Influence of undersea vehicle shafting offset on contact stress and temperature of spherical stern shaft seal

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Abstract. In order to solve the problem that the sealing surface can not adapt itself due to the change of the stern shaft seal of the stern shaft of undersea vehicle. A spherical stern shaft seal structure was proposed in this paper. In order to lay a foundation for the analysis of the sealing performance of the spherical stern shaft seal, a three-dimensional thermal-structural coupling finite element model of the spherical stern shaft seal and the Stern Shaft was established by using ANSYS software. The distribution of contact stress and temperature on the surface of the spherical stern shaft seal was simulated when the Stern Shaft was moving in a series of 0, 1, 2mm in the vertical direction. The influence of shaft motion on contact stress and temperature on surface of spherical stern shaft seal is analyzed.

1. Introduction
Undersea vehicle stern shaft stern shaft seal working conditions are complex, in the start-stop phase, the stern shaft seal is basically in dry friction state, at low speed or normal sailing stage in the mixed friction state or boundary. In use, due to the stern shaft bending shaft alignment or tail bearing wear and other reasons, it is easy to cause the axis deviation, resulting in the dynamic and static ring seal plane is not the same as the axis, the opening bias wear, and even leakage exceed the standard, affecting the performance of the stern shaft seal.

In this paper, a three-dimensional finite element model of the spherical stern shaft seal was established under the special dry friction condition. Under the condition of large depth (500 m), the influence law of stern shaft deviation on the performance of the spherical stern shaft seal (including the deformation of contact pressure and temperature) was studied, and the sealing performance of the spherical stern shaft seal and the following ability of the static ring were evaluated.

2. Three-dimensional geometric model of spherical stern shaft seals
The three-dimensional geometric model of the stern shaft -spherical stern shaft seal was established, as shown in Figure 1. In order to simulate the rotation of the moving ring and shaft together, the Boolean bonding operation was used to bond them together.
3. Three-dimensional thermal-structural coupled finite element model

3.1 modeling

Based on ANSYS, a three-dimensional coupled finite element model of the thermal structure of the spherical stern shaft seal of the stern shaft is established, as shown in Fig. 2. The model is composed of the stern shaft, the static ring, the moving ring, and the springs. (as shown in Fig. 2(a)).

After the three-dimensional solid model of the shaft and spherical stern shaft seal was established, the material properties of each component were defined, and the model was meshed with 33978 elements and 37650 nodes, as shown in Fig. 2(b).

The surface contact pairs of the friction surfaces of the dynamic and static rings in the finite element model are defined. When the software is set, the static global sealing surface (hereinafter referred to as the sealing surface) is the contact surface, and the moving global sealing surface is defined as the target surface. Set the normal stiffness coefficient as 0.1 and friction coefficient as 0.03. The thermal conductivity between the spheres is set at 100 W/(mꞏ℃). The sealing surface contact pair is shown in Fig. 3.

In the stationary ring seat, 8 spring units are uniformly arranged to simulate the action of springs, as shown in Fig. 4. The spring stiffness is $k = 1.4 \times 10^4$ N/m.
The following assumptions are made in the calculation of finite element analysis:

1) The heat flux is uniformly distributed on each unit surface and the temperature field is steady state. The material of the seal ring and the properties of sea water are constant. The influence of heat loss due to thermal radiation, material wear and the change of liquid film viscosity between the end faces is ignored.

2) During the operation of the seal ring, the friction coefficient conforms to Coulomb's law and remains constant. The friction heat is absorbed by the passive static ring friction pair. In the calculation, the heat flux inputs of the static and dynamic rings are treated as boundary heat flux inputs. The influence of the heat generated by the friction between the working surface of the dynamic and static ring and the medium (seawater) in the seal cavity (i.e. the heat of stirring) on the temperature field of the spherical stern shaft seal ring of the stern shaft is ignored.

3.2 boundary conditions

3.2.1 Axis offset simulation

It is assumed that the center of the aft end face of the stern shaft segment coincidences with the center of the sealing sphere. When simulating the axial offset of the stern shaft. Let the length of the stern shaft segment be $l$, which is equivalent to the radius $R_s$ of the sealing sphere. The concentrated force is applied to the heart of the ball to produce the maximum deflection $y$. As shown in Figure 5.
Fig. 5. Diagram of excursion of stern shaft axis

The deflection can be regarded as the vertical offset $y$ of the ball center, that is, as the offset of the stern shaft axis.

$$y = \frac{F l^3}{3 E I}$$  \hspace{1cm} (1)

$$I = \frac{\pi}{64} (d^4 - D^4)$$  \hspace{1cm} (2)

Where: $l$ is the length of the stern shaft segment, mm; $E$ is the elastic modulus of the shaft material, Mpa; $I$ is the polar moment of inertia of the stern shaft section, mm$^4$; $d$ is the outside diameter of the stern shaft, mm; $D$ is the inner diameter of the stern shaft, mm.

