to experiment and application.

Keywords: seal, mechanical seal, main pump seal, leakage, film stiffness

1. Introduction

Nuclear reactor coolant pump (main pump) is an important equipment of nuclear power plants, has been hailed as the heart of the reactor coolant system, mainly composed by the motor, pump, seal (mechanical seal) and other components, wherein the seal is a crucial but vulnerable parts, the main shaft seal is mainly used for leakage control of reactor coolant along the axis, its quality will directly affect the normal operation of the pump. Based on the survey of the main pump's fault, about 70% from the shaft seal, especially the first-class sealing [1].

There's some researchers study on MPS for higher stability and life. Müller based on the laminar flow theory, considering the effect of inertia, studied the performance of external pressured seal [1]. Koga and Fujita pointed out the impact of energy loss at inlet and outlet boundary of seals [2]. Tournerie take tracking research on the flow characteristics of hydrostatic seal face, the change of surface flow from laminar to turbulent is theoretically analyzed, through an influence coefficient matrix to consider the effect of thermal deformation, it is pointing out that the film flow in laminar or approximate turbulent state according to the different temperature of inlet [3-9].

Salant studied the mathematical model of the non-contacting hydrostatic seal by solving the face fluid film Reynolds equation to obtain the seal face balance clearance value, pressure and medium viscosity influence on the balance clearance value, theoretic design methods of seal structure is analyzed, and the method is applied to water medium seal design [10, 11]. Lee calculated the thermal stress of the SiC/SiC composite under different temperature and pressure, and the extreme size and position of the thermal stress is analyzed emphatically [12]. Kim studied the performance of the seal ring(Si_3N_4) with the medium of 300℃ water, pointed out the adjustment of sintering crystallization phase can effectively improve the corrosion resistance [13]. Zhang Xinmin solved the simplified Reynolds equation, calculated dynamic characteristics of partial taper hydrostatic seal and given the calculation formula of force coefficient, compared the dynamic characteristics of full taper, partial taper and non taper, pointed out that partial taper structure is more reasonable structure type [14]. Hong Zhenmin analysis the relationship between the 1st stage seal leakage and the temperature of injected water, in the shaft seal

water temperature range allowed, by reducing 1st stage seal injected water temperature to reduce the leakage. And in practice, by using temperature control method to avoid the nuclear main pump high-high stops due on the 1st stage seal leakage [15]. Dai Veiqi uses numerical method to calculate hydrostatic seal flow field and analyze the characteristics of face pressure [16, 17]. Mu Dongbo developed a theoretical analysis model for the stationary ring in second stage seal of main pump seals in nuclear station by ANSYS, obtained the deformation condition of stationary ring by calculation. The model of liquid film between the seal gap was built by using the software Fluent, and the pressure distribution, speed distribution, lifting force and leakage were obtained. The process of the face distortion of seal ring and the process of the transformation of mechanical seal from contact style to non-contact style were simulated by computers [18].

In summary, to improve the stability and service life of the main pump seal, requires two seal faces in any situation of relative rotation, can realize the non-contact operation with a small space state. Currently, there are two practical ways to implement the non-contact operation: (1) Hydrostatic pressure method(hydrostatic seal), the first stage seal withstands most pressure drop, the opening force and film stiffness mainly formed by hydrostatic pressure, in addition the opening force and film stiffness is not sensitive to the rotating speed of main pump's shaft, so it can effectively avoid seal faces direct contact when the main pump startup and stop, but film stiffness of hydrostatic seal is relatively small, which weakens the capability of anti-interference in large extent, so that its stability decreased, and the possibility of the accident failure increased. (2) Dynamic pressure method (dynamic pressure seal), generally three stages seal bear the same pressure drop, face mainly by fluid-dynamic pressure to form opening force and the film stiffness, the opening force and film stiffness of this seal structure is relatively large, especially in the smaller gap condition, reducing the possibility of end face direct contact, so it has strong ability of resistance to interference and high stability.

