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Sealing Contact Transient Thermal-Structural Coupling Analysis of the Subsea Connector

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Abstract: Taking the subsea collect connector as an example, the sealing characteristics of the sealing structure of the subsea connector under transient thermal–structural coupling were studied. Based on Airy’s thermal stress function, the mathematical model was established, the complex stress function was used to solve the problem, and a three-dimensional transient thermal stress model of the core sealing parts was obtained. The transient thermal–structural coupling stress model of the core seal was obtained by linear superposition principle. The transient temperature field of the subsea collet connector under different working conditions was analyzed. It was found that the larger the temperature difference between the components was, the greater the difference of expansion rate was, and the greater the impact on the sealing performance of LSG was in the process of temperature variation from transient to steady distribution. Numerical simulations of various working conditions under the transient temperature field were carried out. The results showed that sudden change of the temperature and oil–gas pressure will bring about large fluctuations of the maximum contact stress and equivalent stress of the seal, which was easy to fatigue wear, and thus affect the reliability of the sealing performance. Finally, the experiment proves that the sealing ring can maintain sealing performance under the conditions of temperature-cyclic loading. The coupled mathematical model proposed in this paper could be used for the transient thermal–structural coupling theoretical analysis of similar subsea equipment.

Keywords: subsea connector; transient thermal–structural coupling; sealing characteristics



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1. Introduction

The subsea production system mainly consists of the subsea Christmas trees, subsea wellhead, subsea connector, subsea manifold and jumper system [1]. Generally, the subsea connector is used to connect different equipment, of which sealing contact performance is critical to the safety of the subsea production system [2]. The subsea connector works for a long time in a harsh environment with high internal temperatures and pressure and low external temperature and high pressure. The temperature characteristics of the sealing structure with the high internal temperatures and low external temperatures are essential in ensuring the connector’s reliable operation. Therefore, it is necessary to study the sealing performance of the subsea connector under thermal–structural coupling.

In recent years, an increasing number of scholars have begun to study contact problems under structural load or temperature load. Using finite element numerical simulation and experiments, Abid et al. [3–6] studied the sealing performance of the hub connector under different combinations of temperature loads, internal pressure and external loads. They pointed out that the change of the loads would lead to fatigue of the sealing gasket, especially at high temperature. Zhou et al. [7] studied the transient temperature field variation

of the hub by establishing a heat transfer model considering the nonlinearity of the gasket and then analyzed the effect of temperature load on bolt and gasket loads. Avanzini [8] proposed a fatigue endurance prediction method by combining the equivalent strain range at the critical sealing point and the material endurance curve to calculate the fatigue damage of seals under an ultra-high-pressure environment. Omiya et al. [9] analyzed the thermal stress in the bolt hub under internal pressure and temperature loads using the elastic theory and numerical method in 2009. The results were verified by experiments, and the idea that sealing performance improves with increasing temperature was presented. In 2014, Omiya et al. [10] used the elastic–plastic finite element method to obtain the stress and contact stress distributions of the pipe hub with different nominal diameters under the working conditions of temperature cycle and internal pressure rise. The results indicated that temperature must be considered when designing hub connections with large nominal diameters. Sawa [11] investigated the relationship between the thermal expansion coefficient and hub connect sealing through a series of experiments, indicating that when the thermal expansion coefficient of the sealing gasket is greater than that of the hub, the sealing can be maintained for a long time under high-temperature conditions. Luo et al. [12] proposed a thermal fatigue experimental method based on accelerated endurance experimental theory to analyze the causes of avionics connector performance degradation. Abdullah et al. [13] attempted to use an axisymmetric finite element model to analyze contact pressure and thermal stress during a single-disc frictional clutch engagement. The results showed that the thermal deformation caused by local heat on the contact surface significantly affects the contact pressure distribution. Xue et al. [14] proposed a method to calculate the leakage rate of bolt hub connections under high-temperature conditions based on the relationship between leakage rate and service time. The variation law of the bolt hub leakage rate under high temperatures was obtained. The results showed that the leakage rate increased gradually with time. Zhang et al. [15] adopted the finite element method to analyze the stress distribution of hub connection under the combined loads of internal pressure, bending moment and temperature. The results showed that the higher the temperature, the smaller the allowable bending to ensure the sealing performance. Barsoum et al. [16] analyzed the interaction between thermal load and mechanical load on the hub structure of a service port by establishing a coupling thermodynamic non-linear model. The results showed that both yield and leakage might occur under high temperatures, which may give rise to the damage of the service port structure. Tang et al. [17] analyzed the sealing performance of flexible pipe fittings by establishing the hydraulic–thermal finite element model. The results showed that the appropriate high temperature could improve the sealing ability. Chen et al. [18] reconstructed the contact fractal model of the dry gas seal friction interface by combining contact mechanics and probability theory, analyzed the influencing factors of the contact, and obtained the variation law of thermal normal contact stiffness. Using the energy conservation principle, Li et al. [19] established the finite element equations for the stress field and temperature field of the sealing structure of the subsea wellhead connector to carry out thermal–structural coupling analysis. The results showed that sealing could be maintained by reducing the pre-compression when the temperature is excessively high. Wang et al. [20] attempted to analyze the contact finger seal’s leakage, heat transfer and thermal deformation properties by establishing a modified fluid dynamics and heat transfer model for porous media, which was verified by the thermal stress module in ANSYS Workbench. Zhang et al. [21,22] conducted the failure analysis of mechanical seal by mechanical–thermal sequential coupling method and studied the friction behavior and thermal effect of mechanical sealing gasket. Zeng et al. [23] proposed a sealing reliability evaluation method for subsea pipeline connector under thermal–structural coupling, which could improve the efficiency in an effective way. Liu et al. [24] studied the temperature field and thermal error caused by the contact between the ball screw and nut by using the finite difference method, and carried out a temperature and thermal error experiment of the ball screw feeding system, which shows that this method plays a positive role in accurately obtaining the temperature field of the feeding system. Zhang et al. [25] determined the

tooth load distribution and surface contact stress by using the Hertz contact theory, and proposed a method of combining the surface contact model with the finite element model, then obtained the contact characteristics of disposable harmonic drives under full load, which proved that the contact conditions of the gear can realize short-time transmission under full load. Song et al. [26] used the computational fluid dynamics and the finite element method to analyze the thermal–structural coupling of the frictional contact behavior of the disc brake and optimized the diameter of the ventilation hole based on the coupling analysis. The optimized result reduced the thermal stress and deformation of the structure at high temperatures noticeably. Wen et al. [27] analyzed the coupling of the temperature and thermal stress of the disc brake under multiple continuous braking contact conditions by using ABAQUS software. Then the temperature distribution characteristics of the brake disc surface were obtained through experiments and were compared with the finite element simulation results. Gao et al. [28] studied an active heat transfer wall–plate grid sandwich panel structure. The heat transfer performance and thermal structure response were analyzed by numerical simulation method on the basis of considering the mechanical and thermal properties affected by temperature. Ma et al. [29] analyzed the thermal stress, structural stress and fatigue life of aluminum brazing structures with fin–plate–side bar specified in different standards, and evaluated and verified the fatigue life under coupling stress by using finite element simulation software.

As can be seen from the above literature review, previous research mainly studied the contact performance through experiments and the finite element method. Little research can establish an effective mathematical model for calculation and analysis of the variation of mechanical characteristics of the contact structure on the premise of fully considering the interaction of temperature and structure. Different from the research methods of the above scholars, this paper establishes a theoretical calculation model of three-dimensional thermal stress of the subsea connector sealing structure based on generalized Hooke's law and complex stress function. The thermal–structural coupling theoretical calculation model of the subsea connector sealing structure under the combined load of axial preload, internal oil–gas pressure and temperature is established, which is also verified by numerical simulation and experiment in the paper.

2. The Subsea Connector Structure

The subsea collet connector and clamp connector commonly usually use metal gaskets as the primary seal. This paper takes the subsea collet connector as an example for analysis, as shown in Figure 1. The core sealing structure of the subsea collet connector consists of a lenticular sealing gasket (LSG), female and male hub. The actuator ring is driven downward by a special installation tool to make the collet hold the female hub and male hub and apply preload to the sealing structure to complete the connection and sealing. The sealing structure of the subsea connector will generate cross-structural stress and thermal stress when subjected to an external temperature load. If the coupling stress is excessively large, the sealing structure will have an irreversible effect, especially on the sealing contact surface. Therefore, it is necessary to analyze its mechanics and sealing contact properties in detail under the action of thermal–structural coupling. The total height of the subsea connector is 1710 mm; the biggest diameter is of the support board in the bottom connector 980 mm; the biggest outside diameter of the hubs is 338 mm; the internal bore diameter of the hubs and gasket is 140 mm.

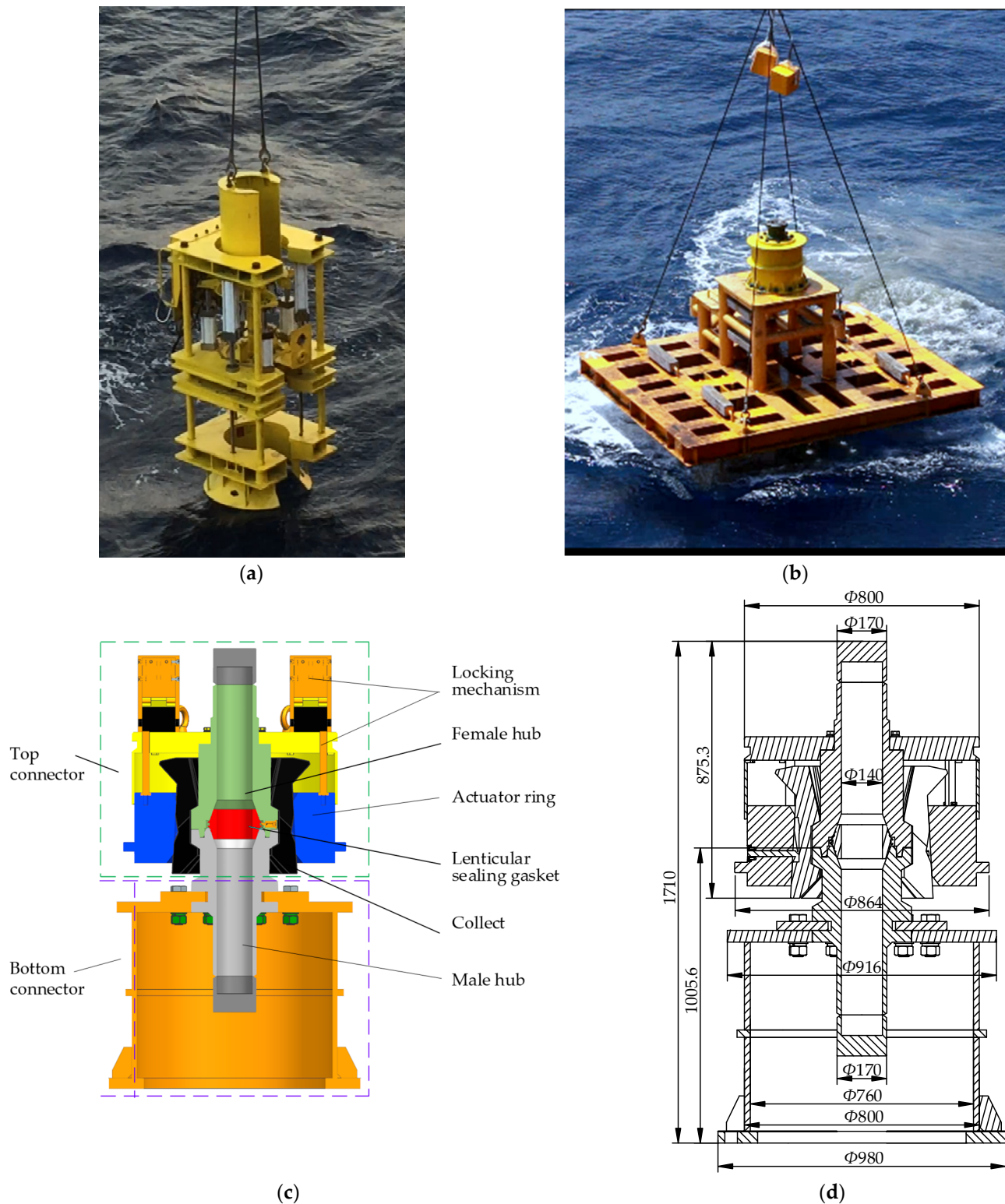


Figure 1. Subsea Connector. (a) Installation tool and top connector; (b) the base and bottom connector; (c) the key structure of the subsea connector; (d) drawing of the connector with the dimensions.

