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Sealing Ability of Spherical Hull Transfer Skirts

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Abstract: An improved spherical, movable transfer skirt for autonomous submersibles has been devised. It was designed to permit the transfer of equipment and personnel from a submersible to the pressure chamber of an oil storage sea-bed structure. It also allowed mating at large vertical angles while the submersible remained horizontal. Seal failure modes and procedures for analyzing the sealing ability of the mating flange of the hull transfer skirt were thoroughly analyzed using conservative estimation methods. In the analysis, sea currents and mating angles were considered. Results showed that when considering the effects of currents, spherical radius and mating angle, their influence on seal ring failure should be considered first. The critical mating depth for a seal ring failure was larger than for either sliding or rotational failure modes. The critical mating depth can be used to determine the mating method of the submersible. The analytical procedures and results can be used as a reference for the design of spherical hull transfer skirts.

Keywords: submersible; transfer skirt; sealing ability; sea current
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1 Introduction

Offshore platform is used to exploit sea-bed energy and mineral resources, such as oil and gas. However, the cost of producing an offshore field purely by deviation drilling techniques from fixed platforms becomes uneconomical in very deep water. There is, therefore, a growing interest in sea-bed systems where oil is produced from wells at predetermined sites on the field and then collected at a single production platform before being either piped ashore or fed to a tanker loading facility. Pressure chambers of the sea-bed system were used to place the equipments and personnel. The method for accessing the sea-bed chamber is to use a free-swimming submersible which is capable of docking or mating with the chamber and transferring personnel at atmospheric pressure. The submersible needs the addition of a transfer skirt (an open-ended pressure vessel) around the hatch, which can seal onto the mating surface surrounding the access hatch of the sea-bed chamber to permit transfer of personnel and materials. Sometimes, sea-bed chamber needs diagonal hatches for equipment transfer. The transfer skirt (TS) allowing mating at a large angle can help increase the supply efficiency.

In early time, most submersibles are attached to a fixed TS system. But the fixed TS system requires high operability and controllability. During recent years, movable transfer skirt has been used, which allows mating at varying angles while the submersible remains in the horizontal position. For the TS hull, its sealing ability is one of the most important performances. Movable TS hull has more joints than fixed one. The contact surfaces of three joints: horizontal, diagonal and mating articulating joint, should be sealed. The large mating angle also affects the sealing ability of the TS system. Hence, it is necessary to investigate the sealing ability of the movable TS hull.

Many investigations about underwater mating device have been reported, but most of these researches focused on rescue bell (Wang *et al*., 2002; Driscoll *et al*., 2000). In recently years, more and more underwater mating devices have been used in deep submergence rescue. Some corresponding investigations were also presented. Schoof *et al*. (2007) thoroughly described the spherical transfer skirt hull of a pressurized rescue module system, whose maximal mating angle was 45°. In their study, the buckling strength of TS hatch was investigated by FEM and hydrostatic tests. George and Miller (2006) introduced the function of the components of the TS and the fabrication process of TS hull. Fu *et al*. (2008) considered the adaptability of the mating between TS of deep submergence rescue vehicle and the mating surface. Tang *et al*. (2009) presented the strength and stability calculation of TS hull by finite element method.

Although some studies have been presented on TS hull, few theories, collateral test results and techniques for sealing ability of the TS hull have been undertaken. Sheng *et al*. (2006) considered that the underwater mating has some key technologies including seal technology. Carey and Moncaster (1977) presented that TS system could be sealed onto the sea-bed chamber by the hydrostatic pressure. The sealing ability of the TS hull will increase with the increase of the mating depth. However, the steady currents may occur around underwater structures even in calm seas, and may reduce the sealing ability of the TS hull. Hu *et al*. (2007) suggested that the sealing ability of movable rescue bell could be calculated by a conservative estimate method. Since the current design

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criteria of submersibles (CCS, 1996) do not present any adequate calculation procedures for sealing ability of TS hull, it may be a reasonable approach for obtaining the critical mating depth which causes seal failure under the influence of sea current levels and mating angles by conservative estimate method. However, the structural configuration, loads of TS, is different from rescue bell. The method and results presented by Hu Y is not suitable for TS hull.