In this paper, combined with the seal practical situations of domestic nuclear power reactor main shaft, select hydrostatic seal as study object. Take seal face pressure, opening force, leakage as the main test indicators, computational analysis of end cone angle and clearance effect on each index are carried out, it is provided the theoretical basis for higher stability hydrostatic seal design.

2. Research Method
2.1. Physical Models and Parameters

According the first stage seal of the main pump seal specimen made by Sichuan Nikki Seal Co., Ltd, the model is shown in Figure 1, Of which: r_o is the outer radius of the seal faces, r_i is the inner radius of the seal face, r_b is the stationary ring side cone portion and the portion of the boundary between the radius of the torus, called turning radius, β is the cone angle of stationary ring, h is the face clearance.
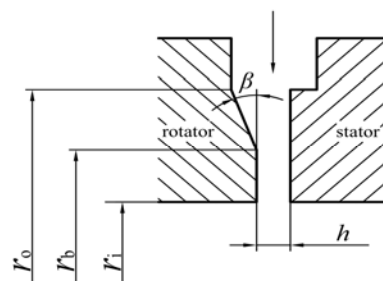

Figure 1. Schematic diagram of sealing surface model

The primary objectives of the research are:
(1) Seal face pressure distribution;
(2) Stiffness and leakage changes with end face cone angle and clearance.

2.2. Control Equation and Simplify

The sealing medium of the main pump seal is water, it can be regarded as an incompressible fluid, and the basic equation is fluid flow Reynolds equation. According to the characteristics of the physical model, can make the following assumptions or approximate:

(1) Medium flow between the seal faces is a continuous laminar flow;
(2) The seal rings are rigid body;
(3) Ignoring the effect of fluid inertia force;
(4) Medium (water) is incompressible;
(5) Ignoring the variety of fluid velocity and gradient in the film thickness direction;
(6) Medium is Newtonian fluid;
(7) The flow process is in the steady state;
(8) Flow is isothermal flow.

Based on the above assumptions, the Reynolds equation can be simplified as follow:

$$\frac{\partial}{\partial r}\left(\frac{rh^3}{12\mu}\frac{\partial p}{\partial r}\right) + \frac{1}{r}\frac{\partial}{\partial \theta}\left(\frac{h^3}{12\mu}\frac{\partial p}{\partial \theta}\right) = \frac{rU}{2}\frac{\partial h}{\partial \theta} \tag{1}$$

Formula (1) is the basic equations of the hydrostatic main pump seal.
In formula (1):
p --medium pressure (Pa);
r --the radius of seal surface at any position (m);
θ --seal face circumferential angle (rad);
h --seal face balance clearance (m);
μ --dynamic viscosity (Pa.s).

2.3. Control Equation Solution

(1) Parallel Face State

When the sealing surface is in the parallel face state, the clearance h is a constant value, and then the radius r is the only integral variable in formula (1). Assume that the seal surface of inner, outer radius respectively are: r_i, r_o, the inner and outer pressure of the medium are: p_i, p_o, integrate of formula (1), get the following conclusion:

(1) Pressure distribution:

$$p_1(r) = \frac{(p_o - p_i)\ln(r)}{\ln(r_o/r_i)} + c_1 \tag{2}$$

(2) Opening force:

$$F_{o1} = \iint p_1(r)r\,dr\,d\theta \tag{3}$$

(3) Leakage:

$$Q_1 = \frac{\pi h^3(p_o - p_i)}{6\mu\ln(r_o/r_i)} \tag{4}$$

(2) Taped Face State

When one of the seal face is conical surface, the clearance h is a variable, then the integral of formula (1) will with the variables of radius r and clearance h at the same time. Now the clearance of h is:

$$h = h_i + (r - r_i)\tan(\beta) \tag{5}$$

Suppose: $k = h_i - (r_i)\tan(\beta)$

The formula (5) substituted in equation (1) and the integral of it:

(1) Pressure distribution:

$$P_2(r) = a\left[\frac{1}{k^3}\ln\left(\frac{r}{k + r\tan(\beta)}\right) + \frac{1}{k^2(k + r\tan(\beta))} + \frac{1}{2k(k + r\tan(\beta))^2}\right] + c_2 \tag{6}$$

In the above formulas: a, c_1, c_2 respectively are the integration constants associated with p_i, p_o, r_i, r_o, β, k.