3. Thermal–Structural Coupling Mathematical Model of the Subsea Connector

Subsea connectors generate structural stress under the action of external load; under the action of temperature load, the connector components expand as the temperature rises and shrink as the temperature drops. Due to the constraint of each component, thermal stress is generated. Therefore, it is necessary to study the coupling of thermal stress and

structural stress [6]. The stress on LSG of the subsea connector can be divided into three parts: first, the thermal stress generated in the structure by the combined effect of internal and external temperatures; second, the structural stress generated by the internal oil–gas pressure acting on the LSG and hub; and third, the structural stress generated by the axial preload in the contact area of the LSG.

3.1. Three-Dimensional Stress Caused by Steady Temperature Field

According to the structural features of the subsea connector, the sealing structure can be simplified to a thick-walled cylinder when the temperature field reaches a steady state. Stresses can represent the stress state at any point inside the cylinder in three directions in the cylindrical coordinate system, including the circumferential stress σ_θ , the axial stress σ_z and the radial stress σ_r . In order to calculate the thermal stress by use of the generalized Hooke's law and the thermal stress function method, assumptions are made as follows: (1) the connector sealing structure is made of isotropic material; (2) the connector is only subjected to internal oil–gas pressure, no other external load; (3) steady heat conduction is conducted between components of connector sealing structure; (4) the female hub and male hub are infinitely long, and the constraints on the hub from components such as the jumper are ignored; (5) the connector sealing structure satisfies the generalized Hooke's law and the small-deformation theory.

Normally, an object expands when the temperature rises. Assuming that the expansion of any section of the micro-unit is not hindered for a temperature variation of τ ($\tau = T - T_0$, T_0 is the initial temperature and T is the final temperature), and the strain under free expansion is [30]:

$$\begin{cases} \varepsilon_{xx} = \varepsilon_{yy} = \varepsilon_{zz} = \alpha^* \tau \\ \varepsilon_{xy} = \varepsilon_{yz} = \varepsilon_{zx} = 0 \end{cases} \quad (1)$$

where: α^* is the thermal linear expansion coefficient. In fact, due to the mutual constraints between the micro-units, thermal stress will be generated, and the strain of each micro-unit is caused by the temperature variation and stress superposition. For plane strain, the relationship between strain and stress under temperature can be expressed by Hooke's Law [30] as:

$$\begin{cases} \varepsilon_{xx} = \frac{1+\nu}{E} \left\{ \sigma_{xx} - \frac{\nu}{1+\nu} (\sigma_{xx} + \sigma_{yy} + \sigma_{zz}) \right\} + \alpha^* \tau \\ \varepsilon_{yy} = \frac{1+\nu}{E} \left\{ \sigma_{yy} - \frac{\nu}{1+\nu} (\sigma_{xx} + \sigma_{yy} + \sigma_{zz}) \right\} + \alpha^* \tau \\ \varepsilon_{zz} = \frac{1+\nu}{E} \left\{ \sigma_{zz} - \frac{\nu}{1+\nu} (\sigma_{xx} + \sigma_{yy} + \sigma_{zz}) \right\} + \alpha^* \tau \\ \varepsilon_{xy} = \frac{(1+\nu)\sigma_{xy}}{E}, \varepsilon_{yz} = \frac{(1+\nu)\sigma_{yz}}{E}, \varepsilon_{zx} = \frac{(1+\nu)\sigma_{zx}}{E} \end{cases} \quad (2)$$

where ν is the Poisson's ratio, and E is the elastic modulus. For the plane, the coordinates are taken as x and y , both are independent variables, and the volume force is not counted, then the balance equation [30] is:

$$\begin{cases} \frac{\partial \sigma_{xx}}{\partial x} + \frac{\partial \sigma_{xy}}{\partial y} = 0 \\ \frac{\partial \sigma_{xy}}{\partial x} + \frac{\partial \sigma_{yy}}{\partial y} = 0 \end{cases} \quad (3)$$

Then the compatibility equation of strain can be given by:

$$\frac{\partial^2 \varepsilon_{xx}}{\partial x^2} + \frac{\partial^2 \varepsilon_{yy}}{\partial y^2} = 2 \frac{\partial^2 \varepsilon_{xy}}{\partial x \partial y} \quad (4)$$

In the plane strain state, it can be obtained from $\varepsilon_{zz} = 0$ in Equation (2) that:

$$\sigma_{zz} = \nu(\sigma_{xx} + \sigma_{yy}) - \alpha^* E \tau \quad (5)$$

By substituting Equation (5) into Equation (2), the relation between strain and stress in plane strain state can be obtained as follow:

$$\begin{cases} \varepsilon_{xx} = \frac{1-\nu^2}{E} (\sigma_{xx} - \frac{\nu}{1-\nu} \sigma_{yy}) + (1+\nu)\alpha^* \tau \\ \varepsilon_{yy} = \frac{1-\nu^2}{E} (\sigma_{yy} - \frac{\nu}{1-\nu} \sigma_{xx}) + (1+\nu)\alpha^* \tau \\ \varepsilon_{xy} = \frac{(1+\nu)\sigma_{xy}}{E} \end{cases} \quad (6)$$

The Airy's thermal stress function χ [30] is introduced so that Equation (3) can satisfy the following equation:

$$\sigma_{xx} = \frac{\partial^2 \chi}{\partial y^2}, \sigma_{yy} = \frac{\partial^2 \chi}{\partial x^2}, \sigma_{xy} = -\frac{\partial^2 \chi}{\partial x \partial y} \quad (7)$$

For the plane strain state, considering Equations (4), (6) and (7), it can be obtained as follows:

$$\Delta \Delta \chi = -\frac{\alpha^* E}{1-\nu} \Delta \tau = -k \Delta \tau \quad (8)$$

where, $k = \alpha^* E / (1 - \nu)$ [30], and:

$$\Delta = \frac{\partial^2}{\partial x^2} + \frac{\partial^2}{\partial y^2} \quad (9)$$

The stress function of the isothermal elasticity problem is the solution of the homogeneous equation, and the stress function χ of the thermal stress problem is the solution of the nonhomogeneous equation. χ can be decomposed into homogeneous equation solution U and particular solution V [30]:

$$\chi(x, y) = U(x, y) - V(x, y) \quad (10)$$

Then the solution of Equation (10) can be transformed into:

$$\begin{cases} \Delta \Delta U = 0 \\ \Delta V = k \tau \end{cases} \quad (11)$$

The temperature load inside the subsea connector is of axisymmetric distribution, and the core component of the connector is an axisymmetric structure. Therefore, the thermal elasticity of the core component can be simplified to plane thermal strain and plane thermal stress. In plane stress state, the thermal stress is solved using the complex stress function, $\phi_1(z)$, $\psi_1(z)$ [30]. The explanation and derivation process are shown in Appendix A. Then the thermal stress component can be obtained as:

$$\begin{cases} \sigma_{rr} = \frac{1}{r} \frac{\partial}{\partial r} (U - V) = \frac{kA}{2} \left(\ln \frac{r_0}{r} - \frac{r_i^2}{r^2} \frac{r_0^2 - r^2}{r_0^2 - r_i^2} \ln \frac{r_0}{r_i} \right) \\ \sigma_{\theta\theta} = \frac{kA}{2} \left(\ln \frac{r_0}{r} + \frac{r_i^2}{r^2} \frac{r_0^2 + r^2}{r_0^2 - r_i^2} \ln \frac{r_0}{r_i} - 1 \right) \end{cases} \quad (12)$$

In the above study, only σ_{rr} and $\sigma_{\theta\theta}$ are obtained, assuming that the LSG is expanded freely in the axial direction and the sum of the normal forces on the section perpendicular to the z -axis is equal to zero, then σ_{zz} can be obtained as follow:

$$\sigma_{zz} = \frac{kA}{2} \left(2 \ln \frac{r_0}{r} + \frac{2r_0^2}{r_0^2 - r_i^2} \ln \frac{r_0}{r_i} - 1 \right) \quad (13)$$

By taking $A = \tau/\ln(r_i/r_0)$ and $k = \alpha E/(1 - \nu)$ into the equation above, the thermal stress of the core sealing component can be obtained:

$$\begin{cases} \sigma_r^\tau = \frac{\alpha^* E \tau}{2(1-\nu) \ln(r_i/r_0)} \left[\ln \frac{r_0}{r} - \frac{r_0^2}{r^2} \frac{r_0^2 - r_i^2}{r_0^2 - r_i^2} \ln \frac{r_0}{r_i} \right] \\ \sigma_\theta^\tau = \frac{\alpha^* E \tau}{2(1-\nu) \ln(r_i/r_0)} \left[\ln \frac{r_0}{r} + \frac{r_i^2}{r^2} \frac{r_0^2 + r^2}{r_0^2 - r_i^2} \ln \frac{r_0}{r_i} - 1 \right] \\ \sigma_z^\tau = \frac{\alpha^* E \tau}{2(1-\nu) \ln(r_i/r_0)} \left[2 \ln \frac{r_0}{r} + \frac{2r_0^2}{r_0^2 - r_i^2} \ln \frac{r_0}{r_i} - 1 \right] \end{cases} \quad (14)$$

where, α^* is the linear expansion coefficient of steel, $^\circ\text{C}^{-1}$; E is the elastic modulus of steel, MPa; ν is the Poisson's ratio of the material; r is the point radius of the solved stress, mm; τ is the temperature difference between the component's inner wall and outer wall, $^\circ\text{C}$; r_i , r_0 are the radius of inner wall and outer wall of the component, respectively, mm.

3.2. Three-Dimensional Stress Caused by Transient Temperature Field

When the oil-gas temperature inside the connector varies, the stress state of the connector is different from the state when the temperature is stable. When the temperature varies drastically, the components inside the connector will have a large temperature difference. Due to the constraints between the components, the thermal deformation of the components will generate a relatively large thermal stress. With excessive thermal stress, there is a risk of damage to connector components, which further leads to sealing failure. During the rapid variation of oil-gas temperature, the heat of oil and gas is transferred to the inner wall of the hub and the inner wall of the LSG in the form of convection and then transferred to the outer wall of the connector by heat conduction and heat radiation. In the process of oil-gas temperature variation, the distribution of temperature inside the connector varies along the radial direction and varies with time. Therefore, the temperature of any node of the connector also varies with time to form a transient temperature field. The unstable thermal conductivity of temperature along the radial direction can be approximated as the thermal conductivity of a flat plate, which Fourier's law can describe as [30]:

$$\frac{dT}{dt} = a_d \frac{d^2t}{dr^2} \quad (15)$$

where T is temperature, $^\circ\text{C}$; t is the temperature rising time, s; r is the distance along the radial direction at any point in the wall, m; and a_d is the thermal conductivity coefficient of steel, m^2/s .

Using the approximate method [30] to solve Equation (15), yields:

$$\Delta T = \frac{\tau}{\Delta t} \frac{\delta^2}{2a_d} \quad (16)$$

where τ is the temperature difference between inner and outer wall, $^\circ\text{C}$; $\Delta T/\Delta t$ is the temperature rise rate of the inner wall from start to stop, that is, the rate of temperature variation with time of the inner wall, $^\circ\text{C}/\text{s}$; and δ is the wall thickness, m.

