

Article

An Assessment Method of Sealing Performance and Stress Intensity Factors at Crack Tip of Subsea Connector Metal Sealing Rings

Yingying Wang ^{1,*}, Cong Wang ², Guoheng Liu ³, Chong Zhang ⁴ and Jianchang Li ¹

¹ College of Safety and Ocean Engineering, China University of Petroleum-Beijing, Beijing 102249, China; 17356511576@163.com

² Shenzhen Branch of CNOOC, Shenzhen 518067, China; 2019211012@student.cup.edu.cn

³ CNOOC Research Institute, Beijing 100027, China; liugh6@cnooc.com.cn

⁴ Zhanjiang Branch of CNOOC, Zhanjiang 524000, China; zhangchong3@cnooc.com.cn

* Correspondence: wyy@cup.edu.cn; Tel.: +86-010-8973-9151

Abstract: Subsea connectors are of a critical part for the sealing of subsea production systems. The working environment makes cracks initiate easily on subsea connector sealing rings. In order to ensure the safety and reliability of the sealing rings, it is necessary to study the crack's influence on them. In this study, the main parameters that may influence stress intensity factors at crack tip are discussed. The sealing requirements of the subsea connector metal sealing rings were conducted. A finite element model was established to obtain the maximum equivalent stress and maximum equivalent plastic strain of crack-free sealing ring. Meanwhile, the influence of crack depths, crack positions, and crack angles on the sealing performance in preload and operating states was simulated through changing XFEM crack's parameters in ABAQUS software, as well as their influence on stress intensity factors at the crack tip. The research shows that although the cracks have little effect on the sealing performance of sealing rings in the early stage, the stress intensity factors increase with the crack depths. Long-term use leads to crack propagation, structure breakage, and sealing failure. The research results are of some reference value for improving the safety and reliability of subsea connectors in practical engineering applications.

Keywords: subsea connector; metal sealing ring; three-dimensional crack; stress intensity factor; sealing performance



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1. Introduction

Subsea connectors are one of the most essential facilities in subsea production systems. They are mainly used for sealing when the subsea equipment is connected. A sealed channel is formed for oil flow and preventing offshore oil and gas leakage, ensuring the safety and reliability of the efficient development of offshore oil and gas fields. Subsea connectors working on the sea floor for a long time makes them affected not only by waves, currents, and other environmental factors, but also the external environmental load, internal fluid pressure, and corrosion. As a result, cracks, corrosion, hydrogen embrittlement or fractures may emerge on the sealing parts, leading to an oil spill and multiple shut-down wells, which causes a large number of economic losses. In the event of a serious oil spill, its harm is difficult to estimate. Operating companies bear the loss of wells and crude oil, the subsequent clean-up of floating oil also consumes huge amounts of money. More importantly, it causes casualties, serious pollution of the marine environment, and destruction of the marine ecosystem [1]. Therefore, the sealing performance of subsea connectors is of great importance. Defects in pressure equipment may occur in the material itself, the production process, or over time in use. Most of these defects are directly manifested in the form of cracks, and some are fatigue cracks produced under alternated

loads. On the one hand, the presence of cracks disrupts the continuity of the material. Furthermore, stress concentration occurs at the crack tip, which constantly causes the crack to expand and eventually, breakage. Thus, cracks are major causes of low-stress brittle fracture, resulting in the sealing failure of subsea connectors. To ensure the performance of subsea connector sealing rings when cracks occur, it is necessary to study the crack's influence on them. Stress intensity factors can characterize the strength of the crack-tip stress field and strain field, so it is also an essential factor for assessing the sealing rings' performance. The main research questions are: (a) how to quantify the performance of subsea connector sealing rings; (b) and how to create different cracks in subsea connector sealing rings.

Regarding of the research on the performance of subsea connector sealing rings, scholars have performed a lot of work. Peng Fei et al. [2] introduced a subsea connector with metal cone gasket, established the calculation model for preload under different conditions, and formed an optimized design for the connector locking mechanism. Nelson and Prasad [3] talked about the sealing behavior of twin gaskets in a flange joint, studied the effect on contact stress with different bolt preloads, and internal pressure by finite element method, and proposed an empirical relation to determine the bolt preload for ensuring the minimum compressive stress. Wenbo Wang et al. [4] identified the failure cause of the steel seal ring in nuclear steam turbines by using metallographic analysis, scanning electron microscopy, nanoindentation, and in situ tensile testing. Kang Zhang et al. [5] proposed a theoretical calculation method for the pressure resistance of the connector sealing mechanism, introduced the expression relationship between the sealing contact load and the locking force by analyzing the load transfer relationship between different components, and derived the compression parsing expression by using the established contact model of the lens seal and the flange. Pogodin et al. [6] carried out the development and operation of new detachable connections with a lens sealing ring to assess the joint influence of technological factors on the tightness of the design.

In the past few decades, scholars have proposed a series of methods to assess the stress intensity factors of cracks. Cwiekala et al. [7] found crack-tip stress intensity factors for cracks nucleating from the edge of incomplete contacts. The general asymptotic formulation was used to incorporate the results in different contact problems as long as the contact was incomplete. Nabavi et al. [8] derived a general weight function to evaluate the thermal stress intensity factors of a circumferential crack in cylinders. The weight function derived was valid for a wide range of thin-to-thick-walled cylinders and relative crack depth and the accuracy of the analysis was examined using the finite element method results. Chang-Young Oh et al. [9] investigated the applicability of existing methods to estimate stress intensity factors due to welding residual stresses, comparisons with finite element (FE) solutions were made for two types of generic welding residual stress profiles, generated by simulating repair welds. Okano et al. [10] numerically investigated the effects of seal welding conditions, such as welding heat input, crack depth, and plate thickness on welding-induced residual stress and stress intensity factor (SIF) around the remaining crack. Bergara et al. [11] calculated stress intensity factors in prospective cracks of offshore mooring chains under service conditions using the conventional finite element method (FEM) by means of contour integrals and the extended finite element method (XFEM) implemented in the Abaqus 2018 software. Hsin Jen Hoh et al. [12] developed a methodology to model and calculate the stress intensity factors and weld toe magnification factors for semi-elliptical surface cracks in a circumferentially welded pipe. Yang Liu et al. [13] investigated the structural fatigue crack growth behavior of observed transverse surface cracks in rail steel and thermite weld subjected to in-plane and out-of-plane loading. Stress intensity factor (SIF) solutions were derived from a finite element model analysis for transverse surface cracks located at five different crack initiation locations in the cross-section of rail steel and thermite weld. Bing Xu et al. [14] established a three-dimensional mathematical model, including an elastic rock mechanics equation and a material flow continuity equation, to simulate horizontal fracture propagation in shallow reservoirs. Daobing Wang et al. [15]

investigated the diversion mechanisms of an opened natural fracture intersected by a hydro-fracture in naturally fractured reservoirs by means of the extended finite element method. Xuekun Lu et al. [16] prepared compact tension test pieces from a tusk of African elephant ivory. Crack-tip strain mapping and crack opening displacement (COD) measurements through the digital image correlation (DIC) technique were made under incremental loading and unloading of cracks for hydrated and dry dentin of different orientations. Mohammad Reza Khosravani et al. [17] used nylon and glass fibers as matrix and fiber reinforced materials, steel semi-circular bending (SCB) specimens were fabricated and examined for comparison and a numerical simulation was performed to study fracture load of SCB steel specimens and verify experimental observations.