(2) Opening force:

$$F_{o2} = \iint p_2(r) r\, dr\, d\theta \tag{7}$$

(3) Leakage:

$$Q_2 = \frac{\pi}{6\mu}a \tag{8}$$

(3) Actual Model Solutions

Actual model area size parameters: inner radius, turning radius, outer radius are respectively take as: r_i, r_d, r_o, the corresponding pressure respectively are: p_i, p_d, p_o, among them p_d is an unknown parameter. Inner face clearance and clearance at conical surface starting position is: h_i, the outer face clearance is: h_o, cone angle is β.

(1) Obtained p_d by the formula (4) and (8):

By the formula (8) calculating the leakage of tapered face of Q_2, by the formula (4) to give the leakage of parallel face of Q_1, finally obtained p_d by $Q_1 = Q_2$.

(2) Substituted p_d in equation (2) and (6) to give the pressure distribution of inner parallel face $p_1(r)$, and the pressure distribution of outer tapered face $p_2(r)$.

(3) By the formula (3) and (7) are solved and add, obtained a total opening force:

$$F_o = F_{o1} + F_{o2} \tag{9}$$

(4) Obtained film stiffness K by F_o the h derivation:

$$K = -\frac{\partial F_o}{\partial h} \tag{10}$$

(5) Obtained the leakage Q by the formula (4) or the formula (8):

$$Q = \frac{\pi(p_o - p_d)}{6\mu}c_1 \tag{11}$$

Or:

$$Q = \frac{\pi h^3(p_d - p_i)}{6\mu\ln(r_d/r_i)} \tag{12}$$

(6) By the formula (10), (11), (12) to give stiffness and leakage ratio S_{KQ}:

$$S_{KQ} = \frac{K}{Q}$$

(13)

3. Results and Analysis
3.1. Specimen Calculation Parameters
Taking the first stage seal test pieces of the main pump seal made in *Sichuan Nikki Seal Co., Ltd* as the reference. Basic parameters are as follows:

$$p_i = 0.55(\text{Mpa}), p_o = 15.5(\text{Mpa}), r_o = 152.5(\text{mm}), r_i = 108.5(\text{mm}), r_d = 121.5(\text{mm}), t_0 = 20.2(°),$$
$$\mu = 1.0e - 3(\text{Pa.s}).$$

Focusing on the end cone angle and clearance change, specifically study the effects of cone angle on pressure distribution, opening force, film stiffness and leakage.
Seal operating parameters are as follows:
Medium: water;
Pressure of High-pressure side (outer side): $p_o = 15.5(\text{Mpa})$;
Pressure of Low pressure side (inner side): $p_i = 0.55(\text{Mpa})$;
Temperature:15~55(℃)
Leakage: $Q = 0.68 \sim 1.2(\text{m}^3/\text{h})$
Shaft speed: $n = 1485(\text{r/min})$.

3.2. Results and Analysis
(1) Leakage
Figure 1(a) shows the relation curves of leakage and end face clearance, the taper angle of the end face was taken from 1' to 8'; figure 2(b) with the taper angle of the end face was taken from 10' to 80'; Figure 3 shows the relation curves of leakage and end face taper angle, and the clearance of the end face was taken from 6-8μm.
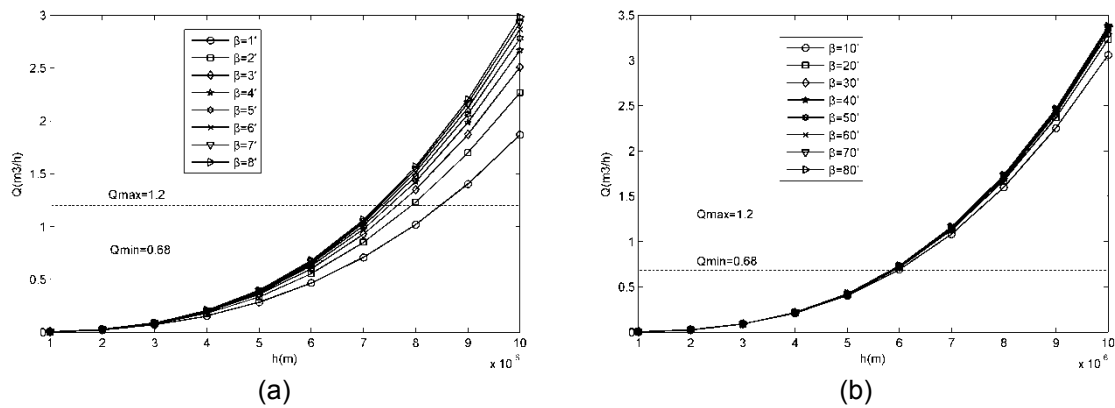