When the vertical offset of the tail axis (0, 1, 2 mm) is determined, the corresponding concentrated force $F$ exerted on the center of the ball can be obtained through Equation (1), which are 0, 3.51e5, 7.02e5 N respectively.

### 3.2.2 External load application

The spring specific pressure constant $p_{sp}$ of the spherical stern shaft seal coupling model is 0.21 MPa, the seawater pressure $p_{w}$ is 5 MPa, and the corresponding water depth is 500 m respectively.

### 3.2.3 constrained boundary

(1) The translational freedom of the fixed end of the spring in X, Y and Z three-dimensional space in all directions and the angular displacement freedom of the three coordinate axes of OX, OY and OZ; Constraint three translational degrees of freedom at the head of the stern shaft in X, Y, and Z directions.

(2) In the case of axis alignment (no vertical deviation), full constraints are applied to the rear end of the stern shaft segment to ensure that no deviation occurs to the axis.

The mechanical boundary conditions were applied to the stern shaft-spherical stern shaft seal coupling model, as shown in Fig. 6.

Fig. 6. Loading results of mechanical boundary conditions
3.2.4 Thermodynamic boundary condition

The convective heat transfer coefficient $\alpha$ between seawater and moving ring is determined by empirical formula.

$$\alpha = \frac{(N_u)\lambda}{d}$$  \hspace{1cm} (3)

Thereinto,

$$N_u = 0.023(R_e)^{0.8} \times (P_r)^{0.4}$$  \hspace{1cm} (4)

$$R_e = \frac{Vd}{\nu}$$  \hspace{1cm} (5)

$$V = \sqrt{\left(\frac{\pi dn}{60}\right)^2 + \mu^2}$$  \hspace{1cm} (6)

Where, $\lambda$ is the thermal conductivity of seawater, $\lambda = 60.85 \text{W/(m·℃)}$; $Nu$ is the Nussel number; $Re$ is Reynolds number; $V$ is the synthetic velocity of seawater at the outer diameter of the moving ring, m/s; $D$ is the outer diameter of the moving ring, m; $U$ is the axial velocity, $U = 10$ m/s; $n$ is the speed of the tail shaft, $n = 200$ r/min; $\nu$ is the kinetic viscosity of seawater, $\nu = 1.304 \text{m}^2/\text{s}$; $Pr$ is the Prandtl number of the medium (seawater) in the seal cavity of the stern shaft, $Pr = 6.22$.

A convective heat transfer boundary is added in the near area of the inner and outer working surfaces of the friction pair. According to different fluid media, the convective heat transfer boundary can be divided into two parts: the inner side (air) convective heat transfer boundary and the outer side (sea water) convective heat transfer boundary.

3.2.5 Methods Verify the conclusion

When underwater vehicle stern shaft axis offset 0, 1 and 2 mm, respectively, in order to calculate the stern - overall deformation of spherical stern shaft seal coupling model, the application type (1) in the center of the shaft after the period of end face (its) on the corresponding load $F$, to simulate a shaft axis on the vertical migration of numerical (i.e., its offset), as shown in Fig. 7.

(a) Vertical without deviation

(b) Vertical offset 1mm
Deviation in Fig. 7(a) numerical $y = 0$, the deviation in Fig. 7(b) numerical $y=0.001195m$, the deviation in Fig.7(c) numerical $y=0.00239m$, the results respectively with the axis of the expected vertical migration value of 0, 1 and 2 mm basic equal, proved the rationality of the simulation method.

4.Thermal-structural coupled finite element analysis

4.1 Contact pressure of sealing surface

Under different working conditions and axial vertical deviation, the contact pressure of the sealing surface changes, which directly affects the deformation and temperature distribution of the sealing surface.

When the depth is 500 m, the rotation speed is 30,90,150,180 r/min, and the vertical offset of the stern shaft axis is 0,1,2 mm, respectively, the contact pressure peak of the sealing surface of the static ring is shown in Table 1.