Substitute Equation (16) into Equation (14) to obtain the approximate expression of the transient thermal stress as follows:

$$\begin{cases} \sigma_r^\tau = \frac{\alpha^* E \tau}{2(1-\nu) \Delta t \ln \frac{r_i}{r_0}} \frac{\delta^2}{2a_d} \left[\ln \frac{r_0}{r} - \frac{r_0^2}{r^2} \frac{r_0^2 - r_i^2}{r_0^2 - r_i^2} \ln \frac{r_0}{r_i} \right] \\ \sigma_\theta^\tau = \frac{\alpha^* E \tau}{2(1-\nu) \Delta t \ln \frac{r_i}{r_0}} \frac{\delta^2}{2a_d} \left[\ln \frac{r_0}{r} + \frac{r_i^2}{r^2} \frac{r_0^2 + r^2}{r_0^2 - r_i^2} \ln \frac{r_0}{r_i} - 1 \right] \\ \sigma_z^\tau = \frac{\alpha^* E \tau}{2(1-\nu) \Delta t \ln \frac{r_i}{r_0}} \frac{\delta^2}{2a_d} \left[2 \ln \frac{r_0}{r} + \frac{2r_0^2}{r_0^2 - r_i^2} \ln \frac{r_0}{r_i} - 1 \right] \end{cases} \quad (17)$$

3.3. Three-Dimensional Stress Caused by Internal Pressure

The subsea connector is subjected to multiple loads simultaneously under working conditions. Since the sealing structure is regarded as a thick-walled cylinder, the three-dimensional stress caused by the internal pressure can be obtained according to the Lamé formula [31]:

$$\begin{cases} \sigma_r^p = \frac{pr_i^2}{r_0^2 - r_i^2} - \frac{pr_0^2 r_i^2}{r_0^2 - r_i^2} \frac{1}{r^2} \\ \sigma_\theta^p = \frac{pr_i^2}{r_0^2 - r_i^2} + \frac{pr_0^2 r_i^2}{r_0^2 - r_i^2} \frac{1}{r^2} \\ \sigma_z^p = \frac{pr_i^2}{r_0^2 - r_i^2} \end{cases} \quad (18)$$

3.4. Thermal–Structural Coupling Stress

To obtain the stress caused by the joint action of multiple loads, it is necessary to couple the various stresses. There are two conditions for stress superposition [32]: (1) The system can be described as a second-order linear differential equation; (2) the action of the factors on the system shall not cause non-linear phenomena. The structural stress and thermal stress of the LSG are both within the elastic range and can be expressed by the linear second-order differential equation. Therefore, the conditions for using the superposition principle are available. By applying the superposition principle, the coupling stress-radial stress σ_r , circumferential stress σ_θ and axial stress σ_z caused by multiple loads can be obtained:

$$\begin{cases} \sigma_r = \frac{\alpha^* E \tau}{2(1-\nu) \Delta t \ln \frac{r_i}{r_0}} \frac{\delta^2}{2a_d} \left[\ln \frac{r_0}{r} - \frac{r_i^2}{r^2} \frac{r_0^2 - r^2}{r_0^2 - r_i^2} \ln \frac{r_0}{r_i} \right] + \frac{pr_i^2}{r_0^2 - r_i^2} - \frac{pr_0^2 r_i^2}{r_0^2 - r_i^2} \frac{1}{r^2} + \sigma_r^o \\ \sigma_\theta = \frac{\alpha^* E \tau}{2(1-\nu) \Delta t \ln \frac{r_i}{r_0}} \frac{\delta^2}{2a_d} \left[\ln \frac{r_0}{r} + \frac{r_i^2}{r^2} \frac{r_0^2 - r^2}{r_0^2 - r_i^2} \ln \frac{r_0}{r_i} - 1 \right] + \frac{pr_i^2}{r_0^2 - r_i^2} + \frac{pr_0^2 r_i^2}{r_0^2 - r_i^2} \frac{1}{r^2} + \sigma_\theta^o \\ \sigma_z = \frac{\alpha^* E \tau}{2(1-\nu) \Delta t \ln \frac{r_i}{r_0}} \frac{\delta^2}{2a_d} \left[2 \ln \frac{r_0}{r} + \frac{r_i^2}{r^2} \frac{r_0^2 - r^2}{r_0^2 - r_i^2} \ln \frac{r_0}{r_i} - 1 \right] + \frac{pr_i^2}{r_0^2 - r_i^2} + \sigma_z^o \end{cases} \quad (19)$$

4. Numerical Simulation of Thermal–Structural Coupling of the Subsea Connector

The simulation analysis of thermal–structural coupling stress of the subsea connector requires the loads, including oil–gas temperature, preload, and oil–gas pressure to be applied to the connector, which increases the complexity of the numerical simulation. In this paper, the temperature field numerical simulation of the subsea connector is conducted first. Next, the node temperature of the components obtained is applied to the connector as steady loads together with the rest of the loads. Finally, the coupling thermal–structural stress state of the connector under the action of the transient temperature field is obtained through the steady solution. Nickel–chromium–iron corrosion-resistant alloy Incoloy825 is selected for the material of the LSG in the simulation, and 12Cr2Mo1 is selected for the material of hub and collet. The material parameters are shown in Tables 1 and 2.

Table 1. Material model of Incoloy825.

Temperature (°C)	Heat Capacity (J·kg ^{−1} ·K ^{−1})	Elastic Modulus (MPa)	Thermal Expansion Coefficient (°C ^{−1})	Thermal Conductivity (W·m ^{−1} ·K ^{−1})	Poisson's Ratio
20	3.603	2.06 × 10 ⁵	1.64 × 10 ^{−5}	14.7	0.25
50	3.744	1.970 × 10 ⁵	1.65 × 10 ^{−5}	15.2	0.25
100	3.901	1.955 × 10 ⁵	1.68 × 10 ^{−5}	15.8	0.25
150	4.103	1.915 × 10 ⁵	1.70 × 10 ^{−5}	16.7	0.25

Table 2. Material model of 12Cr2Mo1.

Temperature (°C)	Heat Capacity (J·kg ⁻¹ ·K ⁻¹)	Elastic Modulus (MPa)	Thermal Expansion Coefficient (°C ⁻¹)	Thermal Conductivity (W·m ⁻¹ ·K ⁻¹)	Poisson's Ratio
20	3.603	2.1×10^5	1.64×10^{-5}	14.7	0.3
50	3.744	1.970×10^5	1.65×10^{-5}	15.2	0.3
100	3.901	1.955×10^5	1.68×10^{-5}	15.8	0.3
150	4.103	1.915×10^5	1.70×10^{-5}	16.7	0.3

4.1. Transient Temperature Field Analysis of the Subsea Connector

Under the rated working conditions, the outer surface of the subsea connector is exposed to low-temperature seawater (3 °C). According to the boundary condition of the third kind, this paper uses the equivalent thermal conductivity and composite thermal conductivity obtained from the analysis in Section 3. The internal surface of the connector is in contact with the high-temperature oil and gas. The internal surface temperature of the connector in contact with oil and gas is set in different working conditions:

1. The condition of uniform temperature rise: after 100 s, the internal surface temperature of the connector rises from 3 °C to 150 °C.
2. The condition of uniform temperature drop: based on the steady temperature distribution, after 100 s, the internal surface temperature of the connector drops from 150 °C to 3 °C.
3. The condition of instantaneous temperature drop: based on the steady temperature distribution, the internal surface temperature of the connector instantaneously drops to 3 °C.
4. The condition of temperature shock: after 100 s, the internal surface temperature of the connector rises from 3 °C to 150 °C, and then after another 100 s, it drops from 150 °C to 3 °C.

4.1.1. Analysis of Temperature Field under the Working Condition of Uniform Temperature Rise

The working condition of uniform temperature rise aims to simulate the process of heat transfer in the connector when the wellhead is just opened. During the simulation, the connector internal temperature is set to rise from 3 °C to 150 °C at a uniform speed within 100 s, and the external environment temperature is 3 °C. As shown in Figure 2, the temperature fields at 50 s, 500 s, 5000 s and 10,000 s are obtained. There is a temperature gradient in the radial section inside of the connector. We can find that:

- After 50 s, the oil–gas temperature inside the connector rises to 76.5 °C. The temperature of the inner wall of the hub and the inner wall of the LSG reaches 75.76 °C, and the heat is mainly concentrated on the LSG. The temperature of the hub close to its inner wall is relatively higher, and the temperature of the hub outer wall, clamp and actuator ring is the same as the temperature of seawater.
- After the temperature rising for 500 s, the oil–gas temperature reaches 150 °C and maintains for a period of time. The overall temperature of the LSG and hub increases significantly, and the heat is transferred to the collet through the outer wall of the hub.
- After the temperature rising for 5000 s, the heat is transferred to the actuator ring. However, the heat on the collet and actuator ring was mainly concentrated near the hub.
- Comparing the temperature field distributions at 5000 s and 10,000 s, the temperature field tends to be stable after the temperature rising for 5000 s.

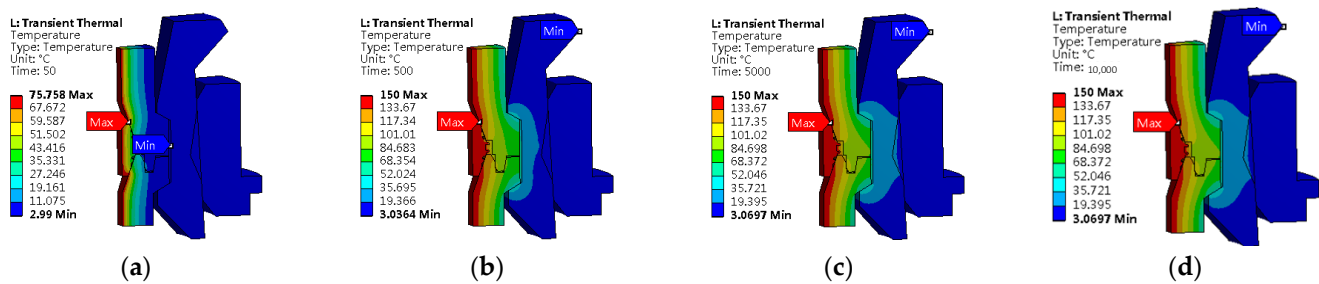


Figure 2. Transient temperature field of subsea collet connector under the working condition of uniform temperature rise. (a) 50 s; (b) 500 s; (c) 5000 s; (d) 10,000 s.

During the temperature rising process, the temperature rise rate of each component is different. The real-time expansion rate and thermal stress are also different, which would lead to variation in the preload of the LSG. Therefore, the temperature rise process will have a certain impact on the sealing performance.

4.1.2. Analysis of Temperature Field under the Working Condition of Uniform Temperature Drop

The working condition of uniform temperature drop aims to simulate temperature variation inside the connector as the wellhead closes and the oil–gas temperature flowing through the subsea collet connector drops gradually. During the simulation, the connector internal temperature is set to drop from 150 °C to 3 °C at a uniform speed within 100 s, and the external environment temperature is 3 °C. Then the temperature fields at 50 s, 500 s, 5000 s and 10,000 s are obtained, as shown in Figure 3. We can find that:

- After the temperature dropping for 10 s, the transient maximum temperature (140.26 °C) is mainly on the LSG.
- After the temperature dropping for 100 s, the transient maximum temperature (106.92 °C) is concentrated in the seawater layer between the LSG and hub.
- After the temperature dropping for 1000 s, the transient heat was mainly concentrated in the seawater layer between the actuator ring and the collet, and the temperature is 6.362 °C.
- After the temperature dropping for 10,000 s, the overall temperature inside the connector has been reduced to seawater temperature.

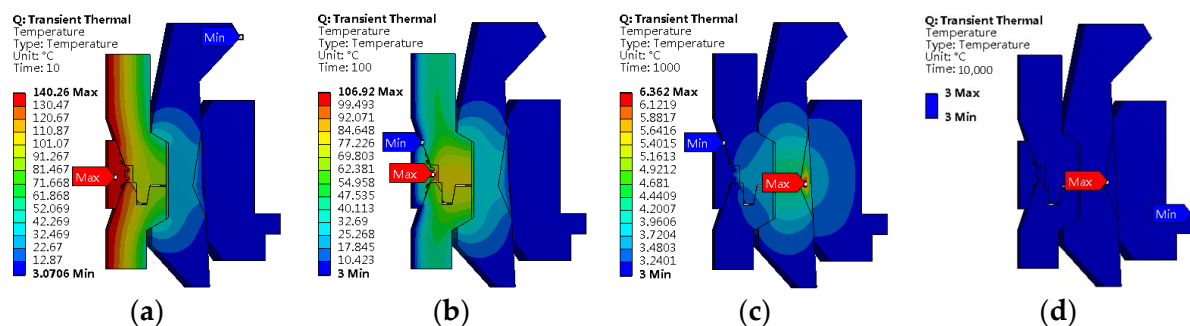


Figure 3. Transient temperature field of subsea collet connector under the working condition of uniform temperature drop. (a) 10 s; (b) 100 s; (c) 1000 s; (d) 10,000 s.