Based on the current research situation at home and abroad, it can be seen that the sealing performance of seal rings is a popular research topic. However, these studies mainly focus on the normal sealing mechanisms of subsea connectors, flange joints, steam turbines, and mandrel hangers [2–6]. There is no research on crack's influence on the sealing performance of seal rings. As for stress intensity factors at crack tip, although there are some studies on cylinders, pipes, valve body, or other simple structures [7–17], there is no research on crack-tip stress intensity factors of subsea connector sealing rings. In conclusion, there are two main shortages in previous research: (a) cracks' influence on the sealing performance of seal rings was not considered; (b) the crack-tip stress intensity factors study only focused on simple structures.

In this study, based on the theory of fracture mechanics, the main parameters that may influence the stress intensity factor at crack tip are discussed. The overall sealing requirements of subsea connector metal sealing rings were conducted under the guidance of sealing criteria such as GB 150-2010 [18]. The three-dimensional model of the subsea connector was established using SolidWorks, and it was later imported in ABAQUS. Using a finite element analysis, the crack area of the sealing ring was found based on the results of maximum equivalent stress and maximum equivalent plastic strain of crack-free sealing rings. Meanwhile, the research goals of this study are: (a) the influence of different crack parameters, such as crack depths, crack positions and crack angles, on the sealing performance in preload state and operating state; (b) and the influence of different crack parameters on stress intensity factors at crack tip. The research results are of some reference value for predicting the fatigue life of sealing ring cracks, thus improving the safety and reliability of subsea connectors in practical engineering applications.

2. Basic Theory of Linear Elastic Fracture Mechanics

Depending on the mechanics of cracks, crack expansion can be divided into three types: opening mode cracks, sliding mode cracks, and tearing mode cracks. As shown in the Figure 1.

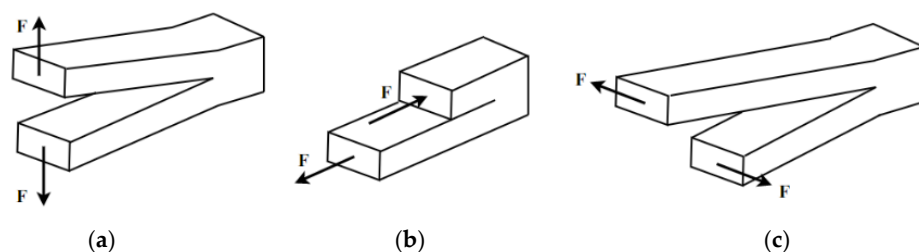


Figure 1. Three different types of cracks. (a) Opening mode cracks, (b) sliding mode cracks, (c) and tearing mode cracks.

Figure 1a shows opening mode cracks (mode I): the pull stress is perpendicular to the crack surface, causing it to open. Figure 1b shows sliding mode cracks (mode II): the shear stress is parallel to the crack surface and perpendicular to the crack front, causing the crack to slide open relatively within the plane. Figure 1c shows tearing mode cracks (mode III): the shear stress is parallel to the crack surface and the crack front, making

the crack relatively staggered. Inside the engineering components, mode I cracks are the most dangerous ones. In order to be safe, the actual composite cracks are often treated as mode I cracks.

2.1. Stress Intensity Factor K_I

In the 1960s, American scientist G.R. Irwin first proposed the concept of stress intensity factors. He saw the crack problem as an oval gap with a very large ratio of long axis to short axis. The problem was solved by elastic mechanics [19–21].

Consider a large plate containing a mode I crack, its force state is shown in Figure 2. The approximate solution of stress components, strain components, and displacement components at any point A(r, θ) near the crack tip can be gained, the expressions are as follows:

$$\left. \begin{aligned} \sigma_x &= \frac{\sigma\sqrt{\pi a}}{\sqrt{2\pi r}} \cos \frac{\theta}{2} \left(1 - \sin \frac{\theta}{2} \sin \frac{3\theta}{2} \right) \\ \sigma_y &= \frac{\sigma\sqrt{\pi a}}{\sqrt{2\pi r}} \cos \frac{\theta}{2} \left(1 + \sin \frac{\theta}{2} \sin \frac{3\theta}{2} \right) \\ \tau_{xy} &= \frac{\sigma\sqrt{\pi a}}{\sqrt{2\pi r}} \sin \frac{\theta}{2} \cos \frac{\theta}{2} \cos \frac{3\theta}{2} \end{aligned} \right\} \quad (1)$$

$$\left. \begin{aligned} \varepsilon_x &= \frac{1}{2G(1+\mu')} \frac{\sigma\sqrt{\pi a}}{\sqrt{2\pi r}} \cos \theta \left[(1-\mu') - (1+\mu') \sin \frac{\theta}{2} \cos \frac{3\theta}{2} \right] \\ \varepsilon_y &= \frac{1}{2G(1+\mu')} \frac{\sigma\sqrt{\pi a}}{\sqrt{2\pi r}} \cos \theta \left[(1-\mu') + (1+\mu') \sin \frac{\theta}{2} \cos \frac{3\theta}{2} \right] \\ \gamma_{xy} &= \frac{1}{2G} \frac{\sigma\sqrt{\pi a}}{\sqrt{2\pi r}} \sin \frac{\theta}{2} \cos \frac{\theta}{2} \cos \frac{3\theta}{2} \end{aligned} \right\} \quad (2)$$

$$\left. \begin{aligned} u &= \frac{\sigma\sqrt{\pi a}}{G(1+\mu')} \sqrt{\frac{r}{2\pi}} \cos \frac{\theta}{2} \left[(1-\mu') + (1+\mu') \sin^2 \frac{\theta}{2} \right] \\ v &= \frac{\sigma\sqrt{\pi a}}{G(1+\mu')} \sqrt{\frac{r}{2\pi}} \sin \frac{\theta}{2} \left[2 - (1+\mu') \cos \frac{\theta}{2} \right] \end{aligned} \right\} \quad (3)$$

where μ is Poisson's ratio, $\mu' = \frac{\mu}{1-\mu}$; G is the shear modulus, $G = \frac{E}{2(1+\mu)}$; E is Yang's modulus; a is half of the crack length; and σ is the external force of the crack body.