The water depth is 500 m, the rotation speed is 180 r·min⁻¹, and the contact pressure cloud diagram of the sealing surface of the static ring with different vertical axes offsets is shown in Fig. 8.

| Depth H/m | Rotation speed n/(r/min) | Vertical displacement /mm | 2 – 0* Difference |
|----------|--------------------------|---------------------------|-------------------|
| 500      | 30                       | 4.06, 4.13, 4.20          | 0.14              |
|          | 90                       | 3.99, 4.06, 4.14          | 0.15              |
|          | 150                      | 3.93, 4.00, 4.07          | 0.14              |
|          | 180                      | 3.90, 3.97, 4.04          | 0.14              |

2 – 0 Note: Represents the peak contact pressure difference between the sealing face when the offset is 2 mm and 0 mm. MPa.

It can be seen from Table 1 and Figure 8 that, under different working conditions, the peak contact pressure of the sealing sphere has a consistent trend with the change of rotational speed and vertical deviation. When the vertical offset is constant, the peak value of contact pressure on sealing surface decreases with the increase of rotating speed. The peak value of sealing surface contact pressure increases with the increase of vertical axis deviation. From the comparison between the axis offset of 2 mm and the axis offset of no axis, it can be seen that at different rotating speeds, the increase range of the peak contact pressure on the sealing surface basically keeps at about 0.14MPa, without obvious fluctuation. The results show that the spherical stern shaft seal has good adaptability and the servo performance of the static ring-spring system.
4.2 Heat flux of sealing surface

The temperature of the sealing surface is closely related to the heat flux. The heat flux vectors of the inner and outer cylinders of the dynamic and static rings are shown in Fig. 9.
It can be seen from Fig. 9 that the heat generated by the friction of the sealing surface of the spherical stern shaft seal is transferred to both the inner and outer sides. According to the calculation formula of heat flux, the heat flux on the outer side of the sealing surface is larger because the contact pressure and linear velocity on the outer side of the sealing surface are both larger.

### 4.3 Temperature distribution on sealing surface

| Depth H/m | Rotation speed n/(r/min) | Vertical displacement /mm | 2 – 0* Difference |
|-----------|--------------------------|---------------------------|-------------------|
|           |                          | 0  | 1  | 2  |                |
| 500       | 30                       | 25.4 | 25.5 | 25.5 | 0.1            |
|           | 90                       | 36.0 | 36.1 | 36.1 | 0.1            |
|           | 150                      | 46.2 | 46.3 | 46.4 | 0.2            |
|           | 180                      | 51.1 | 51.2 | 51.3 | 0.2            |

2 – 0 Note: Denotes the difference of the peak temperature of the sealing surface when the offset is 2 mm and 0 mm, °C.

Based on the analysis in Table 2: (1) The peak temperature of sealing surface increases with the increase of rotating speed when the depth and vertical deviation of axis remain unchanged. This is due to the increase in the speed of heat flux caused by the increase. (2) The vertical deviation of axis has little effect on the peak temperature of sealing surface. Compared with the vertical deviation of the axis of 2 mm, it can be seen that the increase range of the peak temperature of the sealing surface is less than 0.2°C (the change rate is less than 0.7%) at different rotating speeds. The maximum temperature is 51.3 °C, lower than the limit temperature of the sealing ring material and the water film vaporization temperature (100 °C). This also shows that under different working conditions, the temperature change of the sealing surface is very small, and the spherical stern shaft seal has good adaptability and good following between the dynamic and static rings.

When the depth H is 500 m, the speed n is 180 r/min, and the vertical axis offset is 0,1,2mm, the moving global temperature distribution is shown in Fig. 10.
It can be analyzed by the figure: Under the condition of different vertical offset axes, the temperature distribution on the sealing surface of the moving ring is relatively uniform. However, eight equal-length arcs appear obviously in the whole annular distribution area, which is the result of the action of eight springs evenly distributed on the static ring seat.

5. summary
(1) Using ANSYS finite element software, a three-dimensional coupling model of the stern shaft-spherical stern shaft seal was established when the vertical offset of the stern shaft axis was 0 mm, 1 mm and 2 mm, respectively. The cantilever loading method was used to simulate the application of the vertical offset (i.e. the spherical center offset), which proved the rationality of the simulation method.
(2) At the same depth, no matter whether the axis is offset vertically or not, the radial distribution of heat flux on the sealing surface has the same trend as the rotation speed, and the heat flux on the outside is higher than that on the inside.
(3) Under the condition of different vertical axis offsets, the temperature distribution on the sealing surface of the moving ring is relatively uniform, but the whole annular distribution area obviously appears 8 equally long arcs, which is the result of the action of 8 springs evenly distributed on the static ring seat.
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