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Metal Sealing Performance of Subsea X-tree Wellhead Connector Sealer

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Abstract: The metal sealing performance of subsea X-tree wellhead connectors is crucial for the safety and reliability of subsea X-trees. In order to establish the theoretical relation between metal sealing ring's contact stress and its structural parameters and working pressure, a mechanical analysis method for double-cone sealing of high pressure vessels is applied in analyzing the metal sealing ring under the condition of preload and operation. As a result, the formula of the unit sealing load for the metal sealing ring under operation with residual preload is shown in this paper, which ensures that the metal sealing ring has an excellent sealing effect and can prevent the metal sealing ring from yielding. Besides, while analyzing the sealing process of the metal sealing ring, the change rule of contact stress and working pressure is concluded here, putting forward that the structural parameters of the metal sealing ring are the major factors affecting the change rule. Finally, the analytical solution through theoretical analysis is compared with the simulation result through finite element analysis in a force feedback experiment, and both are consistent with each other, which fully verifies for the design and calculation theory on metal sealing ring's contact stress and its structural parameters and working pressure deduced in this paper. The proposed research will be treated as an applicable theory guiding the design of metal seal for subsea X-tree wellhead connectors.

Keywords: sealing ring, metal sealing, mechanical analysis, subsea X-tree wellhead connector, subsea X-tree

1 Introduction

During the trial running of a subsea X-tree, the performance of each component and mechanism affects the usability of this equipment[1–3]. And the performance of the sealing structure directly determines the safety and reliability of the subsea X-tree[4]. At present, the long lifetime of subsea X-trees requires that the metal seal is applied in all major sealing points[5–6].

Studies on metal seal abroad were carried out much earlier, and technologies in this field are also fully mastered in foreign countries. Yet in China, researches into this area have just started. In 1987, overseas researchers analyzed sealing rings made of materials with different yield strengths under loading flexure deformation, and the entirety of the stress from the deformed steel wheel[7–8]. KELLY, et al[9], studied the whole X-tree, including the mechanism guiding subsea wellhead sealing and the selection of different types of sealers. ADAM, et al[10], analyzed the structure characteristics and the principle of metal-to-metal sealing. KUROKOUCHI, et al[11], carried out a sealing performance analysis in conical metal sealers and realized their structural optimization using finite element software. In China, LIU, et al[12], carried out a numerical simulation analysis on metal seal structure by using penalty function algorithm based on contact surface and the finite element numerical calculation method. FEI, et al[13], studied the contact strength and deformation of sealers of high-pressure sealed containers, in the oil, chemical and other industries by means of finite element method. GONG, et al[14], established a finite element model of metal W ring by using the finite element software—ANSYS, and did a nonlinear finite element analysis on the metal W ring's elastic-plastic deformation in a given condition. CUI[15] introduced researches both at home and abroad on metal sealers applied in downhole devices, and exposited the mechanism of metal sealers and analyzed the major factors influencing the sealability.

The performance of metal sealing relates closely to their material, structure, and the environment. Due to the high-temperature and high-pressure environment that subsea connector metal sealers stay in and the corrosive effects from seawater and oil/gas media, the requirement on their performance is high. As there is no standard option for subsea connectors' metal sealers, and foreign countries do not want to share their core technologies, the design theory of subsea wellhead connector metal sealers should be mastered and improved independently to meet the demands in China. This paper focuses on the relation between seal

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ring's contact stress, structural parameters and the operating pressure by analyzing the mechanism of metal sealers on subsea wellhead connectors, in order to provide the design base for seal ring structure.

2 Structure and Theory of Sealing

2.1 Sealing structure

Subsea X-trees are connected to wellheads by the subsea wellhead connectors. The main seal is located at the connected point between the X-tree and the wellhead. The sealing point is shown in Fig. 1. Subsea connector sealing rings which seal the joint clearance between the tree body and the wellhead can prevent seawater from coming into the high pressure well annulus and also prevent inside medium outflow. The VX gasket, studied in this paper, is among various types of the sealing structures, such as Grayloc, AX gasket, and so on, as is shown in Fig. 2.