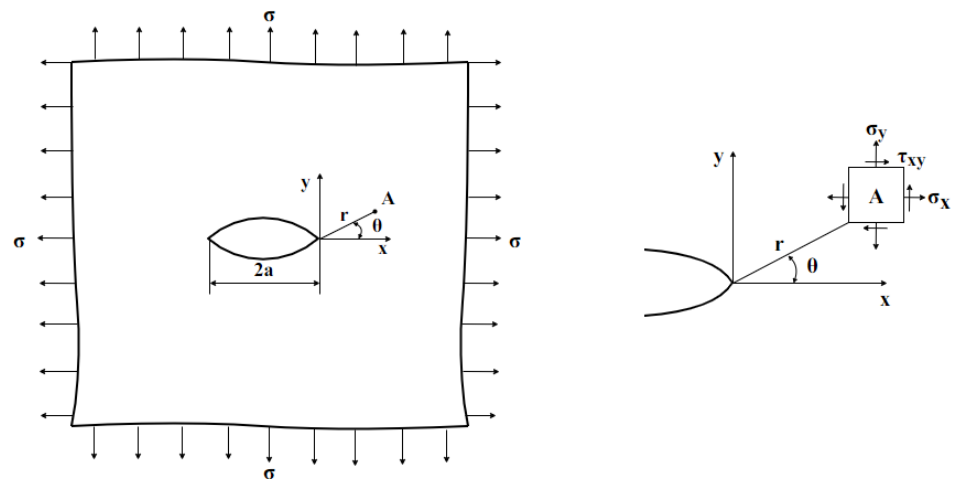


Figure 2. Mode I penetrable crack in an infinite plate.

As can be seen in Equations (1)–(3), every stress component, strain component, and displacement component is related to $\sigma\sqrt{\pi a}$; therefore:

$$K_I = \sigma\sqrt{\pi a} \quad (4)$$

K_I is the stress intensity factor, the subscript I indicates that the crack type is mode I.

When r is much less than a , the error between the approximate solution and the exact solution is very small, the above approximate solution results can be applied in engineering. Equations (1) and (2) can be summed up as:

$$\sigma_{ij} = K_I f_{ij}(r, \theta), \varepsilon_{ij} = K_I f'_{ij}(r, \theta) \quad (5)$$

As can be seen from Equation (5), the stress and strain at any point near the crack tip is proportional to K_I . Therefore, K_I is a physical quantity that can characterize the strength of the crack-tip stress field and strain field, so it is called the stress intensity factor, whose common unit is $\text{MPa} \cdot \text{m}^{1/2}$.

As can be seen from Equation (4), the stress intensity factor K_I is a function of external load σ and crack length a , independent of material characteristics. Similar results can be obtained for cracks in other shapes such as single edge notch tension and three-point bending, so the general expression of the stress intensity factor is:

$$K_I = Y\sigma\sqrt{\pi a} \quad (6)$$

where Y is a function of parameters such as sample geometry, crack configuration, etc., the value of which can be found in *Stress Intensity Factor and Limit Load Handbook* [22]. For the crack form and force condition in Figure 2, Y is 1. Similar results are available for stress intensity factors K_{II} and K_{III} of mode II and mode III cracks.

2.2. Fracture Toughness K_{IC}

As can be seen from Equation (4), when the crack size is determined, K_I increases accordingly as the external load increases. When K_I increases to a certain extent, the crack crazes. If K_I continues to increase, the crack enters the expansion phase, resulting in an unstable fracture. Experiments show that for different materials, the minimum K_I value for unstable fracture varies is called critical stress intensity factor K_{IC} . K_{IC} reflects the toughness of the material against crack fractures, so it is also known as the material's fracture toughness indicator.

Note the difference between K_I and K_{IC} . Stress intensity factor K_I is independent of material performance and depends only on the crack form, the external load, and the crack size. Fracture toughness K_{IC} is the performance parameter of the material, regardless of external load and crack length, etc.

The value of K_{IC} must be measured experimentally according to the standard method, and there are K_{IC} test standards at home and abroad such as BS5447-1997 by the British Standards Institution ASTM-E399 by the American Society for Testing and Materials, and Chinese National Standard GB4161-84. The standards specify the preparation of samples, test equipment, test steps, test results processing and so on. Ganesh Puppala [23] evaluated the fracture toughness and Charpy impact toughness of Inconel 625 structures fabricated by laser-based additive manufacturing, the fracture toughness of Inconel 625 that they estimated was $66 \text{ MPa} \cdot \text{m}^{1/2}$ which provides some references in this research.

2.3. Brittle Fracture Criterion of Linear Elastic Fracture Theory

If the K_I of crack body reaches the K_{IC} of the material, the crack body reaches the critical state of fracture. Therefore, the critical condition for the crack to expand with brittle instability is:

$$K_I \geq K_{IC} \quad (7)$$

As for K_{II} and K_{III} , similar critical conditions exist. The brittle fracture criterion of linear elastic fracture theory can be summed up as:

$$K \geq K_C \quad (8)$$

Linear elastic fracture theory has been well-applied in practical engineering, but it also has an applicable range. As mentioned earlier, Equations (1)–(3) are only applicable

to areas near the crack tip. Only when the plastic zone in this area is less than 1/10 of the crack length a can linear elastic fracture mechanics be used, this range is called small-scale yielding condition.

2.4. Theoretical Calculation of Stress Intensity Factors of Subsea Connector Sealing Rings

The *Stress Intensity Factor and Limit Load Handbook* covers only simple, basic structures such as the pipe cylinder structure shown in Figure 3a and does not contain the shape factor values of all components, such as subsea connectors' sealing rings studied in this study, as shown in Figure 3b. The crack-tip stress intensity factors can only be calculated using the formula when the sealing ring is simplified as a pipe cylinder. In this study, the simplified theoretical model is established to obtain the crack-tip stress intensity factors of sealing rings, as can be seen in Figure 3c and Table 1.

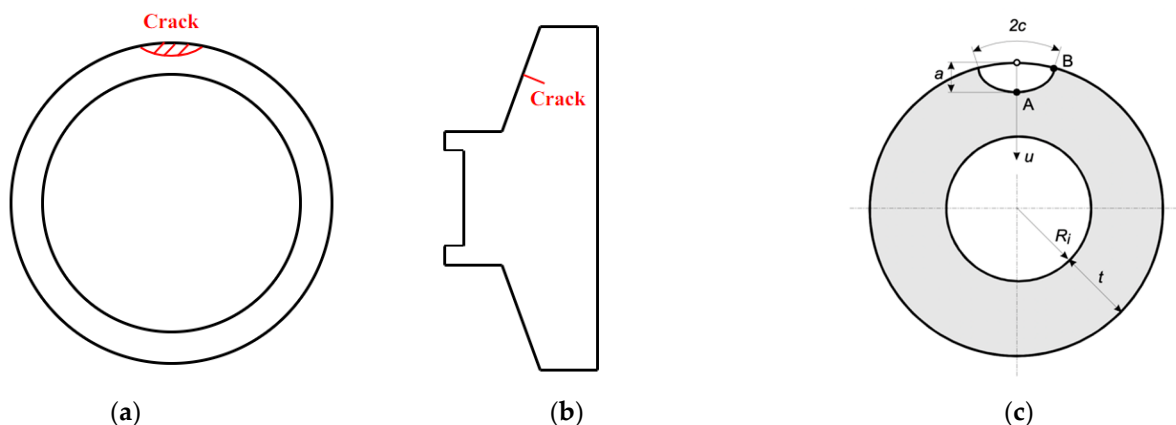


Figure 3. Cracks in different structures. (a) A crack in a pipe cylinder, (b) a crack in a sealing ring, (c) and simplified sealing ring structure.