This paper provides descriptions of the seal failure modes of the spherical transfer skirt (STS) hull, the sealing ability analysis procedures for mating flange of the TS hull, and the curves of critical mating depth versus sea current level and mating angle under different seal failure models. The analysis procedures and results can be used to provide references for the design of STS hull.

2 Details of spherical transfer skirt

A typical STS hull is a spherical shell structure with articulating joints (Fig.1). The cylindrical adapter spool connects TS and the main hull. The spherical shell is divided into two parts, upper hull and lower hull, by a diagonal articulating joint. The upper hull is joined to the adaptor spool and can rotate 180 degrees perpendicular the *OE* axis through a horizontal articulating joint. The lower hull is joined to the upper hull and can also rotate 180 degrees perpendicular the *OC* axis through the diagonal joint. It is the lower hull that attaches to the hatch trunk of the sea-bed chamber during mating operations. The maximal mating angle is twice as large as the diagonal angle θ, and the horizontal, diagonal and mating flanges are used to reinforce the TS hull.

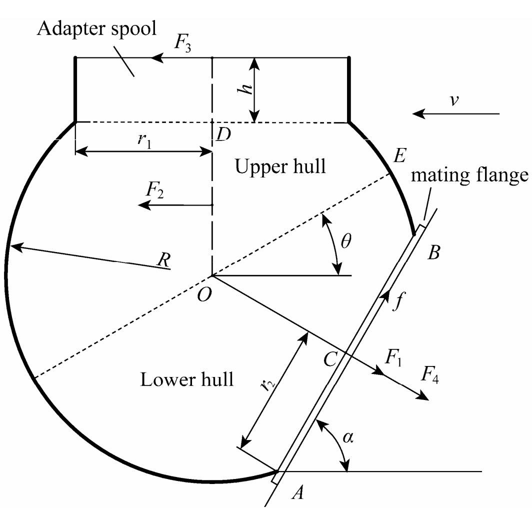

Fig.1 Load distribution of the TS hull

The hatch radiuses of the upper and lower hulls, r_1 and r_2, are assumed to be constant. Different spherical transfer skirts can be obtained with the variety of the diagonal angle. The horizontal, diagonal and mating flanges are used for the reinforcement of the transition region of the adapter spool, upper and lower hull. The design of the diagonal flange and joint presented by Mao *et al*. (2000) is used in this paper. Details of the mating flange and mating surface are shown in Fig.2, from which it can be seen that sea water can flow from *b* to *a* if the seal failure occurs. Swallow-tailed seal groove and *O*-ring are used for three flange seal surfaces.

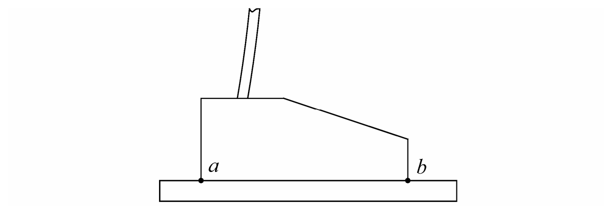

Fig.2 Sketch of the mating flange

3 Mechanical analyses of TS hull

3.1 Seal failure modes

Screw propellers of the submersibles for orientation can help mate with the sea-bed chamber successfully. Thus, it is more important to discuss the sealing ability of the TS system under the conditions that mating is successful. For the TS hull, three flange seal surfaces should be sealed. However, the mating angle has little effect on the sealing ability of horizontal and diagonal flanges, and the sealing ability of the mating flange is comparatively weaker compared with the other two flanges. Therefore, this work focuses on the sealing research of the mating flange.

The TS hull is subjected to water pressure and sea current force when mating is successful. Sea current is assumed to be steady and one way current, and the velocity is v (m/s). Some seal failure modes may occur:

1) Relatively sliding between the mating flange of TS hull and the mating surface of the sea-bed chamber may occur if the frictional force of the contact surface is too small. The sliding causes seal failure of the contact surface.

2) Under the influence of the sea current, TS hull may rotate around one point on the contact surfaces. As a result TS hull will deviate from the contact surface.