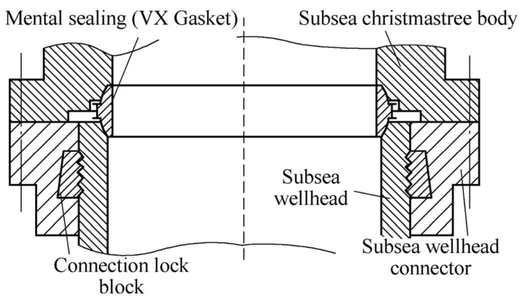

Fig. 1. Sketch of a sealing point

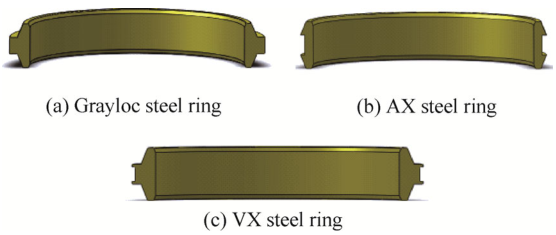

Fig. 2. Different gasket structures

2.2 Sealing theory

The VX ring seal is a self-tight seal, with an amount of interference existing between the gasket and the groove. During install/preload stage, this interference leads to radial compression on the VX gasket. During operation (when internal pressure exists), the increasing of inner pressure will lead to a gradual offset of the inner pressure by the radial and axial components of preload force. On one hand, the elastic spring-back of the gasket pre-compression maintains a considerable amount of preload on the sealing cone. And on the other hand, the internal pressure acting radially on the gasket further increases the pressing force on the sealing surface. The initial and preload states of VX gaskets are shown in Fig. 3.

To guarantee a VX gasket sealing, sufficient radial deformation and cone contact surface are needed, with no leaks around the circumferential direction. When the minimum contact stress of the sealing cone is greater than unit sealing load, the gasket provides sealing effectively.

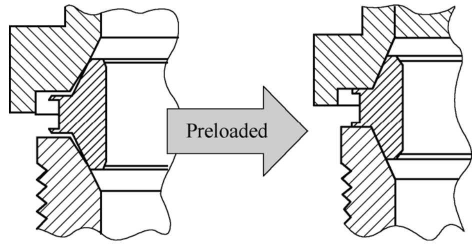

Fig. 3. Initial and preload states of VX gasket

3 Design Theory of the VX Gasket

Before designing, it is necessary to finish the theoretical analysis on the structural force of VX gasket. The structural force analyses are conducted for stages, i.e., preload (installation) and operation (with internal pressure) conditions, in order to obtain the impacts of sealing contact pressure and structure parameters on sealing performance. Assuming that the deformation of VX gasket is always kept within the range of elastic deformation range at all preload (installation) and operation (with internal pressure) conditions.

3.1 Preload condition

Mechanical analysis on the conical surface of the preload VX gasket[16–18] is shown in Fig. 4.

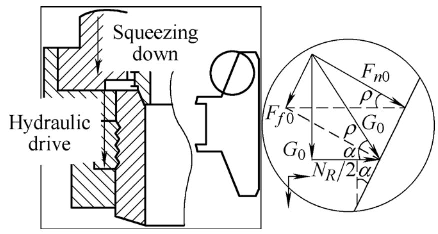

Fig. 4. Mechanical analysis of the sealing surface preloaded

When the seal ring is preloaded, the normal pressure of the conical surface can be calculated in accordance with the following formula:

$$F_{n0} = \pi D_m b q_0, \tag{1}$$

where D_m is the mean diameter of the VX gasket; b is the width of the actual interface which is smaller than the full width; q_0 is the contact pressure, which is no less than sealing specific pressure in order to prevent seal failure.

At this point, the snap ring of the connector starts working. The X-tree body connecting with connector has a downward movement tendency; so friction force F_{f0} on the sealing surface is downward, F_{f0}, F_{n0} and their resultant force G_0 form a force triangle, as shown in Fig. 4. Resultant force can be calculated in accordance with the following formula:

$$G_0 = \frac{F_{n0}}{\cos\rho} = \frac{\pi D_m b q_0}{\cos\rho}, \tag{2}$$

where ρ is the friction angle between the seal and seal groove, which is relevant to materials and surface appearance of the two contacting faces.