Table 1. The basic dimensions of sealing ring.

R_i (mm)	t (mm)	a (mm)	c (mm)
77	20	2	5

According to the *Stress Intensity Factor and Limit Load Handbook*, the stress intensity factor K_I in the deepest point of the crack (A) is provided by:

$$K_I = \sqrt{\pi a} \left(\sum_{i=0}^3 \sigma_i f_i \left(\frac{a}{t}, \frac{2c}{a}, \frac{R_i}{t} \right) + \sigma_{bg} f_{bg} \left(\frac{a}{t}, \frac{2c}{a}, \frac{R_i}{t} \right) \right) \quad (9)$$

where σ_i ($i = 0$ to 3) are stress components which define the axisymmetric stress state σ according to:

$$\sigma = \sigma(u) = \sum_{i=0}^3 \sigma_i \left(\frac{u}{a} \right)^i \text{ for } 0 \leq u \leq a \quad (10)$$

where σ_{bg} is the global bending stress, and σ_i is determined by fitting σ to Equation (10). The coordinate u is defined in Figure 3. f_i ($i = 0$ to 3) and f_{bg} are geometry functions which are provided in the *Stress Intensity Factor and Limit Load Handbook* for the deepest point of the crack (A): $f_0 = 0.905$, $f_1 = 0.56$, $f_2 = 0.425$, $f_3 = 0.347$, $f_{bg} = 0.885$. Since the preload force can also be simplified as a normal stress $\sigma = 34.5$ MPa, when the crack surface is vertical to the sealing plane and its depth is 2 mm, the theoretical stress factor $K_I = 43.94$ MPa·mm^{1/2}. According to Section 2.2, the fracture toughness of Inconel 625 $K_{IC} = 66$ MPa·m^{1/2} = 2087.10 MPa·mm^{1/2}, which is way larger than the theoretical stress factor K_I . Thus, the crack body does not reach the critical state of fracture and the sealing ring structure is relatively safe.

According to the Forman equation:

$$\frac{da}{dN} = \frac{C(\Delta K)^m}{(1-R)K_{IC} - \Delta K} \quad (11)$$

where da/dN is the crack growth rate, C and m are material parameters, ΔK is the stress intensity range. Once the stress intensity factors of subsea connector sealing rings' cracks are obtained, the propagation life of fatigue cracks can be predicted when its experimental data are also gained. Thus, the theoretical calculation of stress intensity factors in this section can provide certain reference value for predicting the fatigue life of Inconel 625 sealing ring cracks in future studies.

3. Simulation of Subsea Connector Crack-Free Sealing Rings

3.1. Finite Element Modeling

A subsea connector consists of a connecting cover, a drive ring, claws, a center base, an upper hub, a metal sealing ring, and a lower hub. The actual and model diagrams are shown in Figure 4a,b. In ABAQUS, a three-dimensional model of a subsea connector is established. In order to simplify the calculation, its core sealing components: the sealing ring, the upper hub, and lower hub are the main research objects. The sealing ring is a key subject, its structure size is shown in Figure 4c and Table 2.

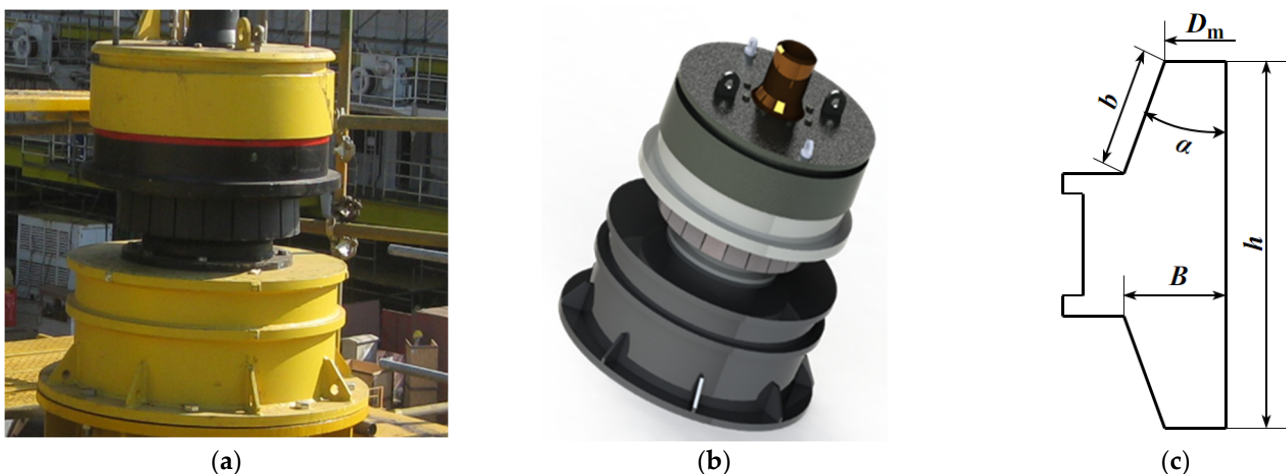


Figure 4. Cracks in different structures. (a) Subsea connector, (b) SolidWorks model of subsea connector, (c) and structure dimension of sealing ring.

Table 2. The basic dimensions of sealing ring.

D_m (mm)	h (mm)	B (mm)	b (mm)	α (°)
164	76	20	28	23

Due to the special working environment of metal sealing rings, it is necessary to choose metal materials with high yield strength, good plasticity, and certain corrosion resistance in order to meet the working requirements. The specific material parameters for each component are shown in Table 3 [24].

Table 3. Material parameters of different parts.

Part	Upper Hub	Lower Hub	Sealing Ring
Material	ASTM A182 F22 (75ksi)	ASTM A182 F22 (75ksi)	Inconel 625
Elastic modulus/(MPa)	210,000	210,000	200,000
Passion's ratio	0.3	0.3	0.3
Yield strength/(MPa)	310	310	345

The contact problem is a complex non-linear problem, and the interaction between the contact surfaces includes the normal and tangential effects. For the normal effect, the default relationship between contact pressure and clearance in ABAQUS is hard contact, i.e., constraints are applied only when there is zero gap between two surfaces. For the tangential effect, the commonly used friction models in ABAQUS are the coulomb friction model and penalty function model. The penalty function model allows the contact surfaces to have elastic slip, which was used in this study to define the friction factor as 0.15 [25]. The metal sealing ring has two pairs of contact, that is, the contact between upper and lower hubs' inside and the outside of the sealing ring.