Table 3 shows the temperature drop time of each component. Due to the initial drop speed of LSG temperature being significantly faster than other components, the instantaneous shrinkage will be greater than other components, which will impact the sealing performance of the connector. However, in a short period, the temperature of each component is reduced by 90%, but it takes a long time to reach the steady distribution of the same temperature as the seawater.

Table 3. The time required for each component to drop to the seawater temperature under the working condition of uniform temperature drop.

Percentage of Temperature Drop (%)	Time for Temperature Drop (s)				
	LSG	Female Hub	Male Hub	Collet	Actuator Ring
20	14.4	124	74.4	102	274
40	19.4	174	114	154	464
60	23.4	234	164	264	974
80	27.4	354	264	364	1230
90	194	505	400	584	1770
100	3170	3870	3670	3570	3470

4.1.3. Analysis of Temperature Field under the Working Condition of Instantaneous Temperature Drop

The working condition of instantaneous temperature drop aims to simulate the temperature variation inside the connector when the subsea collet connector stops working, and the oil–gas temperature is removed instantaneously within 10 s. The temperature field inside the connector under the working condition of instantaneous temperature drop at 10 s, 100 s, 1000 s and 5000 s after the wellhead is closed is obtained by simulation as shown in Figure 4. We can find that:

- After the temperature dropping for 10 s, the maximum instantaneous temperature of the connector is concentrated at the LSG, which is 142.49 °C.
- After the temperature dropping for 100 s, compared with the temperature field figure at 10 s, it is found that after the oil–gas temperature is instantly removed, the inner wall of the hub and the inner wall of the LSG dissipate heat rapidly, and the temperature drops drastically. The heat is mainly concentrated at the junction of the outer wall of the LSG and seawater, and the maximum temperature is 122.8 °C; the temperature field distribution of the collet and actuator ring is similar to that at 10 s; however, the temperature is slightly lower.
- After the temperature dropping for 1000 s, the overall temperature inside the connector is relatively lower, and the highest temperature appears in the seawater layer between the LSG and hub, which is 18.634 °C.
- After the temperature dropping for 5000 s, the connector temperature is close to the seawater temperature, with a maximum temperature of merely 3.0028 °C.

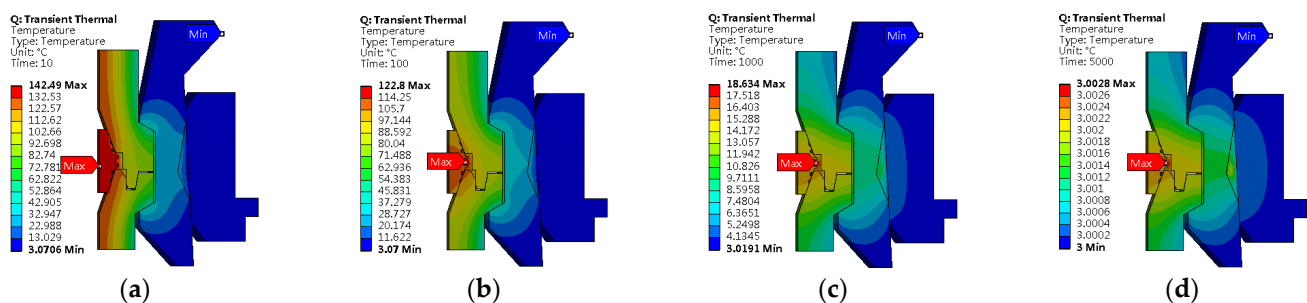


Figure 4. Transient temperature field of subsea collet connector under the working condition of instantaneous temperature drop. (a) 10 s; (b) 100 s; (c) 1000 s; (d) 5000 s.

Table 4 shows the time for a temperature drop of each component under the working condition of instantaneous temperature drop. From Table 4, the time for a temperature drop of each component is relatively close. Therefore, there is no sudden variation in the temperature difference of each component, and there is no sudden variation in the force between each component, which has little impact on the performance of the connector.

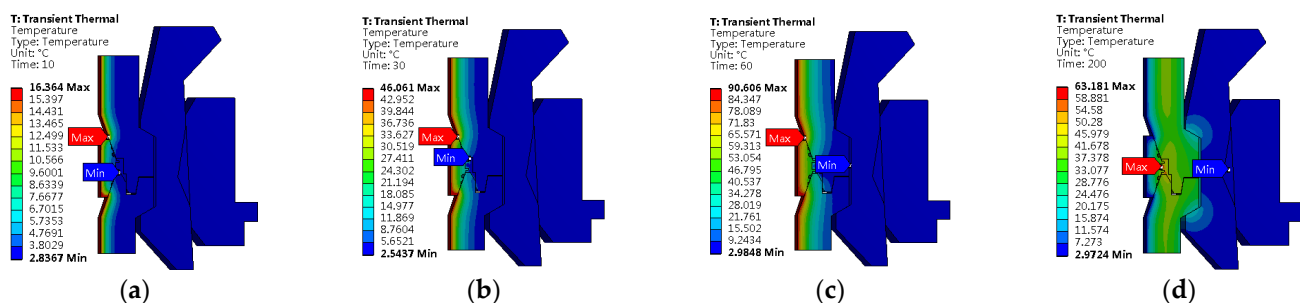
Table 4. The time required for each component to drop to the seawater temperature under the working condition of instantaneous temperature drop.

Percentage of Temperature Drop (%)	Time for Temperature Drop (s)				
	LSG	Female Hub	Male Hub	Collet	Actuator Ring
20	102	200	144	164	324
40	224	344	284	324	505
60	405	534	474	534	724
80	724	874	800	934	1050
90	1050	1240	1140	1640	1440
100	6870	6770	6770	6370	5770

4.1.4. Analysis of Temperature Field under the Working Condition of Temperature Shock

The working condition of temperature shock aims to simulate the temperature variation inside the connector when the wellhead opens quickly and then closes quickly. The connector internal temperature is set to rise from 3 °C to 150 °C within 100 s, and then drop from 150 °C to 3 °C within 100 s; the external environment temperature is 3 °C. Then the temperature fields under temperature shock at 10 s, 30 s, 60 s and 200 s are obtained, as shown in Figure 5. We can find that:

- After 10 s, the heat is mainly concentrated in the inner walls of the LSG and the hub, and the maximum temperature is 16.364 °C.
- After 30 s, the heat gradually diffuses to the outer walls of the LSG and the hub, with a maximum temperature of 46.061 °C.
- After 60 s, the temperature field distribution is similar to that at 10 s. However, the temperature is relatively higher, and the maximum temperature is 90.606 °C.
- After 200 s, the temperature is reduced to 3 °C. The heat is mainly concentrated at the hub, and the maximum temperature is reduced to 63.181 °C.

**Figure 5.** Transient temperature field of subsea collet connector under the working condition of temperature shock. (a) 10 s; (b) 30 s; (c) 60 s; (d) 200 s.

From Table 5, the LSG reaches the peak temperature at the earliest time, and the female hub, male hub, collet, and actuator ring lag behind for a period of time, respectively. Considering the time of the temperature peak and the maximum temperature peak, the lenticular seal, female hub and male hub are subjected to the greatest influence of temperature shock, followed by the collet and actuator ring.

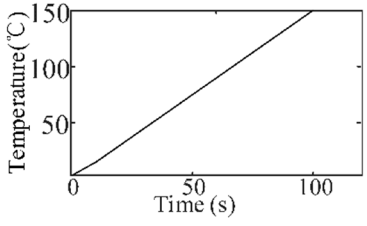
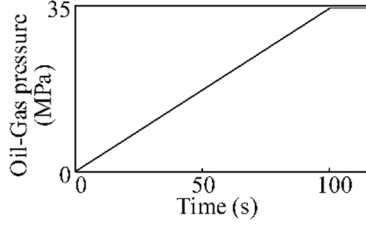
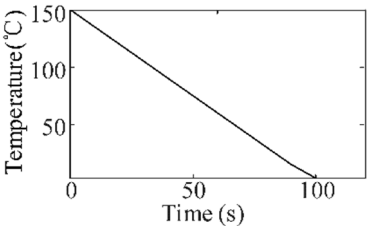
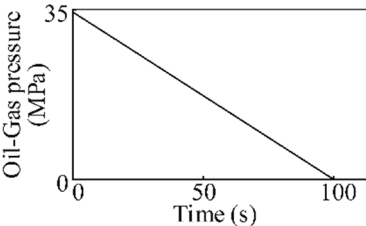
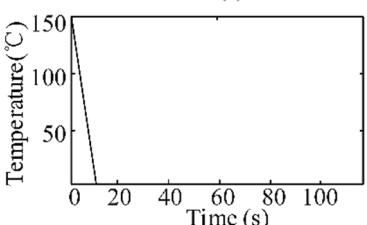
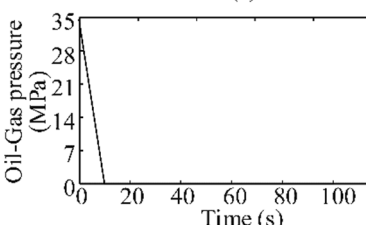
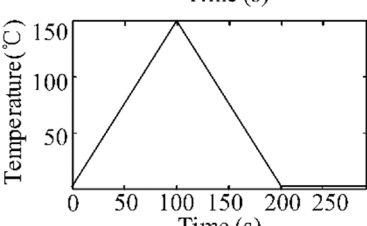
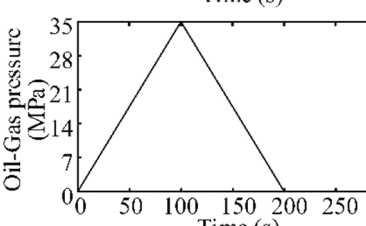
Table 5. The time required for each component to generate a temperature peak under the working condition of temperature shock.

Name of Component	Peak Temperature (°C)	Generation Time (s)	Lag Time (s)
LSG	105.37	118.02	18.02
Female hub	36.112	190.83	90.83
Male hub	54.438	140.83	40.83
Collet	18.352	170.83	70.83
Actuator ring	4.0547	305	205

4.2. Analysis of Coupling Stress Calculation Examples under the Transient Temperature Field

A certain axial preload is applied to the subsea collet connector. Then loads of temperature–pressure rise, temperature–pressure drop, instantaneous temperature–pressure drop, and temperature–pressure shock are applied sequentially to analyze the sealing performance of the connector under the action of transient temperature field and pressure. The four working conditions of the subsea collet connector in the transient temperature field are analyzed. The load conditions are shown in Table 6, respectively. The working condition of temperature–pressure rise aims to analyze the connector performance after the wellhead is opened and a large amount of oil and water mixture is passed inside the subsea collet connector, resulting in a simultaneous rise in oil–gas pressure and temperature; the working condition of temperature–pressure drop refers to the seawater drawn from the vicinity of the wellhead is far more than the oil and gas mixture, and the internal pressure and temperature drop as it flows through the connector, thus to simulate the variation of sealing performance of the connector at this time; the working condition of instantaneous temperature–pressure drop aims to simulate the oil–gas temperature and pressure inside the subsea collet connector, and the stress variation of the components of the connector; the working condition of temperature–pressure shock aims to simulate the performance variation of the connector when the wellhead is opened and closed rapidly, the high-temperature and high-pressure oil and gas passes through and is suddenly cut off.

Table 6. Transient temperature field load conditions.