3) Under the influence of the sea current, the vertical compression force of the contact surface may be too small, which causes seal ring failure.

3.2 Mechanical analysis of the TS hull

Let us assume that the mating surface is fixed. Load distribution of TS hull is shown in Fig.1. It can be seen that TS hull is subjected to five forces: the force provided by water pressure F_1, the current force on the TS hull F_2, the current force on the main hull F_3, the reaction force provided by the mating surface F_4, and the friction force of the contact surface f.

1) Force provided by water pressure

$$F_1 = \pi r_2^2 \gamma g H \quad (1)$$

where γ is the density of sea water, H the operating depth, and g the acceleration of gravity.

2) Current force on the TS hull

$$F_2 = \frac{1}{2}\gamma S_2 C_d v^2 \quad (2)$$

where C_d is the resistance coefficient, C_d=1.0. S_2 is the encounter surface area of the TS hull, and will vary with the variety of the mating angle. When the mating angle α=0°, S_2 is expressed as

$$S_{21} = 2r_1h_1 + R^2(\pi - \arcsin\frac{r_1}{R} - \arcsin\frac{r_2}{R}) + r_1d_1 + r_2d_2 \quad (3)$$

where, d_1 and d_2 are the vertical distances from the center of the sphere to upper and lower hatch sections, d_1=L_{OD}, d_2=L_{OC} (see Fig.1). When the mating angle α is up to maximal, S_2 is described as

$$S_{22} \approx 2r_1h_1 + R^2(\pi - \arcsin\frac{r_1}{R}) + r_1d_1 \quad (4)$$

Let us assume that S_2 is proportional to the mating angle. The encounter surface area of the TS hull at any mating angle is expressed as below:

$$S_2 = \frac{\alpha}{\alpha_{\max}}(S_{22} - S_{21}) + S_{21} \quad (5)$$

where $\alpha_{\max}$ is the maximal value of the mating angle.

3) Current Force on the main hull

$$F_3 = \frac{1}{2}\gamma S_3 C_d v^2 \quad (6)$$

where S_3 is the encounter surface area of the main hull. Because the main hull can remain in the horizontal position at any angle when mating to the sea-bed chamber, S_3 is equal to the maximal cross section area of the submersible.

4) Reaction force of the mating surface

$$F_4 = F_1 - (F_2 + F_3)\sin\alpha \quad (7)$$

5) Friction force of the contact surface

$$f = \mu[F_1 - (F_2 + F_3)\sin\alpha] \quad (8)$$

where μ is the friction coefficient of the contact surface.

Sometimes, external forces, such as the down pressure provided by the main hull and the tension provided by the winch in the lower hull, need be applied to help the mating between TS and the mating surface. In this paper let us assume this kind of external force is from the winch force, which is denoted by F_0. F_0 Shares the same action point and direction with F_1. Now Eqs.(7), (8) can be expressed as:

$$F_4 = F_0 + F_1 - (F_2 + F_3)\sin\alpha \quad (9)$$

$$f = \mu[F_0 + F_1 - (F_2 + F_3)\sin\alpha] \quad (10)$$

4 Sealing ability of the mating flange

4.1 Sliding failure

If the sliding force is larger than the static friction force, the TS hull will slip along the mating surface and cause seal failure. Thus, sliding failure will not occur when

$$f \geq (F_2 + F_3)\cos\alpha \quad (11)$$

By substituting Eqs.(2), (6) and (10) into Eq.(11), the critical mating depths or the critical winch forces for sliding failure under different current levels and mating angles can be obtained.

4.2 Rotation failure

Under the influence of the sea current, the TS hull may rotate around point A on the contact surface. Rotation failure will not occur when

$$F_0L_0 + F_1L_1 \geq F_2L_2 + F_3L_3 \quad (12)$$

where L_0, L_1, L_2 and L_3 are the moment arms to point A for F_0, F_1, F_2 and F_3. L_0=L_1=r_2. For the TS hull, the most dangerous state is: L_2=R+d_1/3, L_3=R+d_1+h. The critical mating depths or the critical winch forces for rotate failure under different current levels and mating angles can be obtained with the aid of Eq.(12).