The mesh of the sealing ring contact surface is divided into 10 parts, 0.5 mm each, with 11 grid nodes shown in Figure 5. In operating state, the subsea connector is in direct contact with the high-pressure oil inside the pipeline and external seawater, so the connector is under the load of internal oil pressure and external seawater pressure. Based on the actual situation, the internal oil pressure p is 34.5 MPa and is vertical to the inner surfaces of upper hub, lower hub, and sealing ring. The external seawater pressure p' is 15 MPa and is vertical to the outer surface of lower hub. The preload force applied to the upper hub end surface F is 445 kN. Moreover, the vertical movement of the lower hub end surface is constrained. The load distribution is shown in Figure 6 and Table 4.

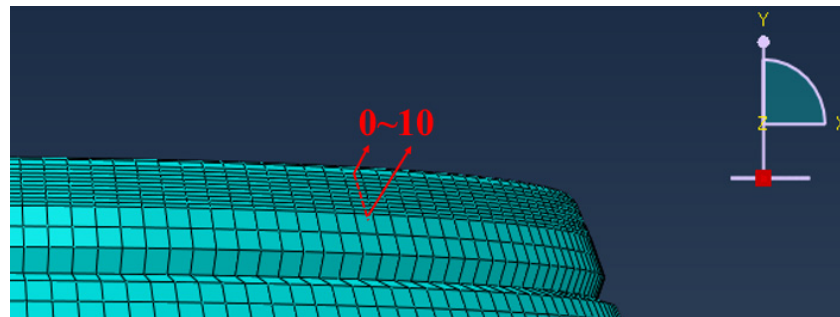


Figure 5. Meshing node of the sealing surface.

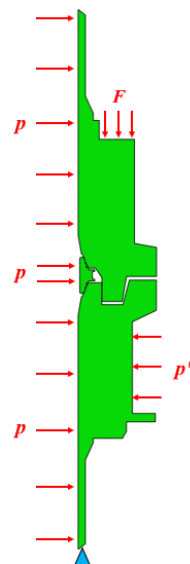


Figure 6. Loading distribution of the subsea connector.

Table 4. Input loading parameters.

Internal Oil Pressure p	External Seawater Pressure p'	Preload Force F
34.5 MPa	15 MPa	445 kN

3.2. Finite Element Analysis Results

The distribution of Mises stress and plastic deformation in the operating state of the crack-free model are shown in Figure 7a,b, respectively. The maximum Mises stress value is 497.8 MPa and the maximum plastic deformation value is 0.016, both at the sealing ring's contact surface. The maximum Mises stress value is greater than the sealing ring's yield strength 345 MPa. It can be seen that due to the preload force and the media pressure, the sealing ring's contact surface has a certain degree of plastic deformation, providing a reference area for the subsequent preset cracks.

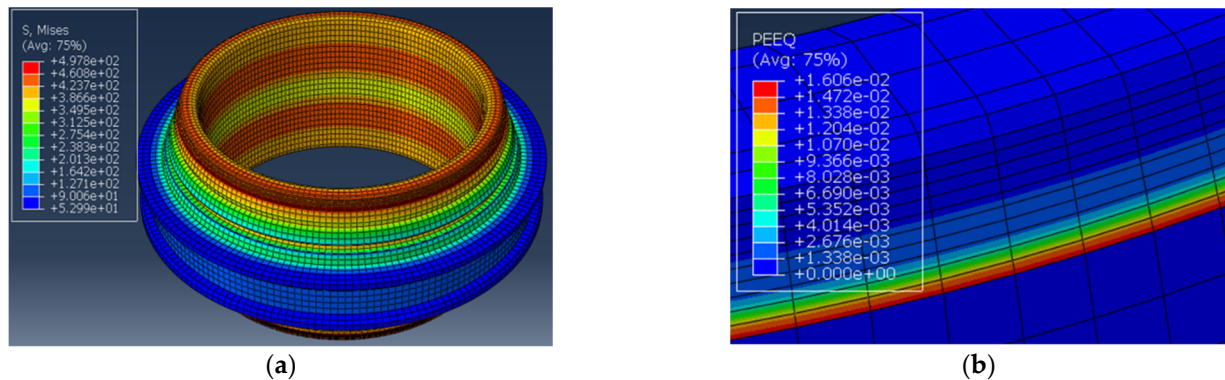


Figure 7. Finite element analysis results of sealing ring without a crack. (a) Mises distribution at operating state, (b) and plastic deformation distribution at operating state.

4. Calculation of Sealing Performance of Subsea Connectors' Metal Sealing Rings

4.1. Preset the Crack

The maximum Mises stress and the maximum plastic deformation of the crack-free sealing ring are both in the contact area between the sealing ring and the hubs, which is the key research area to determine the sealing performance. In order to study the effect of crack depths, crack positions, and crack angles on the sealing performance of subsea connectors, three-dimensional cracks with different depths, positions and angles are preset in the contact area. First, create a new rectangle shell part as the crack, then assemble it in the contact area of the sealing ring. In the interaction module of ABAQUS, create a new crack, define it as XFEM; select the sealing ring as the region where the crack occurs; pick the specific crack location and define its contact properties. Its specific parameters in the finite element program ABAQUS are shown in Figure 8.

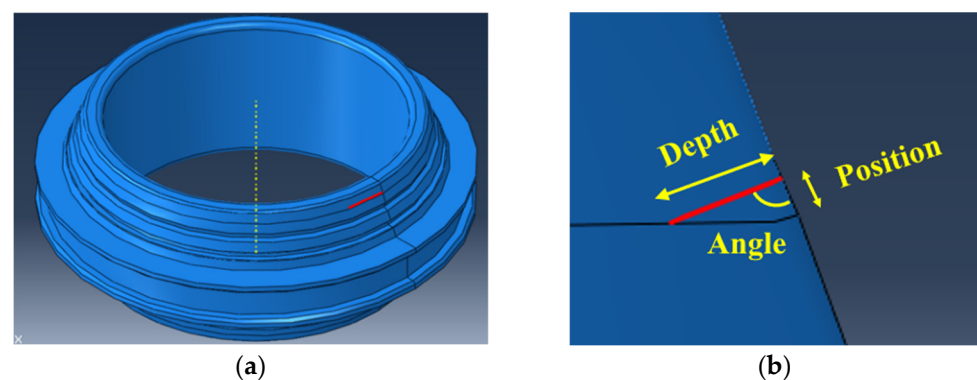


Figure 8. Preset the crack. (a) Preset the 3D crack, (b) parameters of the crack.

4.2. The Influence of Crack Depths on Sealing Performance

Because the subsea connectors' sealing rings are steel, the gasket factor m is 6.5 and the sealing pressure ratio y is 179.3 MPa based on GB 150-2010. In order for sealing rings to meet the sealing requirements, in the preload state, the contact stress q_0 must be greater

than the sealing pressure ratio y , and in operating state the contact stress q must be greater than the product of the gasket factor m and the internal pressure p . At the same time, BUCHTER HH [26] states that the total sealing width b should be at least 1.5 mm~2 mm in order to achieve a reliable metal sealing. Therefore, the following sealing requirements are established: $q_0 > y$, $q > mp$, $b \geq 2$ mm.