Resultant force G_0 can be decomposed into an axial force and a radial force. It can be considered that the axial force is the axial preloaded force W_0, and the radial force is resilience force N_R when the gasket is compressed. And the force on a single sealing surface is $N_R/2$

$$W_0 = G_0 \sin(\alpha+\rho) = \frac{\pi D_m b q_0}{\cos\rho}\sin(\alpha+\rho), \tag{3}$$

$$\frac{N_R}{2} = G_0 \cos(\alpha+\rho) = \frac{\pi D_m b q_0}{\cos\rho}\cos(\alpha+\rho), \tag{4}$$

where α is the included angle between the conical surface of the gasket and the vertical direction.

When the preload gasket is compressed to the maximum, the compression quantity is 2Δ. And when the radial deformation is 2Δ, circumferential strain ε_θ and stress σ_θ are respectively calculated:

$$\varepsilon_\theta = \frac{\pi(D_m - 2\Delta) - \pi D_m}{\pi D_m} = -\frac{2\Delta}{D_m}, \tag{5}$$

$$\sigma_\theta = E_R \varepsilon_\theta = -E_R \frac{2\Delta}{D_m}, \tag{6}$$

where E_R is the elasticity modulus of the gasket material.

Because of its axisymmetric structure, half of the gasket is intercepted for the static equilibrium analysis and the free-body diagram is shown in Fig. 5. The resilience force of the gasket is $N_R/\pi D_m$ within the unit circumferential length. According to the static equilibrium condition, it can be obtained that:

$$\int_0^\pi \frac{N_R}{\pi D_m} R_m \sin\varphi \,\mathrm{d}\varphi = -2F_R\sigma_\theta = 2F_R E_R \frac{2\Delta}{D_m}, \tag{7}$$

where F_R is the sectional area of gasket; R_m is the radius of the mean diameter circle; φ is the included angle between the resilience force of gasket in unit circumferential length and the section of gasket.

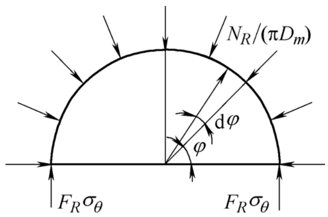

Fig. 5. Static equilibrium of gasket in circumference

The relation between resilience force N_R and circumferential stress σ_θ can be obtained with Eq. (7):

$$N_R = \frac{4\pi F_R E_R \Delta}{D_m}. \tag{8}$$

When the radial deformation of gasket is 2Δ and the force is $N_R/2$ in single surface, Eq. (8) is used to calculate resilience force N_R. Combining this equation with Eq. (3) and Eq. (4), the relation between contact pressure and the radial deformation can be obtained:

$$q_0 = \frac{2F_R E_R \Delta}{D_m{}^2 b} \bullet \frac{\cos\rho}{\cos(\alpha+\rho)} = \frac{W_0}{\pi D_m b} \bullet \frac{\cos\rho}{\sin(\alpha+\rho)}. \tag{9}$$

Therefore, when gasket is preloaded, the radial deformation of gasket Δ (the radial preloaded compression quantity) or axial pre-tightening force will play a decisive role in sealing specific pressure. The radial preloaded compression quantity Δ can be reflected by the magnitude of interference between the seal and seal groove, i.e., the downward displacement of the snap ring of connector in hydraulic driving.

3.2 Operation condition

Under the influence of inner pressure and preload, the operation contact press should be greater than the following equation:

$$F_n = \pi D_m b m p, \tag{10}$$

where F_n is the compression force of gasket coefficient; p is the inner pressure; m is the gasket coefficient.

(1) Force analysis. During operation, axial forces of the connectors are composed of four parts: preloaded axial load W_0, axial load caused between inner pressure and connector W_P, axial load caused by the inner pressure applied on the inner side surface of gasket W_1' and axial load generated by the preload gasket's spring back W_2'.

1) Preloaded axial load;

Calculated of W_0 is as shown Eq. (3).

2) Axial load caused between inner pressure and connector W_P:

$$W_P = -p s_P, \tag{11}$$

where p is the inner pressure; s_P is the area of connector loading the inner pressure in axial direction.

3) Axial load W_1' caused by the inner pressure applied on the inner side surface of the gasket.