Case	Load Condition		
	Axial Preload	Transient Temperature Load	Oil-Gas Pressure Load
Temperature-pressure rise	63.8 kN		
Temperature-pressure drop	63.8 kN		
Temperature-pressure instantaneous drop	63.8 kN		
Temperature-pressure shock	63.8 kN		

4.2.1. Coupling Stress Analysis under the Working Condition of Temperature–Pressure Rise

The maximum contact stress variation of the LSG within 1000 s is shown in Figure 6a: from 0 to 90 s, the maximum contact stress of the lenticular seal continues to rise. At 90 s, the contact stress reaches a peak value of 487.26 MPa, which is 10 s earlier than the time when the temperature and pressure reach the peak. From 90 s to 110 s, the maximum contact stress of the seal decreases to 458.6 MPa; from 110 s to 1000 s, the maximum contact stress slowly fluctuates up to 472.6 MPa and reaches stability. The variation of the maximum equivalent stress of the LSG is shown in Figure 6b. According to the coupling stress theory analysis in 3 sections, the maximum equivalent stress of LSG is directly related to the sealing contact stress and has a similar variation law: from 0 to 90 s, the maximum equivalent stress rises gradually and reaches a peak of 258.91 MPa at 90 s; from 90 s to 130 s, the maximum equivalent stress gradually decreases to 232.29 MPa; from 130 s to 1000 s, the maximum equivalent stress slowly fluctuates up to 250.14 MPa. From 0 to 90 s, as the temperature difference inside the connector increases, the temperature of the inner wall of the LSG and the female and male hubs rise faster. As the temperature difference between the inner and outer walls of the LSG gradually increases, the inner wall expansion of the LSG is greater than the outer surface. The high oil and gas pressure makes the LSG actively press the female and male hubs, resulting in a continuous rise in contact stress. After 90 s, the temperature difference gradually decreases until a steady temperature distribution is presented. The contact stress firstly decreases and then increases, and then gradually stabilizes. The increase of temperature and pressure significantly influences the maximum contact stress of the LSG, which proves that the metal seal is slightly deficient in temperature compensation.

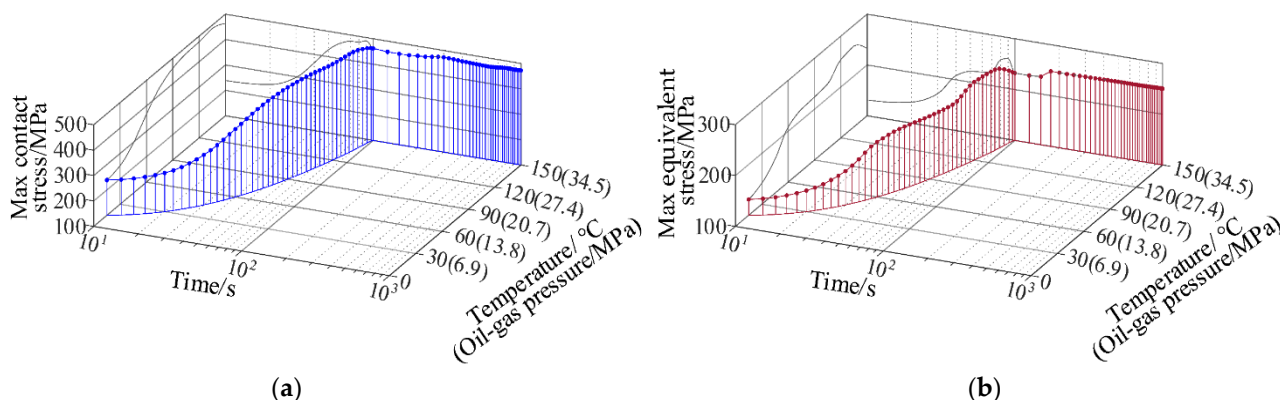


Figure 6. Variations of maximum contact stress and maximum equivalent stress of LSG under the working condition of temperature–pressure rise. (a) Maximum contact stress; (b) maximum equivalent stress.

4.2.2. Coupling Stress Analysis under the Working Condition of Temperature–Pressure Drop

The variations of maximum contact stress and the maximum equivalent stress of the LSG are shown in Figure 7. It can be seen that: as the temperature and pressure drop at the same time, the tendency of the LSG to actively compress the female and male hubs is slowed down, and the contact stress decreases. From 0 to 100 s, the maximum contact stress decreases to 272.21 MPa; from 100 s to 300 s, as the connector starts to dissipate heat to the outside, the temperature difference between the inside and the outside gradually decreases, the contact stress decreases slowly, and the maximum contact stress decreases slowly to 210.06 MPa; from 300 s to 1000 s, the maximum contact stress fluctuates slowly but does not change much, and finally stabilizes at 215.46 MPa. From 0 to 300 s, the maximum equivalent stress generally shows a decreasing trend, during which it fluctuates slightly due to heat dissipation, and it decreased to 107.15 MPa at 300 s. From 300 s to 500 s, the maximum equivalent stress has a certain change, the maximum value is 112.55 MPa, and the minimum value is 106.6 MPa after 500 s, and then tends to be stable.

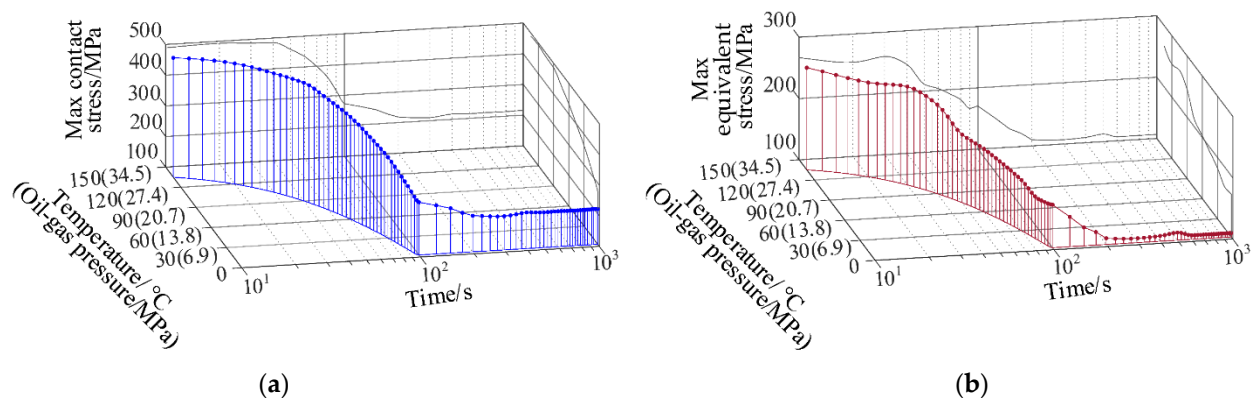


Figure 7. Variations of maximum contact stress and maximum equivalent stress of the LSG under the working condition of temperature–pressure drop. (a) Maximum contact stress; (b) maximum equivalent force.

4.2.3. Coupling Stress Analysis under the Working Condition of Temperature–Pressure Instantaneous Drop

The oil–gas pressure and temperature are both set to the lowest under the working condition of temperature–pressure instantaneous drop after 10 s. The maximum contact pressure of the LSG and the equivalent stress variation are shown in Figure 8. From 0 to 10 s, due to the rapid drop of oil–gas pressure and temperature, the heat dissipation efficiency of the LSG and the inner wall of the hub is higher than that of the overall hub, which slows down the influence of temperature and pressure removal. Therefore, the maximum contact stress slowly decreases to 445.68 MPa, and there is no sudden drop of contact stress; from 10 s to 300 s, as the temperature and oil–gas pressure no longer vary, the hub, collet and actuator ring continue to dissipate heat, and the sealing contact stress is quickly reduced to 220.88 MPa; from 300 s to 1000 s, the temperature of the hub is consistent with the temperature of the internal and external environment, and the heat is mainly concentrated in the collet. As the heat of the collet continues to dissipate, the expansion decreases, resulting in a tendency to compress the female and male hubs so that the maximum sealing contact stress slowly rises to 228.9 MPa. After 500 s, the collet maintains this value after the heat dissipates, thus reaching a steady state. Under this working condition, as shown in Figure 8b, from 0 to 10 s, different from the variation of contact stress, the maximum equivalent stress of the LSG presents a slight increase, and the peak stress is 266.78 MPa; from 10 s to 300 s, the maximum equivalent stress of the LSG continues to drop to 99.84 MPa; from 300 s to 1000 s, the maximum equivalent stress of the LSG slowly rises to 106.67 MPa and remains stable, which is consistent with the reason for the variation of the maximum contact stress.

4.2.4. Coupling Stress Analysis under the Working Condition of Temperature–Pressure Shock

The maximum contact stress and the maximum equivalent stress of the LSG under the working condition of temperature–pressure shock are shown in Figure 9. From 0 to 20 s, the initial impact of temperature–pressure shock is not obvious, and the maximum contact stress rises slowly to 296.81 MPa; from 20 s to 90 s, the temperature of both the LSG and the hub begins to rise rapidly, and also by the effect of increased oil–gas pressure, the tendency of the LSG to compress the hub outward is more obvious. Therefore, the maximum contact stress quickly rises to the peak value, which is 487.26 MPa, and the peak time is 10 s earlier than the peak oil–gas temperature and pressure; from 90 s to 100 s, with the increase of temperature, the maximum contact stress is reduced to 462.11 MPa for the expansion rate of hub and LSG is closer; from 100 s to 200 s, with the decrease of temperature and oil–gas pressure, the tendency of the LSG to actively compress the hub is rapidly reduced, and the maximum contact stress rapidly drops to 197.75 MPa; from 200 s to 2000 s, as the temperature and the oil–gas pressure load are restored to the initial

state, the contact stress fluctuates with the components are dissipated outwards, and the maximum contact stress stabilizes at 212.33 MPa after 2000 s. As shown in Figure 9b, under the impact of temperature–pressure shock, the variation trend of maximum equivalent stress of the LSG is similar to that of contact stress, except that there is a small fluctuation at the peak. At 110 s, the stress peak is 260.46 MPa, which is 10 s later than the temperature and pressure peak; from 200 s to 800 s, the maximum equivalent stress fluctuates with the outward heat dissipation, and finally reaches 124.36 MPa after 2000 s, and then remains stable. Temperature–pressure shock greatly impacts the performance of the LSG with large fluctuations, which may give rise to fatigue wear, which should be avoided.

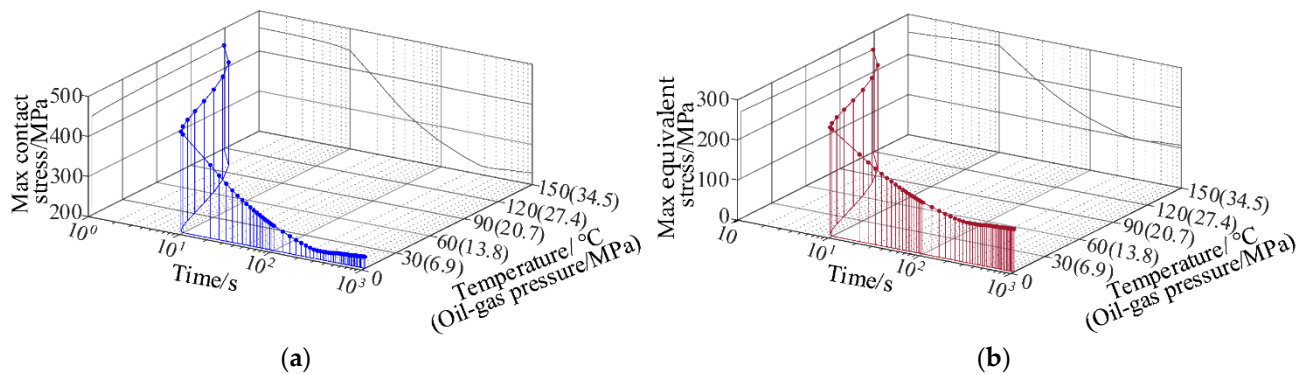


Figure 8. Variations of maximum contact stress and maximum equivalent stress of LSG under the working condition of temperature–pressure instantaneous drop. (a) Maximum contact stress; (b) maximum equivalent force.