The crack depths include 0.5 mm, 1 mm, 2 mm, 3 mm, 4 mm, and 5 mm. Figure 9a,b are the contact stress of 11 nodes at the sealing surface with different crack depth in preload and operating states, respectively. Comparing to the sealing requirements, the effective sealing width of the preload state and operating state meets the requirements of wider than 2 mm. Moreover, the deeper the crack depth is, the wider the sealing width is. The 1 mm crack area's contact stress in the operating state is shown in Figures 10 and 11, which showcase the contrast between its initial state and operating state. As can be seen from Figures 10 and 11, in the early stages of cracks, the contact state with the hub has the following changes compared with the absence of cracks: the crack divides the sealing ring's contact surface into the upper and lower parts, and the pressure makes an offset occur between them. Then the upper part and the contact surface of upper hub form a separate contact, in which the maximum contact stress is reduced compared with the crack-free sealing ring, indicating that the crack causes load reassignment at the contact surface. The upward shift of the maximum contact stress causes the effective sealing width area to move up. Moreover, the deeper the crack is, the more obvious the offset between the upper and lower parts is, resulting in a more uniform contact between the upper part and upper hub, a smaller maximum contact stress value, and wider effective sealing width.

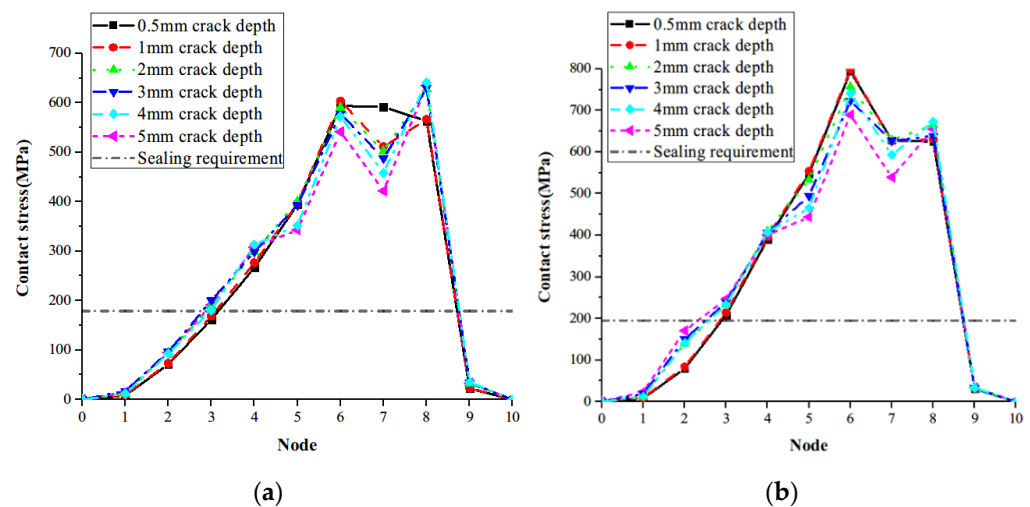


Figure 9. The contact stress distribution of sealing surface along the node at different crack depths. (a) Contact stress in preload state, (b) and contact stress in operating state.

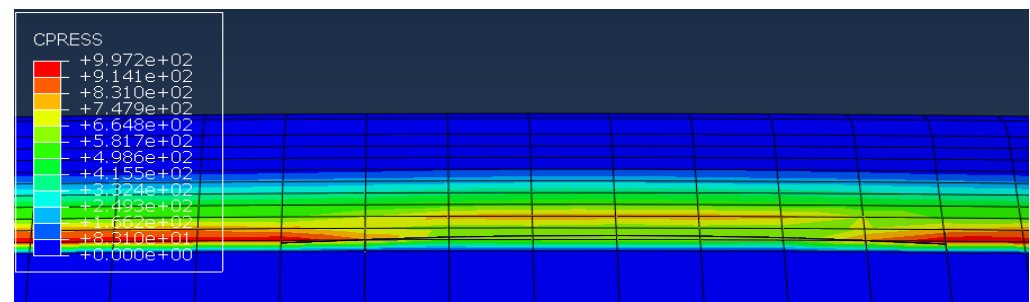


Figure 10. The contact stress of 1mm crack area in operating state.

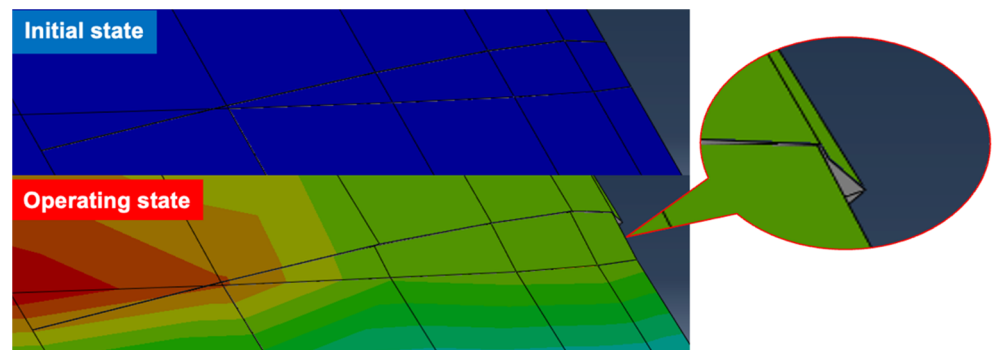


Figure 11. The contrast between initial state and operating state of 1 mm crack area.

4.3. The Influence of Crack Positions on Sealing Performance

The distance from the crack to node 10 is the crack position, whose values are 0 mm, 0.5 mm, 1 mm, and 1.5 mm. Figure 12a,b are the contact stress of 11 nodes at the sealing surface in different crack positions in preload and operating states, respectively. Comparing to the sealing requirements, the effective sealing width of preload state and operating state meets the requirements of wider than 2 mm. According to Figure 12, when the crack position is 0.5 mm, contact stress and sealing width are the smallest and the easiest to cause sealing failure. As can be seen from Figure 13, the maximum Mises stress center of the crack-free sealing ring is 0.5 mm from node 10, and cracks are most likely to occur there, which verifies the rationality of the calculation results.