4.3 Seal ring failure

4.3.1 Vertical compression force

The vertical compression force of contact surface affects sealing ability of the seal ring. Its magnitude increases with the increase of the water pressure. However, the vertical compression force is asymmetric in circumferential direction. It is maximal at point A while minimal at point B. Therefore, seal ring failure of the mating flange may occur at point B. Let us assume that the vertical compression force distributes linearly from point A to point B. The forces per unit length at point A and B are denoted by Q_A and Q_B. Based on the moment equilibrium equation, we can obtain

$$F_4L_4 = F_0L_0 + F_1L_1 - F_2L_2 - F_3L_3 \quad (13)$$

The force per unit length at point B can be expressed as

$$Q_B = 10^{-3}(\frac{3}{2\pi r_2^2} \cdot F_4L_4 - \frac{F_4}{\pi r_2}) \quad \text{(N/mm)} \quad (14)$$

Combining Eq.(9) and Eq.(13), Q_B can be derived.

4.3.2 Sealing ability of seal ring

When the maximal contact pressure of seal ring is less than water pressure (working pressure), the seal of seal ring fails, namely

$$\sigma_{B\max} \geq p \tag{15}$$

where p is the working pressure, p=0.01H (MPa). $\sigma_{B\max}$ can also be expressed as

$$\sigma_{B\max} = \sigma_0 + Kp \tag{16}$$

where σ_0 is the compression stress provided by Q_B, and K is the pressure transfer coefficient. In the present study, swallow-tailed seal groove and O-ring seal were used. The similar structure with the same size was discussed by finite element method presented by Hu *et al*. (2007). According to their research, K was obtained by fitting some discrete results. These discrete results were calculated by a simple finite element analysis using Mooney-Rivlin model. The variation curve of maximal contact pressure versus working pressure for a typical sea ring is shown in Fig.3. The pressure transfer coefficient, K, was obtained with the aid of the variation curve and least square method. In this paper, K = 0.756. The values of σ_0 and Q_B corresponding to different amounts of compression of the O-ring were also obtained by finite element method (see Fig.4), by which the relationship between σ_0 and Q_B was regressed

$$\sigma_0 = -1\times10^{-4} Q_B^{\ 2} + 0.072 Q_B + 1.328 \tag{17}$$

By substituting Eq.(14) into Eq.(17), the critical mating depth or the critical winch forces for seal ring failure under different current levels can be obtained.

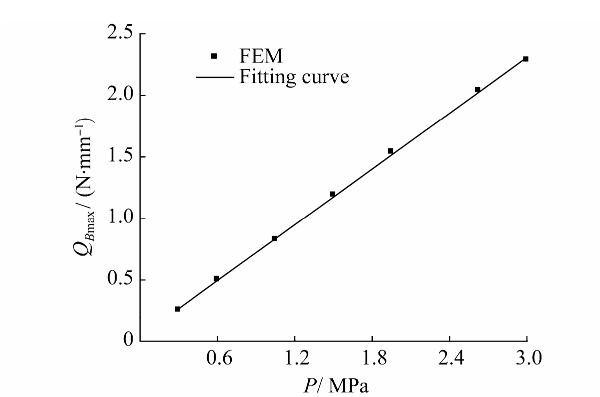

Fig.3 Variation of maximal contact pressure versus working pressure for a typical sea ring

5 Examples

5.1 Details of analytical model

Details of an analytical model are shown in Table 1. R_1 in Table 1 is the radius of the cylindrical main hull of the submersible. Based on the formulae illustrated in section 3, the values of the forces and the corresponding moment arm to point A can be calculated. Relationship between critical mating depth and other parameters (R, v and α) can also be obtained. In the next sections we'll discuss the variations of critical mating depth versus other parameters.