As internal pressure acting on the inside of the seals, the seal ring will produce a radial self-tightening force. This force can be divided into the corresponding axial force W_1 and the normal compaction force F_{n1} on the cone. The radial self-tightening force on a single gasket sealing surface is

$$N_p = \pi D_m h p, \tag{12}$$

where h is the effective height of the internal pressure.

With the increase of its inner pressure, the connector has an upward tendency and the gasket moves downwards relatively, generating friction F_{f1} on the cone as shown in Fig. 6. The resultant force of friction F_{f1} and normal compaction force F_{n1} can be divided into radial self-tightening force $N_p/2$ and axial force W_1:

$$W_1 = \frac{N_p}{2} \bullet \tan(\alpha-\rho) = \frac{1}{2}\pi D_m hp\tan(\alpha-\rho), \qquad (13)$$

W_1 is the reacting force of W_1' acting on the connector:

$$W_1' = -W_1 = -\frac{1}{2}\pi D_m hp\tan(\alpha-\rho). \qquad (14)$$

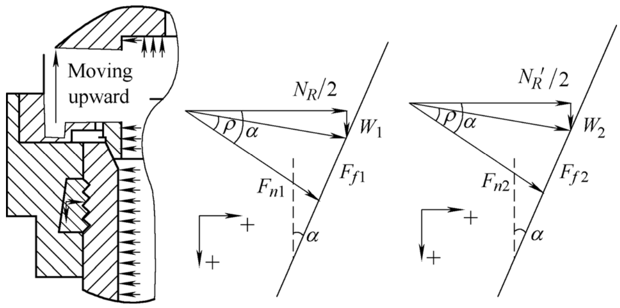

Fig. 6. Forces in operation condition

4) Axial load W_2' generated by the spring-back preload gasket.

Because of the influence of its inner pressure, the connector moves upward[19–21], and the pre-compression of gaskets is released when preloaded. This changes preload spring-back force to $N_R'/2$, which can be calculated from Eq. (8):

$$N_R' = \frac{4\pi F_R E_R \Delta'}{D_m}, \qquad (15)$$

where Δ' is the compression after spring-back.

The resultant force of friction and normal compaction force caused by spring-back can be divided into radial force $N_R'/2$ and axial force W_2. The force triangle is shown in Fig. 6.

$$W_2 = \frac{N_R'}{2} \bullet \tan(\alpha-\rho) = \frac{2\pi F_R E_R \Delta'}{D_m}\tan(\alpha-\rho), \qquad (16)$$

W_2 is the reacting force of W_2' acting on the connector, and

$$W_2' = -W_2 = -\frac{2\pi F_R E_R \Delta'}{D_m}\tan(\alpha-\rho). \qquad (17)$$

Then the joint force on the connector in the opposite direction of preload force can be calculated as:

$$W_\Sigma = W_P + W_1' + W_2' = -ps_P - \frac{1}{2}\pi D_m hp\tan(\alpha-\rho) - \frac{2\pi F_R E_R \Delta}{D_m}\tan(\alpha-\rho). \qquad (18)$$

(2) Operation process analysis. As the inner pressure increases, the result of the pressure axial force is the offset axial preload p is

$$p_{\min} = \frac{G}{\frac{1}{2}\pi D_m h\tan(\alpha-\rho) + s_p}. \qquad (19)$$

At this moment, the minimum operation contact press consists of radial load caused by inner pressure and radial preloaded compression.

Then, the operation contact pressure is caused by

$$W'' = \frac{N_p + N_R'}{2} \bullet \tan(\alpha-\rho) = F_n\frac{\sin(\alpha-\rho)}{\cos\rho}. \qquad (20)$$

Thus it could be acquired that VX gasket is pressure-compact. When designing the high-pressure vessel of double-cone sealers, self-tightening factor is the ratio of the contact stress to operation sealing pressure, and its value represents the seal pressure self-tightening ability:

$$\Psi = \frac{W_1}{W''} = \frac{h}{2bm}\frac{\cos\rho}{\cos(\alpha-\rho)}. \qquad (21)$$

The contact pressure is

$$q = \frac{N_R' + N_p}{2\pi D_m b} \times \frac{\cos\rho}{\cos(\alpha-\rho)} = \frac{W''}{\pi D_m b\tan(\alpha-\rho)} \times \frac{\cos\rho}{\cos(\alpha-\rho)} \geqslant mp. \qquad (22)$$

(3) Operation contact pressure. Δ' is unknown in Eq. (20). Obviously Δ' is related to p. So Eq. (22) cannot thoroughly explain the relation between operation contact pressure q and other structure parameters.