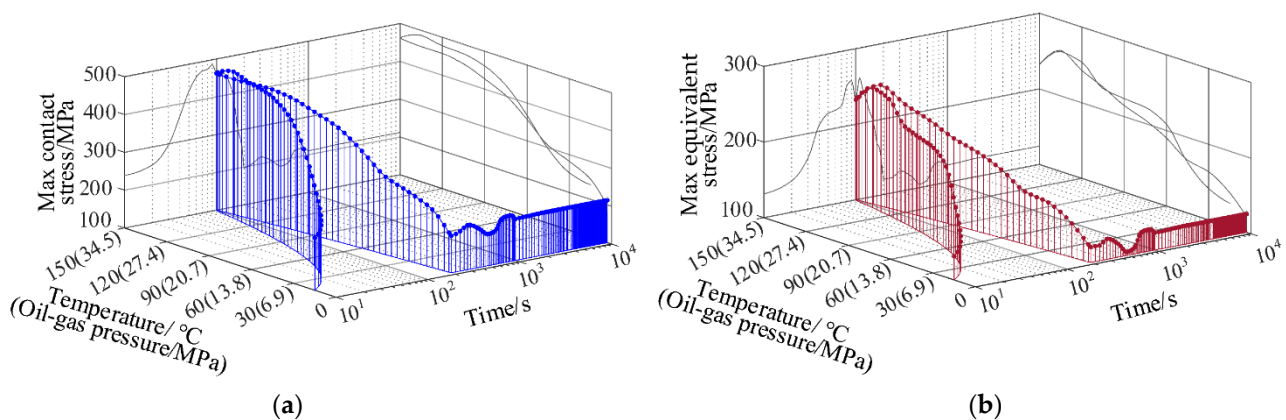


Figure 9. Variations of maximum contact stress and maximum equivalent stress of LSG under the working condition of temperature–pressure shock. (a) Maximum contact stress; (b) maximum equivalent stress.

5. Temperature Cycle Test of Sealing Performance

The temperature cycle test is mainly used to test the sealing performance and mechanical properties of the subsea collet connector under temperature cycling variations. The temperature load is applied inside and outside the connector by the temperature cycle test device. The pressure is applied inside the connector with the hydraulic pump. The temperature and strain variations of the key sampling point are obtained through the temperature sensor and strain gauge of the key test point of the connector.

5.1. Temperature Cycle Test Device and Process

Figure 10 shows the schematic diagram of the temperature cycle test device, and Figure 11 shows the physical object. The test device mainly consists of the temperature cycle test box, the subsea collet connector, the liquid nitrogen container, and the console station. The internal medium of the subsea collet connector is hydraulic oil, and the whole

test process is operated through the console station. For the test process of temperature rising, control the heater 12 to heat the hydraulic oil in the inner cavity of the connector to the specified temperature. For the high-temperature insulation process, the temperature sensor 10 in the hub of the connector and the control system jointly adjust the operation of the heater 12 to maintain the internal high temperature of the connector; for the test process of temperature dropping, start the cryogenic liquid pump 2, liquid nitrogen vaporizes through vaporizer 3 and enters temperature cycle test box 5 until the temperature in connector decreases to the specified value; for the low-temperature insulation process, the work of the cryogenic liquid pump is adjusted through the air temperature sensor 9 and the control system to maintain a constant low temperature inside the box; for the process of pressure rising, the connector internal is pressed by the hydraulic pump 17. In addition to the temperature dropping, the temperature inside the box is maintained at about 3 °C.

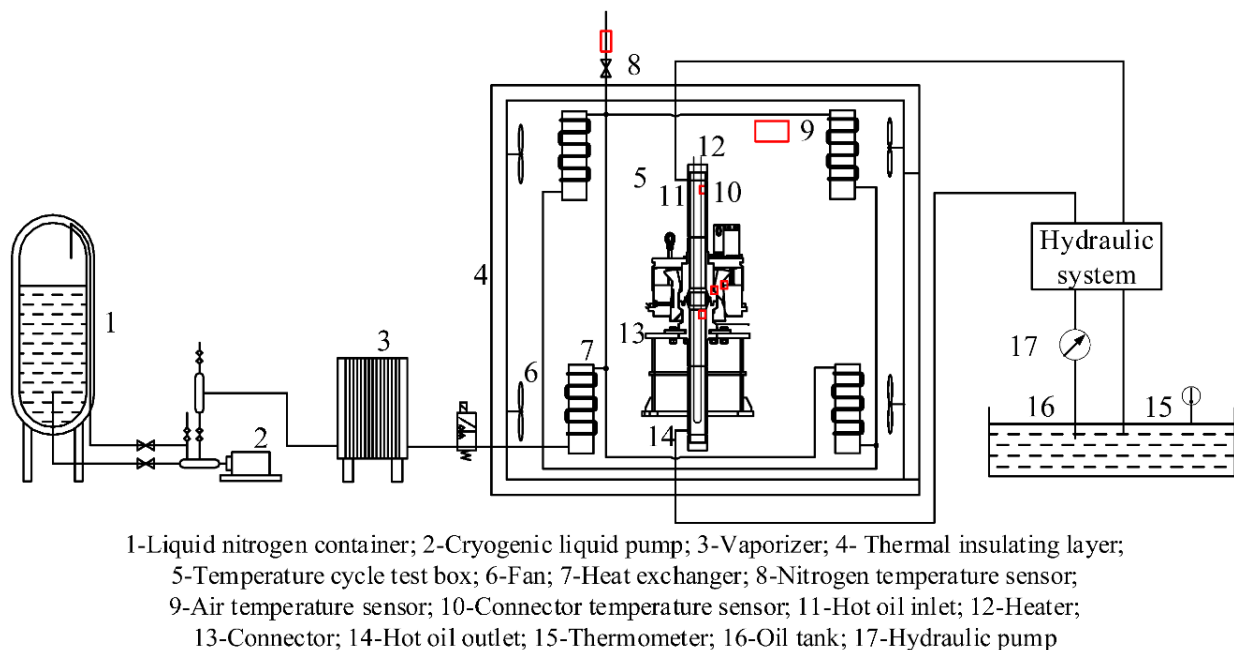


Figure 10. Schematic diagram of temperature cycle test.

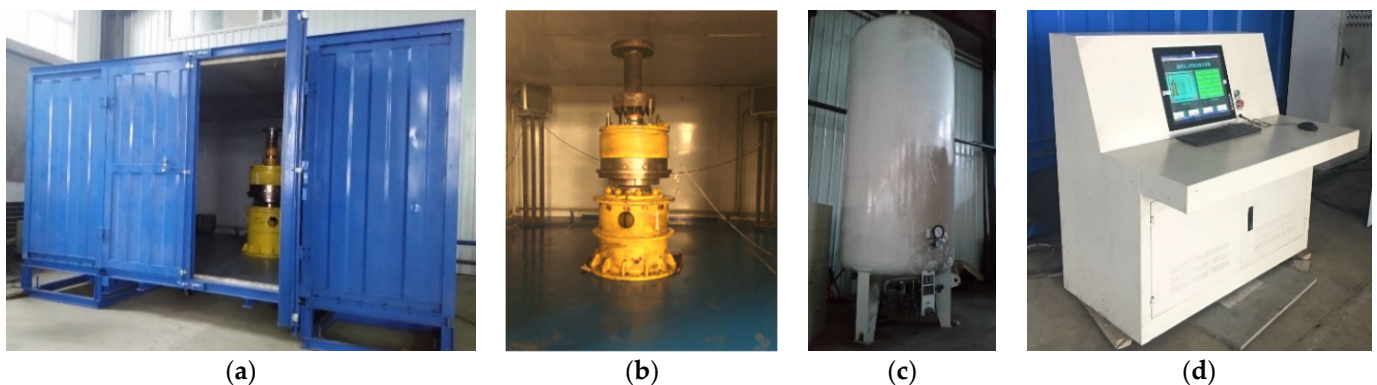


Figure 11. Device of temperature cycle test. (a) Temperature cycle test box; (b) connector and connection line; (c) liquid nitrogen container; (d) console station.

The flow chart of the temperature cycle test of the subsea collet connector is shown in Figure 12. The test can be divided into the following main steps:

1. The temperature of the temperature cycle test box outside the connector is set to 3 °C throughout, the initial temperature of the inner cavity of the connector is room

temperature, and the internal pressure is 0 MPa. The inner cavity of the subsea collet connector is heated to 150 °C.

2. Pressurize the connector cavity, increase the internal pressure to 34.5 MPa, hold the pressure for 60 min, and then depressurize to 0 MPa.
3. Reduce the temperature of the connector cavity to 3 °C.
4. Increase the internal pressure of the connector to 34.5 MPa, hold the pressure for 60 min, and then depressurize to 0 MPa.
5. Increase the temperature of the connector cavity to room temperature.
6. Increase the internal pressure of the connector to 34.5 MPa at room temperature and then raise the temperature of the connector cavity to 150 °C, maintaining the inner cavity pressure of the connector at more than 0.5 times of 34.5 MPa.
7. Set the internal pressure of the connector to 34.5 MPa and hold it for 60 min.
8. Reduce the temperature of the connector cavity to 3 °C and maintain the internal pressure of the connector at 34.5 MPa for more than 0.5 times during the temperature drop process.
9. Set the internal pressure of the connector to 34.5 MPa and hold it for 60 min.
10. Increase the temperature of the connector cavity to room temperature, maintain the internal pressure of the connector at more than 0.5 times 34.5 MPa during the temperature rise.
11. Depressurize the connector to 0 MPa and raise the internal temperature of the connector from room temperature to 150 °C.
12. Set the internal pressure of the connector to 34.5 MPa, hold it for 60 min, and then depressurize it to 0 MPa.
13. Reduce the temperature of the connector cavity to 3 °C.
14. Set the internal pressure of the connector to 34.5 MPa, hold it for 60 min, and then depressurize it to 0 MPa.
15. Increase the temperature of the connector cavity to room temperature.
16. Set the internal pressure of the connector to 34.5 MPa, hold it for 60 min, and then depressurize it to 0 MPa.
17. Set the initial pressure in the connector cavity to 10% of 34.5 MPa, and then hold this pressure for 60 min.
18. Depressurize the internal pressure of the connector to 0 MPa, and then end the test.

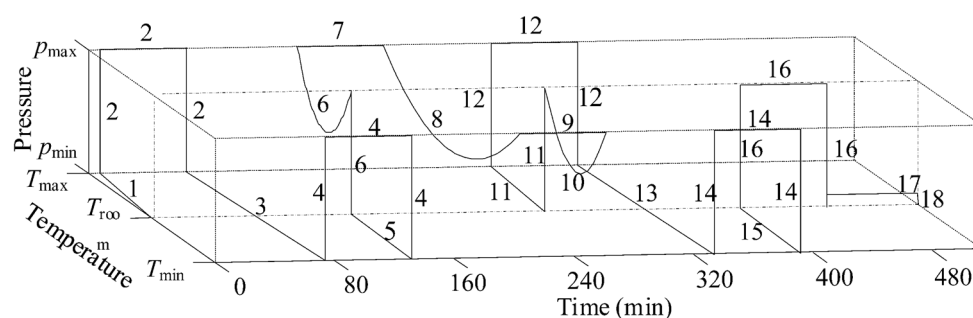


Figure 12. Flow chart of temperature cycle test loading.

To ensure the stability of the pressure-holding process, the temperature variation rate during the whole process should be less than 0.5 °C/min. In the pressure-holding stage, the temperature load applied should maintain or exceed the specified test temperature and be no higher than the specified test temperature of 11 °C. During the whole test, it is required that the deep-water jaw connector has no leakage visible to the naked eye. During the entire test process, there shall be no visible leakage in the subsea collet connector. In the pressure-holding process, the same as the hydrostatic pressure test, an acceptable pressure settling rate should not be exceeding 5% of the applied pressure or 3.45 MPa, taking the smaller of the two.

5.2. Analysis of Temperature Data

The holding-pressure data of the temperature cycle test is shown in Table 7.

Table 7. Holding-pressure data of the temperature cycle test.