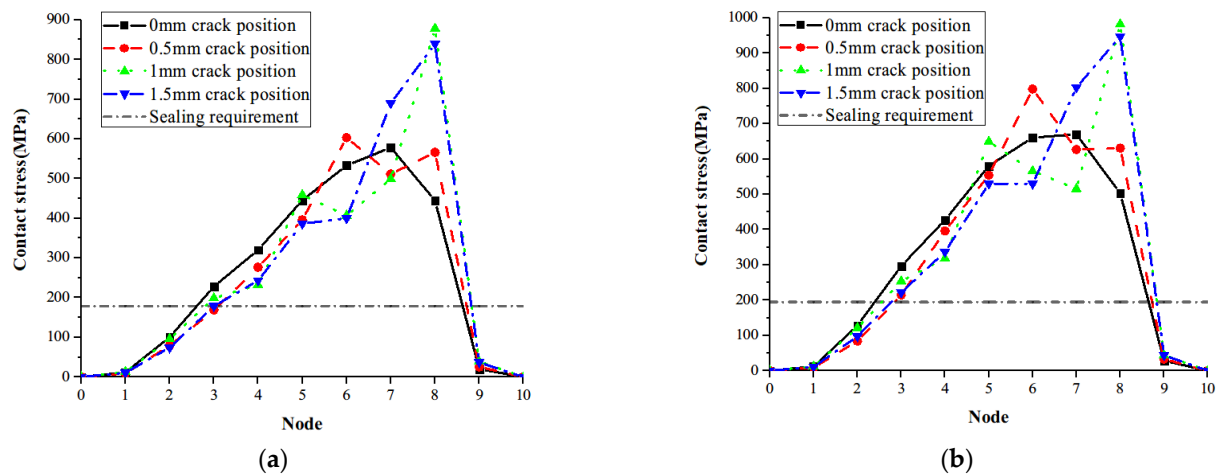


Figure 12. The contact stress distribution of sealing surface along the node at different crack positions. (a) Contact stress in preload state (b) and contact stress in operating state.

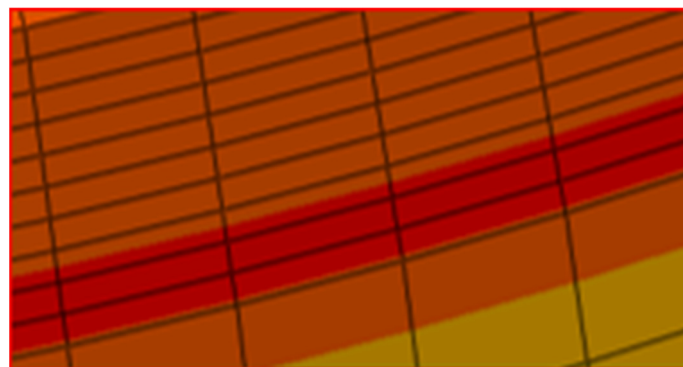


Figure 13. Maximum Mises stress position of sealing ring without a crack.

4.4. The Influence of Crack Angles on Sealing Performance

The angle between the crack and the contact surface is the crack angle, whose values are 30° , 60° , 90° , 120° , and 150° . Figure 14a,b are the contact stress of 11 nodes at the sealing surface at different crack angles in preload and operating states, respectively. Comparing to the sealing requirements, the effective sealing width of preload state and operating state meets the requirements of wider than 2 mm. According to Figure 14, when the crack angle is 90° , the contact stress and sealing width are minimal, the sealing ring is the most prone to fail. As can be seen from Figure 15, the maximum Mises stress range is roughly vertical to the contact surface of the crack-free sealing ring, and cracks are most likely to be produced at this angle, which explains the rationality of the calculation results.

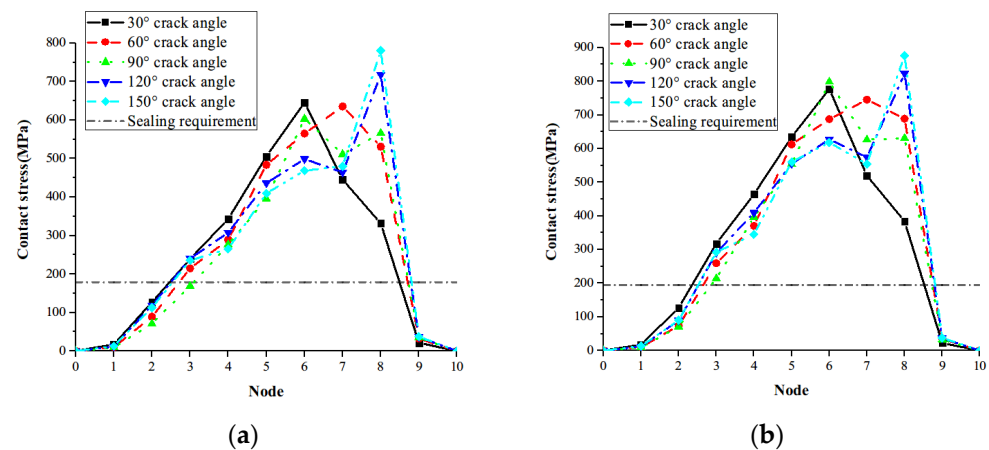


Figure 14. The contact stress distribution of sealing surface along the node at different crack angles. (a) Contact stress in preload state (b) and contact stress in operating state.

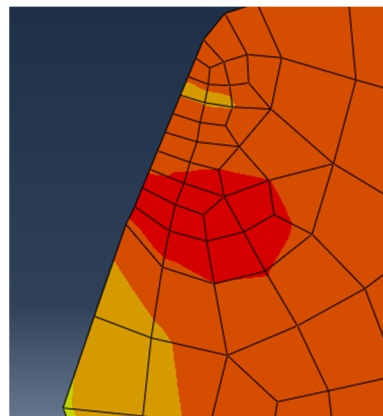


Figure 15. Maximum Mises stress angle of sealing ring without a crack.

5. Calculation of Stress Intensity Factors at Crack Tip of Subsea Connectors' Metal Sealing Rings

5.1. The Influence of Crack Depths on Stress Intensity Factors

Because each node at the crack tip is subjected to different forces, the K_I values of different nodes at crack tip are different. There are seven nodes along the front edge of the sealing ring crack, resulting in seven K_I values. Figure 16 describes the trend of K_I values with different crack depths. The relationship between K_I and crack depths at different nodes are shown in Figure 17. According to Figures 16 and 17, crack-tip stress intensity factor basically increases with the increase in crack depth. It can be seen that the deeper the crack depth is, the more prone the sealing ring is to brittle fracture. This is consistent with the theory of fracture mechanics and proves the correctness of finite element simulation results.

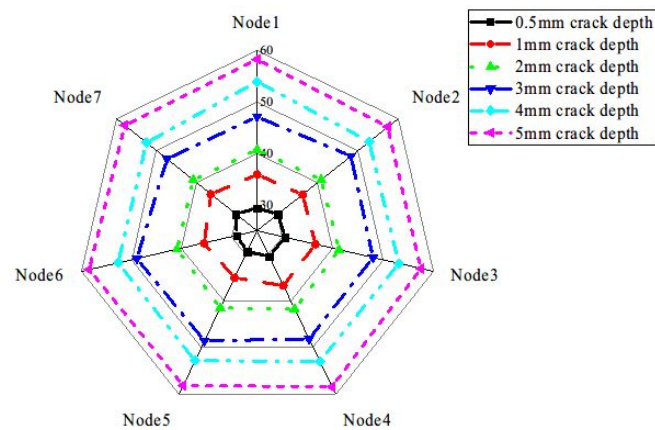


Figure 16. Stress intensity factors of different crack depths.