Due to the influence of the inner pressure, radial deformation Δ expands to Δ', and the amount of the expiation is δ

$$\Delta - \Delta' = \delta. \qquad (23)$$

At the same time, the upward move of the connector is ΔL_l

$$\Delta' = \Delta - \Delta L_l \tan\alpha = \Delta - (ps_p + W'' - W_0)\frac{L_1}{E_1 f_1}\tan\alpha, \qquad (24)$$

where E_1 is the elasticity modulus of the connector's material, L_1 is the length of the elongate components of the connector, f_1 is the sectional area of the elongate components of the connector.

Combine Eq. (20) and Eq. (24),

$$W'' = \left[\frac{1}{2}\pi D_m h\tan(\alpha-\rho) - \frac{2\pi F_R E_R L_1 s_p}{D_m E_1 f_1}\tan(\alpha-\rho)\tan\alpha\right]\times p\Big/\left[1+\frac{2\pi 2\pi F_R E_R L_1}{D_m E_1 f_1}\tan(\alpha-\rho)\tan\alpha\right] + \frac{2\pi F_R E_R}{D_m}\left(\Delta+\frac{W_0 L_1}{E_1 f_1}\tan\alpha\right)\tan(\alpha-\rho)\Big/\left[1+\frac{2\pi 2\pi F_R E_R L_1}{D_m E_1 f_1}\tan(\alpha-\rho)\tan\alpha\right]. \tag{25}$$

The contact pressure is

$$q = \left\{\left[\frac{1}{2}\pi D_m h - \frac{2\pi F_R E_R L_1 s_p}{D_m E_1 f_1}\tan\alpha\right]p + \left(\Delta+\frac{W_0 L_1}{E_1 f_1}\tan\alpha\right)\frac{2\pi F_R E_R}{D_m}\right\}\Big/\left[1+\frac{2\pi 2\pi F_R E_R L_1}{D_m E_1 f_1}\tan(\alpha-\rho)\tan\alpha\right]\times \frac{\cos\rho}{\pi D_m b\sin(\alpha-\rho)} \geqslant mp. \tag{26}$$

3.3 Structural strength design

When designing seals, the size of the structure should be determined in the first place. The axial preload acquired according to calculations in the last section can ensure sealing performance. Then with the determined structure size and axial preload, the strength of seals can be verified. By adjusting the structure size and axial preload, the stress can finally meet check rules.

3.3.1 Preload condition

The strength danger section of the VX gasket is its sealing surface. Set sealing surface as the datum plane at this point. Three principal stresses are respectively: normal stress σ_0 on the sealing surface, shear stress τ_0 and hoop stress σ_θ, the values of which are respectively:

$$\sigma_0 = -\frac{W_0\cos\rho}{\pi D_m b_0\sin(\alpha+\rho)}, \tag{27}$$

$$\tau_0 = \frac{W_0\sin\rho}{\pi D_m b_0\sin(\alpha+\rho)}, \tag{28}$$

$$\sigma_\theta = -\frac{2\Delta}{D_m}E_R. \tag{29}$$

3.3.2 Operating condition

The strength danger section of the VX gasket is the surface and inner place of the sealing element. Under this condition, the three principal stresses whose datum plane is the sealing face are: normal stress σ_0', shear stress τ_0' and circumferential stress σ_θ'. Their values respectively are:

$$\sigma_0' = -\frac{W''\cos\rho}{\pi D_m b_0\sin(\alpha-\rho)}, \tag{30}$$

$$\tau_0' = -\frac{W''\sin\rho}{\pi D_m b_0\sin(\alpha-\rho)}, \tag{31}$$

$$\sigma_\theta' = -\frac{2\Delta'}{D_m}E_R. \tag{32}$$

While the three principal stresses whose datum plane is inner place of sealing element are: normal stress σ_0'', shear stress σ_m'', and circumferential stress σ_θ'', which are, respectively

$$\sigma_0'' = -p, \tag{33}$$

$$\sigma_m'' = -\frac{W''}{\pi(B-b_0\sin\alpha)D_m}, \tag{34}$$

$$\sigma_\theta'' = -\frac{2\Delta'}{D_m}E_R. \tag{35}$$

σ_s is the yield strength of the seal material. Use the fourth strength theory of VX gasket for stress checking. The fourth strength theory is a distortion energy density theory, suitable for the condition where similar three-way stress can cause plastic deformation. The criterion is

$$\sqrt{\frac{1}{2}[(\sigma_1-\sigma_2)^2+(\sigma_1-\sigma_3)^2+(\sigma_2-\sigma_3)^2]} \leqslant [\sigma_s]. \tag{36}$$