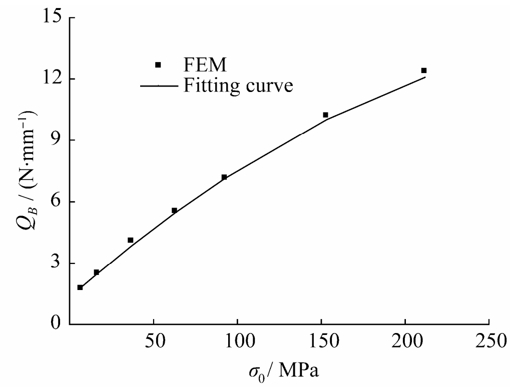

Fig.4 Variation of contact pressure versus compression stress for a typical sea ring

Table 1 Details of the analytical model

r_1/R_1	r_2/R_1	h/R_1	S_3/R_1^2	γ	μ	$\alpha_{\max}$
0.5	0.6	0.2	9.6	1025	0.3	45°

5.2 The influence of sea current and mating angle

The radius of spherical shell is assumed to be constant, i.e. R=0.9R_1. The critical mating depths for three seal failure modes under different current levels and mating angles can be derived and sketched in Figs.5~7. It can be seen that the critical mating depth increases with the increase of the current level. With the decrease of the mating depth, the seal ring failure will occur firstly, then rotation failure and lastly sliding failure. Under a certain current level, the sliding failure firstly occurs at α=28°, whilst the rotation and seal ring failures firstly occur at α=45° and at α=0°, respectively. The maximal value of H_{cr} for sliding and rotation failures is 14% larger than the minimal one, whilst the magnitude is 9% for seal ring failure. However, for the TS hull with $R \leq 0.9R_1$ and $\alpha_{\max} \leq 45°$, the seal failure may not occur if the mating depth $H > 25$ m.

When the mating depth is too lower and cannot provide enough water pressure, the seal failure occurs. So it is more important to obtain the minimal external forces (the critical winch force in this paper) under danger mating depth. The critical winch force, $(F_0)_{cr}$ for rotational failure mode and seal ring failure mode under different current levels and mating depths can be derived and sketched in Fig.8, Fig.9. The critical winch force for other sea failure modes can also be obtained using the analysis procedure presented in section 4.

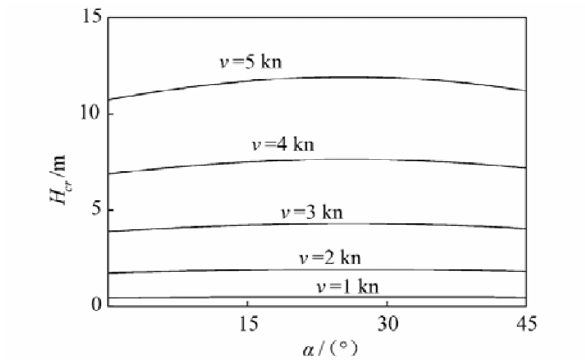

Fig.5 Variation of critical mating depth for sliding failure versus mating angle under different current levels

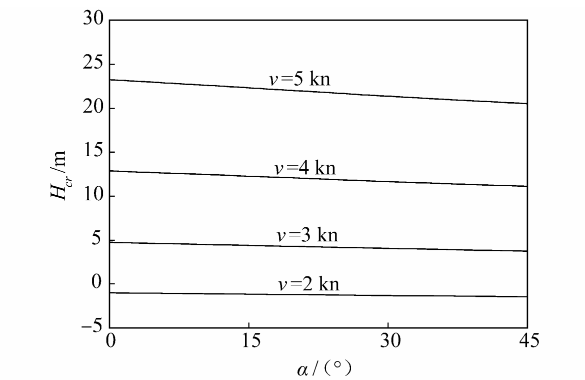

Fig.6 Variation of critical mating depth for rotation failure versus mating angle under different current levels

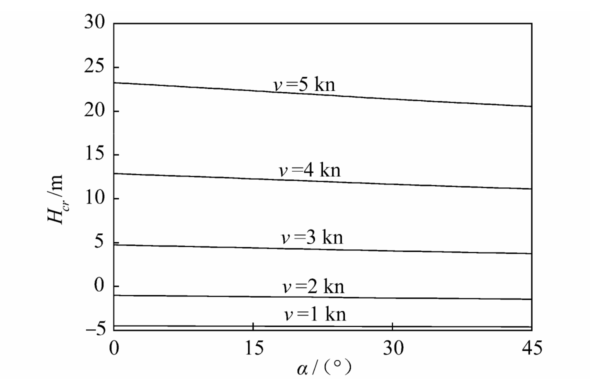

Fig.7 Variation of critical mating depth for seal ring failure versus mating angle under different current levels