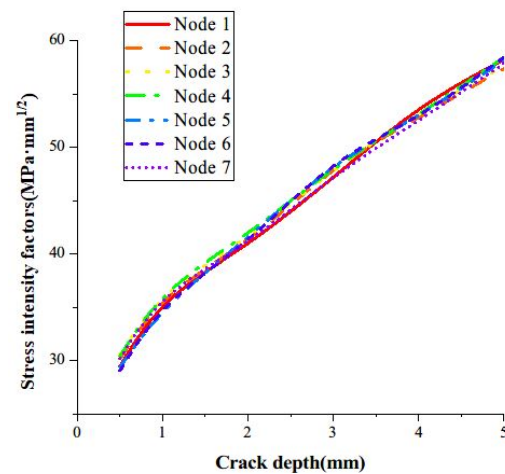


Figure 17. Stress intensity factors of different nodes.

5.2. The Influence of Crack Positions on Stress Intensity Factors

Figure 18 shows the trend of K_I values in different crack positions. The relationship between K_I and crack positions at different nodes are shown in Figure 19. According to Figures 18 and 19, crack-tip stress strength factors are relatively close. When the crack position is 1 mm from node 10, stress intensity factors of node 1~node 7 are the largest, the sealing ring is the easiest to fracture. Moreover, the crack position on the sealing performance results has a 0.5 mm difference.

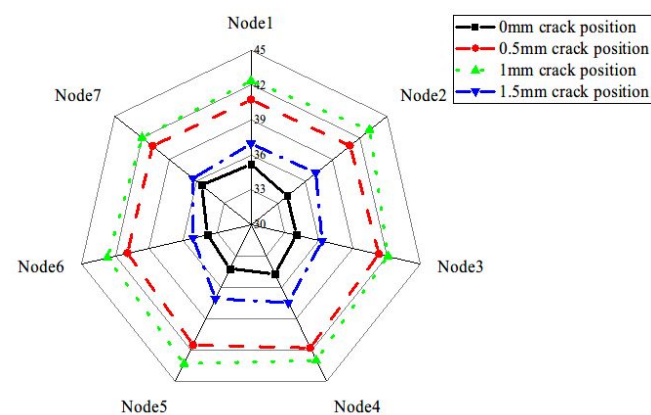


Figure 18. Stress intensity factors of different crack positions.

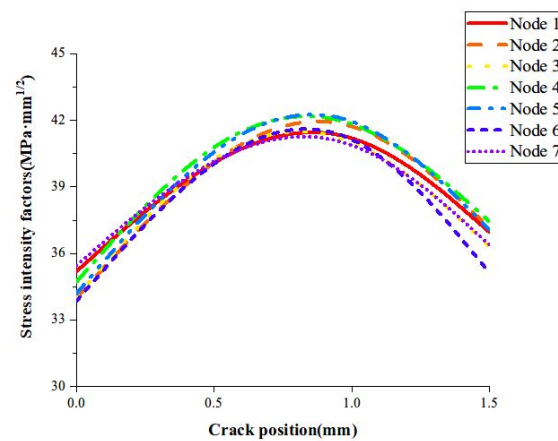


Figure 19. Stress intensity factors of different nodes.

5.3. The Influence of Crack Angles on Stress Intensity Factors

Figure 20 shows the trend of K_I values at different crack angles. The relationship between K_I and crack angles at different nodes are shown in Figure 21. According to Figures 20 and 21, crack-tip stress intensity factors are relatively close, but when the crack angle is 90° , the crack-tip stress intensity factors are the largest, the sealing ring is more prone to brittle fracture. Moreover, it is consistent with the crack angle on the sealing performance results.

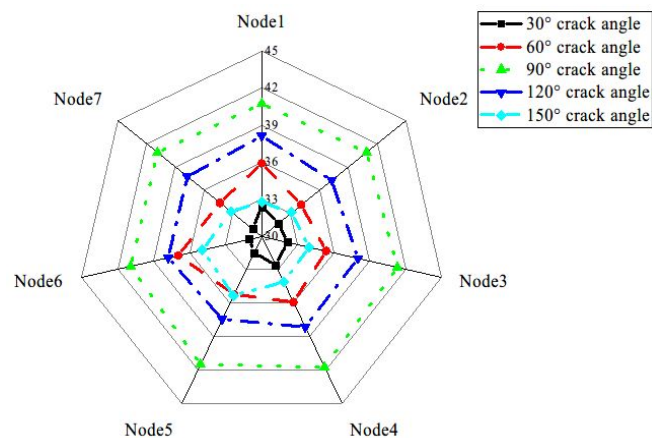


Figure 20. Stress intensity factors of different crack angles.

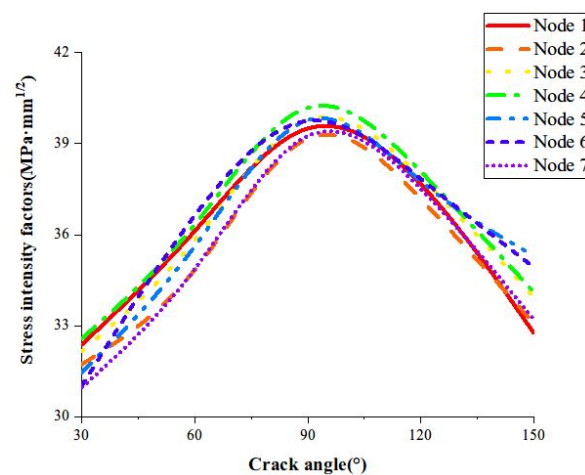


Figure 21. Stress intensity factors of different nodes.

6. Conclusions

The sealing of subsea connectors' metal sealing rings is the key feature of a subsea production system and is quite important. This study mainly analyzed the sealing performance and stress intensity factors of sealing ring when a crack occurs. At first, the finite element analysis of the subsea connectors' crack-free metal sealing rings was simulated using ABAQUS. Then, by presetting the XFEM crack and changing its parameters, the effect of different crack depths, positions, and angles on sealing performance and crack-tip stress intensity factors was studied in this study. According to the research results, the following conclusions can be drawn:

- (1) When it is at work, the maximum Mises stress value on the contact surface of the subsea connectors' metal sealing rings is greater than the material yield strength. In other words, it is a stress concentration area and is in a plastic deformation state, which is prone to cracking;
- (2) The deeper the crack depth is, the wider the sealing width is. When the crack position is 0.5 mm from node 10 and the crack angle is 90° , contact stress and sealing width are the smallest, and the most likely to cause sealing failure. The maximum Mises stress range of the crack-free sealing ring is 0.5 mm from node 10 and vertical to the contact surface, causing cracks to most likely emerge in this area;
- (3) The deeper the crack depth is, the greater the crack-tip stress intensity factor is and the more prone the sealing ring is to brittle fracture, which is consistent with the fracture mechanics theory. Stress intensity factors are the largest when crack position is 1 mm from node 10, which is a 0.5 mm difference from the sealing performance results. When crack angle is 90° , the crack-tip stress intensity factors are the largest and the sealing ring is more prone to brittle fracture, it is the same as the sealing performance results;
- (4) In practical application, future research should consider the potential effects of multiple cracks and their interaction and influence on the overall performance of the sealing ring, as well as the relationship between corrosion, hydrogen embrittlement, and cracks.

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