4 Result and Comparison

Based on the analytical analysis, we obtain the seal face contact stress analytical equation from preload phase to operation phase, e.g., Eqs. (9), (22), and (32). They reveal the relations between specific seal pressure and the structure parameter of the VX seal gasket.

It can be seen that there is a close relation between q_0 and axial compressions Δ. The larger axial compression is, the more the axial preload is needed producing bigger seal contact stress, thus easier for sealing. Under operation condition, internal pressure will elevate the connector, hence the amount of compression must be bigger than the maximum lift displacement, in order to ensure sealing[22–23]. But there is still a problem. Bigger axial preload will lead to larger radial compression, which may generate yield stress of the VX gasket. Therefore it cannot be reused, and even may collapse. This issue needs to be taken into consideration during designing. In order to ensure sealing, it is necessary to obtain the radial compression and the applied axial preload when designing.

Under operation condition, the rise of internal pressure may cause a period of contact stress's reduction at sealing surface of the VX gasket, i.e., during operation, when residual preload exists, contact stress decreases while internal pressure increases. At this stage, the contact stress is at its lowest value, consisting of the radial load caused by inner pressure and radial preloaded compression.

With the internal pressure continuing to increase, contact stress can be calculated using Eq. (29). It can be seen that q and p are in a linear relation, but it is the factor $\frac{1}{2}D_m h - \frac{2F_R E_R L_1 s_p}{D_m E_1 f_1}\tan\alpha$ that determines the value of the slope and whether it is positive or negative. When the slope is positive, q will increase as p increases; otherwise, q will decrease as p increases, as is shown in Fig. 7. As it can be seen, the parameters are structural parameters of the VX gasket, so when designing the VX gasket, the factor value should be positive, which means to increase the width of the sealing surface.

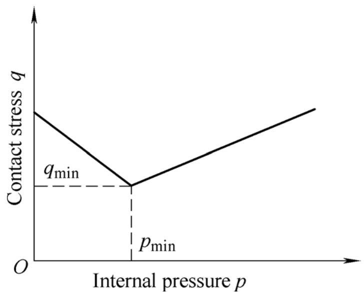

Fig. 7. Relation between contact stress and internal pressure

Putting design into practice, there exists a cone angle difference β between the cone angle of the VX gasket and the seal groove[24–26]. This will make the seal surface produce stronger sealing stress under preload, making sealing more feasible.

5 Force Feedback Experiments

Case: There is a connector connected to a 18-3/4 inch wellhead. The design pressure is 34.5 MPa (5000 psi). The rationality of the design of the VX gasket needs to be verified. The weight G of the subsea X-tree body is 50 t, and subsea X-trees, subsea wellheads and subsea wellhead connectors are made of alloy steel 8630, and their elasticity modulus E is 210 000 MPa, Poisson's ratio μ is 0.3, and yield strength σ_s is 550 MPa. The VX gasket is made of austenitic stainless steel 316L, its elasticity modulus E is 195 000 MPa, Poisson's ratio μ is 0.3, yield strength σ_s is 287 MPa, unit sealing load y is 179.5 MPa and gasket coefficient m is 6.5. Relevant structural parameters of the gasket are shown in Table 1, and friction angle ρ is 8.5° here.

Table 1. Structural parameters of the VX gasket

Mean diameter D_m/mm	Height h/mm	Thickness B/mm	Width b_0/mm	Included angle α/(°)	Cone angle difference β/(°)
510	102	15	30	22.75	1/4

According to Eqs. (9), (26) and (36) and the structural parameters in Table 1, the range of radial preloaded compression can be obtained as 0.201–0.447 mm, and axial preload as 864–1853 kN.