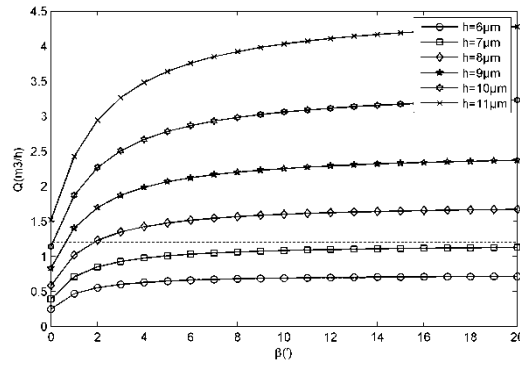Figure 2. Leakage Changes with Clearance

Figure 3. Leakage Changes with End Cone Angle

From Figure 2-3, observing the allowable range of clearance under different cone angles, while the amount of leakage within the allowable range:

a) Overall amount of leakage increases with the end cone angle increased, the variation can be roughly differentiated two areas: the small cone angle range (less than about 2-4'), the amount of leakage reduced rapidly with the end cone angle decreases, and the amount of leakage decrease tended to increase with the cone angle reduced; the large cone angle range (more than about 2-4'), the amount of leakage raised slowly with the end cone angle increases, and the amount of leakage increase tended to decrease with the cone angle raised. As shown in Fig.3, take the end clearance of 6 microns, when the end face cone angle is greater than 4', the amount of leakage only slightly increased.

b) The optimal clearance value is about 6~7μm. Under the condition of design leakage is $Q = 0.68 - 1.2 \, \mathrm{m^3/h}$, the possible clearance value is about 6~10μm (only if the clearance is in this range, it may satisfy the leakage $Q = 0.68 - 1.2 \, \mathrm{m^3/h}$), and in particular when the end clearance of about 6~7μm, the different end cone angle state, the amount of leakage is in a given range within the design value while the cone angle in a large scope of variation, selecting the clearance as the design value, can effectively prevent the cone angle changes or deformation or machining error effect the leakage significantly.

(2) Film Stiffness

Figure 4 shows the relation curves of the end face fluid film stiffness and end clearance, in which the cone angle of the end face were taken from 1' to 8' and 0.1' to 0.8' , Figure 5 shows the film stiffness under the end clearance were 6~10μm.
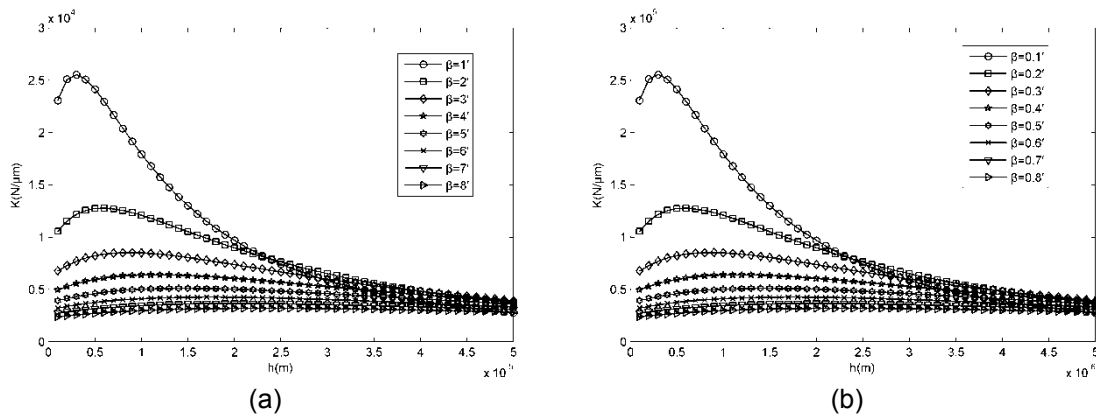