Steps	Temperature (°C)	Pressure (MPa)	Holding Pressure (min)	Pressure Drop (MPa)
1	20.8–150.2	0	-	-
2	150.2	34.6	60	0.9
3	150.2–3.1	0	-	-
4	3.1	34.8	60	0.2
5	3.1–20.2	0	-	-
6	20.2–151.0	17.3–34.0	-	-
7	151.0	34.5	60	0.7
8	151.0–3.3	17.9–34.2	-	-
9	3.3	34.5	60	0.5
10	3.3–20.5	17.5–34.6	-	-
11	20.5–152.8	0	-	-
12	152.8	34.8	60	0.7
13	152.8–3.1	0	-	-
14	3.1	34.2	60	0.6
15	3.1–21.9	0	-	-
16	21.9	34.6	60	0.1
17	21.9	3.5	60	0.1
18	21.9	0	-	-

1. It can be seen that the connector is pressure-held at different temperatures. The maximum pressure drop is 0.9 MPa in the second stage, which is 2.6% of the initial loading pressure and is within the allowable pressure settling range. After the temperature and pressure cycle loading test, the connector maintains a pressure of 3.5 MPa in the 17th stage. The pressure drop is 0.1 MPa, 2.86% of the initial loading pressure and less than 5% of the loading pressure. This shows that the connector still enjoys good sealing performance after temperature and pressure cycles.
2. A comparison of the composite LSG after the test is shown in Figure 13.
 - Figure 13a shows the LSG after the hydrostatic pressure test of 1.5 times the rated internal pressure. The LSG has uniform indentation without disorderly wear and scratch. The seal width is approximately 2.17 mm measured by a vernier caliper with an accuracy of 0.1 mm; meanwhile, the rubber O-ring is in good shape without scratch and distortion.
 - Figure 13b shows the LSG after the temperature cycle test. The LSG has pronounced indentation. There are some short scratches near the indentation, and the overall seal width seems to be increased. The width of the indentation and scratch is about 4.92 mm, which is 2.27 times that of the hydrostatic pressure test; through observation, the rubber O-ring is in good shape without excessive wear.
 - The most obvious results emerge from the test is that under the multiple effects of temperature load, pressure load and preload, rapid temperature rising and dropping will lead to increasing wear of the spherical surface of the LSG. The composite sealing structure can effectively address the insufficient temperature compensation capacity of metal to ensure the sealing performance of the subsea collet connector under loads of temperature and pressure cycles.



Figure 13. The state of the composite LSG after the test. (a) After the hydro-steady internal pressure test; (b) after the temperature cycle test.

6. Conclusions

In this paper, the heat transfer model of the seawater layer between various components of the subsea collet connector was established and solved, and the transient temperature field distribution of the connector was obtained; the stress and deformation of each component of the subsea connector under thermal stress were considered, and the thermal–structural coupling analysis model of the transient temperature field was established. The comparison between the thermal stress theoretical analysis and numerical simulation of key sealing components of the connector was conducted. Analysis of the working condition of pressure shock under the transient temperature field was also performed. It can be proved that the coupled mathematical model proposed in this paper can be used for the transient thermal–structural coupling theoretical analysis of similar subsea oil and gas equipment.

1. The transient thermal–structural coupling of the subsea connector was mathematically modelled. The mathematical modelling of transient thermal–structural coupling of the subsea connector was carried out. The calculation models of radial stress, circumferential stress and axial stress of the LSG under combined loads of the axial preload, internal oil–gas pressure and temperature were obtained.
2. Numerical simulations of the transient temperature field of the subsea collet connector under four working conditions of uniform temperature rise, uniform temperature drop, instantaneous temperature drop, and temperature shock were carried out, and the study showed that: under each working condition, in the process of temperature variation from transient to steady distribution, the larger the temperature difference between the components, the greater the difference in expansion rate, the greater the impact on the sealing performance of the LSG.
3. Numerical simulations of the working conditions of temperature–pressure rise, temperature–pressure drop, instantaneous temperature–pressure drop, temperature–pressure shock under the transient temperature field were carried out. The results showed that the smooth variation of oil–gas pressure and temperature had little impact on the sealing performance of the connector; when the temperature and oil and gas pressure varied suddenly, the maximum contact stress and equivalent stress of the sealing surface would fluctuate greatly, which was prone to fatigue and wear, and thus to affect the reliability of the sealing performance, this situation should be avoided.
4. After the temperature cycle test, the final connector holding pressure drop is 2.86% of the initial loading pressure, indicating that the connector still enjoys good sealing performance after the temperature cycle. After the temperature cycle test, there is an increased seal width of the LSG. The rubber O-ring is in good shape without excessive wear.

Although this paper has studied the transient thermal–structural coupling of the subsea connector and obtained some research results, the sealing of the subsea connector

under the temperature load still involves many problems. In future studies, we will carry out the research on the wear and fatigue characteristics of the microstructure of the sealing surface under irregular temperature disturbance, and find a way to solve the problem of sealing surface wear.

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Appendix A

By using the complex stress function, $\phi_1(z)$, $\psi_1(z)$, the thermal stress can be solved in plane stress state. S is the simply connected region bounded by L_0 in the z -plane. According to the analytic function:

$$\begin{cases} z = \omega(\zeta) \\ \zeta = \xi + i\eta \end{cases} \quad (A1)$$

Map S to the region bounded by the unit circle in the ζ -plane, then the stress function in the ζ -plane is:

$$\begin{cases} \phi_1(z) = \phi_1[\omega(\zeta)] = \phi(\zeta) \\ \psi_1(z) = \psi_1[\omega(\zeta)] = \psi(\zeta) \end{cases} \quad (A2)$$

and:

$$\phi'_1(z) = \phi_1[\omega(\zeta)] = \phi(\zeta) \quad (A3)$$

Thus, the partial differentiation of the solution U of the homogeneous equation can be performed:

$$\frac{\partial U}{\partial x} + i \frac{\partial U}{\partial y} = \phi(\zeta) + \frac{\omega(\zeta)}{\omega'(\bar{\zeta})} \bar{\phi}'(\bar{\zeta}) + \bar{\psi}(\bar{\zeta}) \quad (A4)$$

On the boundary of the unit circle ($\sigma = e^{i\theta}$), it can be obtained as follow:

$$\frac{\partial U}{\partial x} + i \frac{\partial U}{\partial y} = \phi(\sigma) + \frac{\omega(\sigma)}{\omega'(\bar{\sigma})} \bar{\phi}'(\bar{\sigma}) + \bar{\psi}(\bar{\sigma}) = f_1(\sigma) + if_2(\sigma) \quad (A5)$$

where

$$f_1 + if_2 = i \int_{s_0}^s (X_s + iY_s) ds + \frac{\partial V}{\partial x} + i \frac{\partial V}{\partial y} \quad (A6)$$

The particular solution V on the right side of the above equation is not mapped to the ζ -plane. Suppose the points $z = x + iy$ and $z_1 = x_1 + iy_1$ in z -plane are mapped to the points $\zeta = \xi + i\eta = re^{i\theta}$ and $\zeta_1 = \xi_1 + i\eta_1 = re^{i\varphi}$ in ζ -plane, then differentiation of the particular solution V leads to the following equation:

$$\frac{\partial V}{\partial x} - i \frac{\partial V}{\partial y} = -\frac{k}{2\pi\omega'(\zeta)} \int_{S_\zeta} \frac{\tau\omega'(\zeta_1)\bar{\omega}'(\bar{\zeta})}{\zeta_1 - \zeta} R dR d\varphi \quad (A7)$$

while $d\varphi = 1/i \cdot d\zeta_1/\zeta_1$, and substitute it into Equation (A7) to obtain:

$$\frac{\partial V}{\partial x} - i \frac{\partial V}{\partial y} = -\frac{k}{2\pi\omega'(\zeta)} \left(\int_0^r + \int_r^1 \right) \int_{s_0} \frac{\tau\omega'(\zeta_1)\overline{\omega}'(\overline{\zeta})}{\zeta_1(\zeta_1 - \zeta)} R dR d\zeta_1 \quad (\text{A8})$$

In Equation (A8), the first integral in the bracket is performed for $|\zeta_1| < |\zeta|$, and the second integral is performed for $|\zeta_1| > |\zeta|$.

Using the following coordinates relations:

$$dz = e^{i\alpha}|dz|, d\zeta = e^{i\theta}|d\zeta|, e^{iz} = \frac{\zeta}{r} \frac{\omega'(\zeta_1)}{|\omega'(\zeta_1)|} \quad (\text{A9})$$

The relations between the stresses represented by polar coordinates and rectangular coordinates can be obtained as follows:

$$\begin{cases} \sigma_{rr} + \sigma_{\theta\theta} = \sigma_{xx} + \sigma_{yy} \\ \sigma_{\theta\theta} - \sigma_{rr} + 2i\sigma_{r\theta} = \frac{\zeta^2\omega'(\zeta)}{r^2\overline{\omega}'(\overline{\zeta})} (\sigma_{yy} - \sigma_{xx} + 2i\sigma_{xy}) \\ u_r + iu_{\theta} = \frac{\zeta\overline{\omega}'(\overline{\zeta})}{r|\omega'(\zeta)|} (u_x + iu_y) \end{cases} \quad (\text{A10})$$

Simplifying the hub and the LSG as a hollow cylinder, the two concentric circles of the hollow cylinder can be mapped to two concentric circles C_i, C_0 with the radius of r_i, r_0 , respectively, on ζ -plane. By distinguishing subscripts of C_i and C_0 by $m = 1, 2$, the boundary condition of the point σ on C_m can be obtained from Equation (A5):

$$\frac{\partial U}{\partial x} + i \frac{\partial U}{\partial y} = \phi(\sigma) + \frac{\omega(\sigma)}{\overline{\omega}'(\frac{a_m^2}{\sigma})} \overline{\phi}'(\frac{a_m^2}{\sigma}) + \overline{\psi}(\frac{a_m^2}{\sigma}) = [f_1(\sigma) + if_2(\sigma)]_m \quad (\text{A11})$$

Due to its circular dually connected domain, the stress functions $\phi(\zeta)$ and $\psi(\zeta)$ on ζ -plane can be represented by the Laurent series as follows:

$$\begin{cases} \phi(\zeta) = \phi_p(\zeta) + \phi_q(\zeta) = \sum_{n=0}^{\infty} \alpha_n \zeta^n + \sum_{n=1}^{\infty} \alpha_{-n} \zeta^{-n} \\ \psi(\zeta) = \psi_p(\zeta) + \psi_q(\zeta) = \sum_{n=0}^{\infty} \beta_n \zeta^n + \sum_{n=1}^{\infty} \beta_{-n} \zeta^{-n} \end{cases} \quad (\text{A12})$$

The temperature distribution under the axisymmetric steady temperature field is:

$$\tau = A \ln r + B \quad (\text{A13})$$

Under this condition, the particular solution of Equation (11) is obtained as:

$$V = k \left\{ Ar^2 \ln r + (B - A)r^2 \right\} / 4 \quad (\text{A14})$$

By differentiating x, y in Equation (A14), it can be obtained as follow:

$$\frac{\partial V}{\partial x} + i \frac{\partial V}{\partial y} = \frac{k}{2} \left(A \ln r - \frac{A}{2} + B \right) \zeta \quad (\text{A15})$$

According to the Equation (A11), the following can be obtained:

$$\frac{1}{2\pi i} \int_{C_m} \left(\phi(\sigma) + \frac{\omega(\sigma)}{\overline{\omega}'(\frac{a_m^2}{\sigma})} \overline{\phi}'(\frac{a_m^2}{\sigma}) + \overline{\psi}(\frac{a_m^2}{\sigma}) \right) \frac{d\sigma}{\sigma - \zeta} = \frac{1}{2\pi i} \int_{C_m} \left(\frac{\partial V}{\partial x} + i \frac{\partial V}{\partial y} \right) \frac{d\sigma}{\sigma - \zeta} \quad (\text{A16})$$

Its conjugation form is:

$$\frac{1}{2\pi i} \int_{C_m} \left(\bar{\phi} \left(\frac{a_m^2}{\sigma} \right) + \frac{\omega \left(\frac{a_m^2}{\sigma} \right)}{\bar{\omega}'(\sigma)} \phi'(\sigma) + \psi(\sigma) \right) \frac{d\sigma}{\sigma - \zeta} = \frac{1}{2\pi i} \int_{C_m} \left(\frac{\partial V}{\partial x} + i \frac{\partial V}{\partial y} \right) \frac{d\sigma}{\sigma - \zeta} \quad (A17)$$