(1) When the influence from preload force exists, the finite element analysis shows that the general trends of VX gasket's contact stress is consistent with the theoretical conclusion, as VX gasket's contact stress increases along with the increase of axial preload. And when the axial preload is within the given range, the VX gasket can meet unit sealing load requirement and will ensure sealing (as is shown in Fig. 8). When the axial preload increases to a certain extent, the increase of contact stress slows down. This is because the contact width increases along with the increase of the preload, causing distribution of contact stress on the sealing surface to decline.

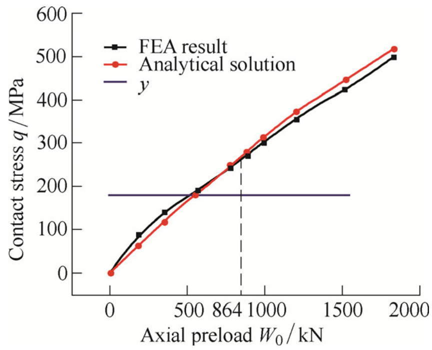

Fig. 8. Axial preload vs. contact stress

(2) When the preload force influence disappears, set $\Delta = 0.35$ mm, and the corresponding W_0=1753 kN. Here, operation contact surface pressure q satisfies operating seal pressure conditions. According to Eq. (19), $p_{\min}$=3.1 MPa. By calculating, VX gasket contact surface stress is still greater than the minimum value, $q_{\min} \geqslant mp$, meeting sealing requirements.

Finite element analysis shows that, as the operating pressure increases, contact stress of the VX gasket will increase at first and then decrease. As it can be seen from Fig. 9, the minimum value of contact stress appears when operation internal pressure is around 3 MPa, and will increase linearly with the increase of operating pressure, consistent with the theoretical trend of contact stress (as is shown in Fig. 7). In Fig. 9, the fitting curve slope of the contact stress obtained by finite element calculation equals to 2.93, while its counterpart calculated with the formula is 3.12. The reason why the two results are different is because the contact width value in theoretical calculation is smaller. Therefore, before preloading, larger radial compression has to be selected.

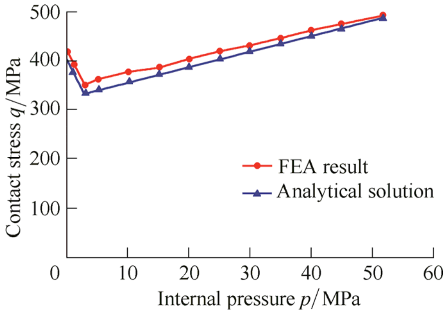

Fig. 9. Internal pressure vs. contact stress

According the above case, the design of the 34.5 MPa (5000 psi), 183/4 inch (476.25 mm) of subsea wellhead's supporting 183/4 inch subsea wellhead connector, the VX gasket is rational. Comparing the analytical solution through theoretical analysis with the simulation result through finite element analysis in a force feedback experiment, both are consistent with each other, fully verifying for the design and calculation theory on metal sealing ring's contact stress and its structural parameters and working pressure deduced in this paper.

6 Conclusions

(1) The VX gasket is a self-tight sealing. There is an amount of interference—radial pre-compression Δ, between the gasket and the seal groove. Radial pre-compression must be bigger than the upward distance of the connector to ensure sealing. During designing stage, it is necessary to calculate the amount of radial pre-compression and the axial preload force.

(2) Under operation condition, when preload remains, contact pressure decreases as the inner pressure increases. When $p_{\min}=\dfrac{G}{\frac{1}{2}\pi D_m\tan(\alpha-\rho)+s_p}$, the contact pressure is at its minimum value, which should meet sealing requirements.

(3) When the preload force influence disappears, q and p is in a linear relation. But the slope of the linear line is decided by factor $\frac{1}{2}D_m h-\dfrac{2F_R E_R L_1 s_p}{D_m E_1 f_1}\tan\alpha$. The slope is preferably to be positive.

(4) It can be seen that the VX gasket is a narrow line contact. With the increase of preload or inner pressure, it is difficult to calculate the exact value of sealing width. Besides, to prevent excessive compression or collapse of the VX gasket, it is necessary to combine Finite Element Analysis to ensure sealing.

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