(a)

(b)

Figure 4. Film Stiffness (cone angle from 1 to 8 'and 0.1 to 0.8')
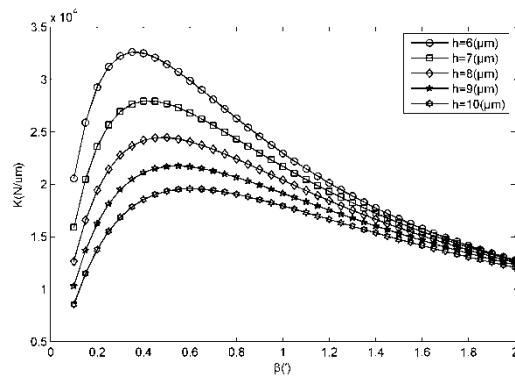
Figure 5. Film Stiffness (clearance from 6~10μm)

From Figure 4-5:

a)   While the end cone angle is given, film stiffness increased with clearance first and then decreased. and the maximum value obtained in a certain clearance value. As shown in Figure 4(a), when $\beta = 2'$, film stiffness carve obtained maximum value in about $h = 6\mu m$. in the range of: $h < 6\mu m$, the film stiffness rapidly raised with the end cone angle increases, in the range of: $h > 6\mu m$, film stiffness present the trend decreases with the end cone angle increases, in which the clearance at the near vicinity of extreme, stiffness decreased greatly, while away from the extreme zone, the stiffness decreased gradually stable, and finally close to zero .

b)   While the end clearance is given, film stiffness increased with cone angle first and then decreased. As shown in Figure 5, when $h = 7\mu m$, film stiffness carves obtained maximum value in about $\beta = 0.4'$. in the range of $\beta < 0.4'$, the film stiffness rapidly raised with the end cone angle increases, in the range of $\beta > 0.4'$, film stiffness present the trend decreases with the end cone angle increases, in which the cone angle at the near vicinity of extreme, stiffness decreased greatly, while away from the extreme zone, the stiffness decreased gradually stable, and finally close to zero.

c)   Stiffness value is extremely sensitive to cone angle changes near its extreme value area. In this area, a very small cone angle change can cause dramatic changes in stiffness, especially in the range less than the extreme cone angle, the influence is more remarkable. Selecting slightly larger than cone angle β at extreme value while design, can reduce the end cone angle changes caused by different factors influence on the stiffness value.

Theoretical perfect cone angle value is about 0.4~2.5'. according to amount of leakage value, the reasonable clearance is about 6~7μm, with this clearance from Figure 5, film stiffness carve obtained maximum value in about $\beta = 0.4'$, further increase the range of the end face cone angle, found face cone angle of greater than about $2.5'$, its stiffness decreased one order of magnitude from 10^4 down to 10^3, the end face cone angle of greater than about 20', its stiffness decrease an order of magnitude from 10^3 down to 10^2.