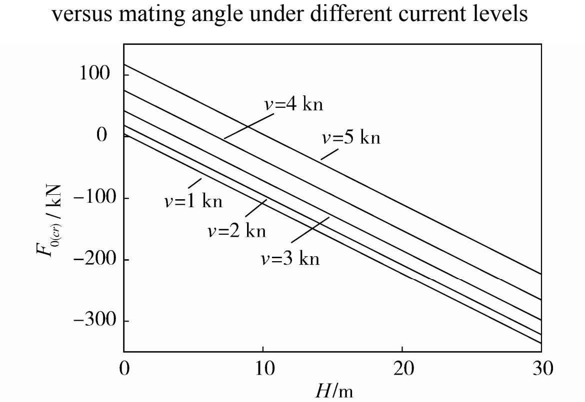

Fig.8 Variation of critical winch force for rotational failure versus current levels under different mating depths

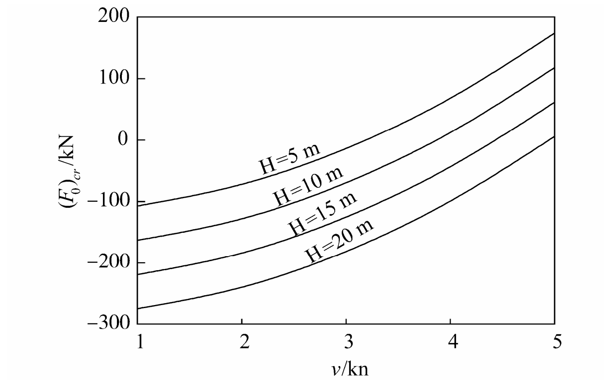

Fig.9 Variation of critical winch force for seal ring failure versus current levels under different mating depths

5.3 The influence of the radius

Variation of critical mating depth for three seal failure modes versus spherical shell radius when v=0.3 kn and α=45° is sketched in Fig.10, from which it can be seen that critical mating depths increase with the increase of the radius of the spherical shell, but the increments are lower for sliding and rotation failures whilst larger for seal ring failure. Therefore, seal ring failure should be firstly considered in the sealing research of the TS hull.

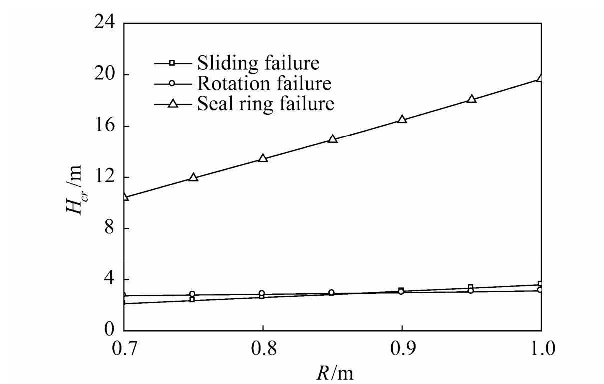

Fig.10 Variation of critical mating depth versus spherical shell radius with v =0.3 kn and α =45°

6 Conclusions

In this paper, the sealing ability of STS hull was studied by theoretical method. The seal failure modes were discussed and estimate formulae for checking the sealing ability of the mating flange were presented. Results show that under the influence of the current level, spherical radius and mating angle, the seal ring failure should be firstly considered. The critical mating depth for the seal ring failure is larger than that for the other failure modes. The critical mating depth can be used to determine the mating method of the submersible, namely, mating under wet or dry conditions.

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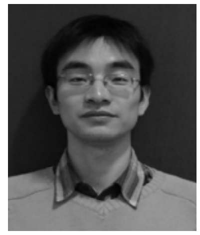

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