(3) Opening Force

Figure 6 shows the relation curves of the end face film opening force and end clearance, with the cone angle of the end face were taken from 1' to 8', fig.7 shows the relation curves of the opening force and cone angle, with the clearance h of the end face was taken from 1~8μm.
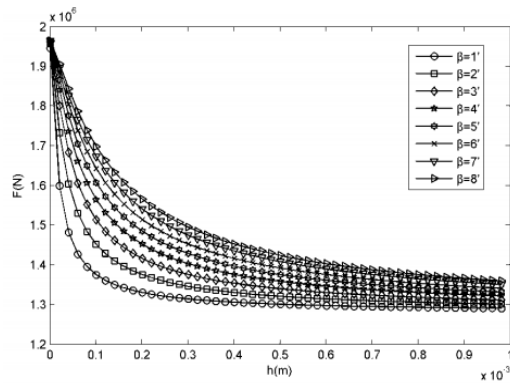
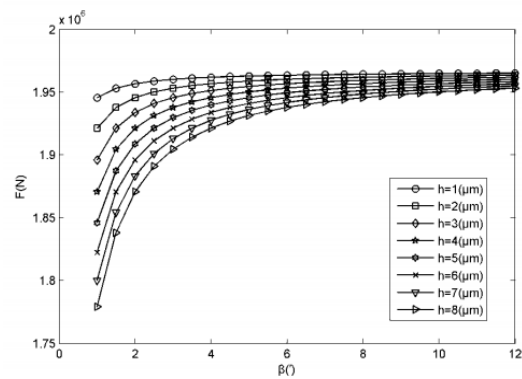Figure 6. Opening Force Changes with Clearance (the cone angle from 1-8')

Figure 7. Opening Force Changes with Cone Angle ($h = 1 - 8\mu m$)

a) While the end cone angle is given, film opening force decreases with the clearance increases, as shown in Figure 7, in the range of $h < 0.2mm$, the film opening force rapidly reduced with the clearance increases, in the range of $h > 0.2mm$, the film opening force reduces slowly with the clearance increases, and eventually approaches a constant value.

b) While the clearance is given, film opening force increases with the end cone angle increases, as shown in Fig.7, in the range of $\beta < 2 \sim 4'$, the film opening force rapidly rises with the end cone angle increases, in the range of $\beta > 2 \sim 4'$, the film opening force rises slowly with the end cone angle increases, and eventually approaches a constant value.

4. Conclusion

Main pump hydrostatic mechanical seal theoretical calculation formulas were deduced based on the Reynolds equation simplified, it is take the leakage as the basic constraint conditions, the calculation focusing on leakage, stiffness and opening force, the end cone angle and clearance change been the main changed parameters. The results show that:

a) The clearance has greater influence on the amount of leakage than the cone angle. When the leakage value is given, it can obtain a certain clearance range make the leakage not out of the limits while the cone angle changes to any value. In the case of this paper, the optimal clearance value is about 6~10µm with the design leakage is $Q = 0.68 - 1.2\,\mathrm{m^3/h}$.

b) Overall amount of leakage increases with the end cone angle up. the variation can be roughly differentiated two areas: the small cone angle range (less than about 2-4'), the amount of leakage rapidly reduced with the end cone angle decreases, and the amount of leakage decrease tended to increase with the cone angle reduced; the large cone angle range (more than about 2-4'), the amount of leakage raised slowly with the end cone angle increases, and the amount of leakage increase tended to decrease with the cone angle raised. as shown in Figure 3, the end clearance take 6 microns, when the end face cone angle is greater than 4', the amount of leakage only increased slightly.

c) Reasonable clearance value of about 6~7µm. Under the amount of leakage in the system design is: $Q = 0.68 - 1.2\,\mathrm{m^3/h}$, the possible value of clearance is: 6~7µm, and in particularly when the clearance is about 6~7µm, under different end cone angle state, the amount of leakage is in a given range within the design. Take the clearance as the design value, can effectively prevent the cone angle changes or deformation or machining error effect the leakage significantly.

d) Theoretical perfect cone angle value is 0.4~2.5'. At this condition, the film stiffness of has a large value and relatively small changes amplitude, on the other hand it is

preferably fitted with face clearance from 6~7μm, so that leakage is expected within the control range.

In this paper, the calculation method is based on the basic theory of fluid, according to the seal structure and the characteristics of the flow field obtained, can describe the end face fluid film properties better. It is provide a reliable theoretical support for MPS design and key parameter optimization, and conducive to any research related to experiment and application.

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