By complex integration of the inner and outer boundaries according to the Cauchy integral formula and its generalization formula, the following can be obtained:

$$\left\{ \begin{array}{l} \sum_{n=1}^{\infty} \alpha_{-n} \zeta^{-n} + \sum_{n=3}^{\infty} n \alpha_n r_i^{2(n-1)} \zeta^{-n+2} + \sum_{n=1}^{\infty} \beta_n r_i^{2n} \zeta^{-n} = 0 \\ \sum_{n=1}^{\infty} \alpha_n r_i^{2n} \zeta^{-n} + r_i^2 a_1 \zeta^{-1} - r_i^2 \sum_{n=1}^{\infty} n \alpha_{-n} \zeta^{-n-2} + \sum_{n=1}^{\infty} \beta_{-n} \zeta^{-n} = \frac{k}{2} \left(A \ln r_i - \frac{A}{2} + B \right) \frac{a_i^2}{\zeta} \\ \sum_{n=1}^{\infty} \alpha_n \zeta^n + a_1 \zeta + 2r_0^2 a_2 - \sum_{n=1}^{\infty} n \alpha_{-n} r_0^{-2(n+1)} \zeta^{n-2} + r_0 + \sum_{n=1}^{\infty} \beta_{-n} r_0^{-2n} \zeta^n = \frac{k}{2} \left(A \ln r_0 - \frac{A}{2} + B \right) \zeta \\ r_0 + \sum_{n=1}^{\infty} \alpha_{-n} r_0^{-2n} \zeta^n + r_0^2 \sum_{n=2}^{\infty} n \alpha_n \zeta^{n-2} + \sum_{n=1}^{\infty} \beta_n \zeta^n = 0 \end{array} \right. \quad (A18)$$

According to the Equation (A18), the following can be obtained:

$$\left\{ \begin{array}{l} \alpha_1 = \frac{k}{4} \left\{ \frac{A(r_0^2 \ln r_0 - r_i^2 \ln r_i)}{r_0^2 - r_i^2} - \frac{A}{2} + B \right\} \\ \beta_{-1} = -k \left\{ \frac{A r_0^2 r_i^2}{2(r_0^2 - r_i^2)} \ln \frac{r_0}{r_i} \right\} \\ \alpha_n = \beta_{-n} = 0, (n = 2, 3, \dots) \\ \alpha_{-n} = \beta_n = 0, (n = 2, 3, \dots) \end{array} \right. \quad (A19)$$

By substituting Equation (A19) into Equation (A12), the following can be obtained:

$$\left\{ \begin{array}{l} \phi(\zeta) = \frac{k}{4} \left\{ \frac{A(r_0^2 \ln r_0 - r_i^2 \ln r_i)}{r_0^2 - r_i^2} - \frac{A}{2} + B \right\} \zeta \\ \psi(\zeta) = -k \left\{ \frac{A r_0^2 r_i^2}{2(r_0^2 - r_i^2)} \ln \frac{r_0}{r_i} \right\} \frac{1}{\zeta} \end{array} \right. \quad (A20)$$

Then:

$$U = \frac{k}{4} \left\{ \frac{A(r_0^2 \ln r_0 - r_i^2 \ln r_i)}{r_0^2 - r_i^2} - \frac{A}{2} + B \right\} r^2 - k \left\{ \frac{A r_0^2 r_i^2}{2(r_0^2 - r_i^2)} \ln \frac{r_0}{r_i} \right\} \ln r \quad (A21)$$

References

1. Leffler, W.L.; Pattarozzi, R.; Sterling, G. *Deepwater Petroleum Exploration & Production: A Nontechnical Guide*, 2nd ed.; Pennwell Corp: Tulsa, OK, USA, 2011.
2. Wang, W.; Yun, F.; Sun, H.; Wang, L.; Yan, Z.; Wang, G.; Gong, H.; Jiao, K.; Liu, D.; Hao, X. The Research and Experiments on Contact Sealing Theory of the Underwater Clamp Connector. *Machines* **2021**, *9*, 262. [\[CrossRef\]](#)
3. Abid, M.; Ullah, B. Three-dimensional nonlinear finite-element analysis of gasketed flange joint under combined internal pressure and variable temperatures. *J. Eng. Mech.* **2007**, *133*, 222–229. [\[CrossRef\]](#)
4. Abid, M.; Awan, A.W.; Nash, D.H. Determination of load capacity of a non-gasketed flange joint under combined internal pressure, axial and bending loading for safe strength and sealing. *J. Braz. Soc. Mech. Sci* **2014**, *36*, 477–490. [\[CrossRef\]](#)
5. Abid, M.; Khan, N.B. Behavior of gasketed bolted pipe flange joint under combined internal pressure, axial, and bending load: Three-dimensional numerical study. *Proc. Inst. Mech. Eng. Part E J. Process Mech. Eng.* **2018**, *232*, 314–322. [\[CrossRef\]](#)
6. Abid, M.; Nash, D.H.; Javed, S.; Wajid, H.A. Performance of a gasketed joint under bolt up and combined pressure, axial and thermal loading—FEA study. *Int. J. Press. Vessel. Pip.* **2018**, *168*, 166–173. [\[CrossRef\]](#)
7. Zhou, X.J.; Qiu, X.Q.; Zhang, B.; Chen, G.M. Finite Element Analysis of Bolted Flange Joint under Transient Temperature Field. *Press. Vessel Technol.* **2007**, *24*, 8–11, 16.
8. Avanzini, A.; Donzella, G. A computational procedure for life assessment of UHP reciprocating seals with reference to fatigue and leakage. *Int. J. Mater. Prod. Technol.* **2007**, *30*, 33–51. [\[CrossRef\]](#)

9. Omiya, Y.; Sawa, T. Thermal Stress Analysis and the Sealing Performance Evaluation of Bolted Flange Connection at Elevated Temperature. In Proceedings of the ASME 2009 Pressure Vessels and Piping Conference, Prague, Czech Republic, 26–30 July 2009.
10. Omiya, Y.; Sawa, T. Stress Analysis and Sealing Performance Evaluation of Bolted Pipe Flange Connections With Smaller and Larger Nominal Diameter Under Repeated Temperature Changes. In Proceedings of the ASME 2014 Pressure Vessels and Piping Conference, Anaheim, CA, USA, 20–24 July 2014.
11. Sawa, T.; Takagi, Y.; Torii, H. Effect of Material Properties of Gasket on the Sealing Performance of Pipe Flange Connections at Elevated Temperature. In Proceedings of the AMSE Pressure Vessels and Piping Conference, Boston, MA, USA, 19–23 July 2015.
12. Luo, Y.Y.; Liu, X.Y.; Hao, J.; Wang, Z.; Lin, X.M. Research on thermal fatigue failure mechanism of aviation electrical connectors. *Acta Armamentarii* **2016**, *7*, 1266–1274. [\[CrossRef\]](#)
13. Abdullah, O.I.; Schlattmann, J.; Jobair, H.; Beliardouh, N.E.; Kaleli, H. Thermal stress analysis of dry friction clutches. *Ind. Lubr. Tribol.* **2018**, *72*, 189–194. [\[CrossRef\]](#)
14. Xue, J.L.; Chen, X.D.; Fan, Z.C.; Wang, L. Calculation of Leakage Rate for Bolted Flanged Joint during the Long Term Service at High Temperature. In Proceedings of the AMSE Pressure Vessels and Piping Conference, Prague, Czech Republic, 15–20 July 2018.
15. Zhang, L.; Xiao, G. Analysis of the metal-to-metal contact flange joint subject to external bending moment. *Proc. Inst. Mech. Eng. Part E J. Process Mech. Eng.* **2018**, *233*, 234–241. [\[CrossRef\]](#)
16. Barsoum, I.; Barsoum, Z.; Islam, M.D. Thermomechanical Evaluation of the Performance and Integrity of a HDPE Stub-End Bolted Flange Connection. *J. Press. Vessel. Technol.* **2019**, *141*, PVT-18-1259. [\[CrossRef\]](#)
17. Tang, L.P.; He, W.; Zhu, X.H.; Zhou, Y.L. Sealing Performance Analysis of an End Fitting for Marine Unbonded Flexible Pipes Based on Hydraulic-Thermal Finite Element Modeling. *Energies* **2019**, *12*, 2198. [\[CrossRef\]](#)
18. Chen, J.; Ding, X.; Zhang, W.; Yan, R.; Sun, B. Fractal prediction model for the thermo-elastic normal contact stiffness of frictional interfaces in dry gas seals. *J. Vib. Shock* **2020**, *39*, 257–263, 284. [\[CrossRef\]](#)
19. Li, Y.; Zhao, H.; Wang, D.; Li, X.; Zhao, T. Studied on Thermal Coupling and Sealing Characteristics of the Subsea Wellhead Connector Metal Seal. *IOP Conf. Ser. Earth Environ. Sci.* **2020**, *527*, 012005. [\[CrossRef\]](#)
20. Wang, Q.; Hu, Y.P.; Ji, H.H. Leakage, heat transfer and thermal deformation analysis method for contacting finger seals based on coupled porous media and real structure models. *Proc. Inst. Mech. Eng. Part C J. Mech. Eng. Sci.* **2020**, *234*, 2077–2093. [\[CrossRef\]](#)
21. Zhang, H.; You, H.; Lu, H. Mechanical-thermal sequential coupling simulation for predicting the temperature distribution of seal rings—Part I. *Seal. Technol.* **2020**, *2020*, 5–8. [\[CrossRef\]](#)
22. Zhang, H.; You, H.; Lu, H. Mechanical-thermal sequential coupling simulation for predicting the temperature distribution of seal rings—Part II. *Seal. Technol.* **2020**, *2020*, 5–9. [\[CrossRef\]](#)
23. Zeng, W.; Xue, Y.X.; Sun, Y.H.; Zhao, J.; Xie, H.; Ren, T. Sealing reliability assessment of deep-water oil and nature gas pipeline connector considering thermo-mechanical coupling. *Proc. Inst. Mech. Eng. Part M J. Eng. Marit. Environ.* **2021**, *1*, 196–208. [\[CrossRef\]](#)
24. Liu, H.; Rao, Z.; Pang, R.; Zhang, Y. Research on Thermal Characteristics of Ball Screw Feed System Considering Nut Movement. *Machines* **2021**, *9*, 249. [\[CrossRef\]](#)
25. Zhang, Y.; Wang, G.; Pan, X.; Li, Y. Calculating the Load Distribution and Contact Stress of the Disposable Harmonic Drive under Full Load. *Machines* **2022**, *10*, 96. [\[CrossRef\]](#)
26. Song, J.H.; Kang, S.W.; Kim, Y.J. Optimal design of the disc vents for high-speed railway vehicles using thermal-structural coupled analysis with genetic algorithm. *Proc. Inst. Mech. Eng. Part C J. Mech. Eng. Sci.* **2021**, *11*, 09544062211059112. [\[CrossRef\]](#)
27. Wen, M. An Analysis of the Coupling Between Temperature and Thermal Stress of Disc Brakes Based on Finite Element. *Int. J. Heat Technol.* **2021**, *39*, 1819–1827. [\[CrossRef\]](#)
28. Gao, L.; Zhao, X.C.; Wang, B. Thermostructural Responses of Metallic Lattice-Frame Sandwich Structure for Hypersonic Leading Edges. *J. Thermophys. Heat Transf.* **2021**, *35*, 708–714. [\[CrossRef\]](#)
29. Ma, H.Q.; Jia, J.W.; Ding, R.X.; Luo, X.M.; Peng, D.G.; Hou, C.Q.; Wang, G.; Zhang, Y.J. Numerical investigation on fatigue life of aluminum brazing structure with fin-plate-side bar under the low temperature thermal-structure cyclic stress. *Eng. Fail. Anal.* **2022**, *131*, 18. [\[CrossRef\]](#)
30. Hetnarski, R.B. *Encyclopedia of Thermal Stresses*; Springer: Dordrecht, The Netherlands, 2014.
31. Cheng, X.; Peng, W.; Sun, L.; Li, X.; Yin, Z. Analysis of the fracture pattern and failure time of LNG outer tanks under the action of thermal load. *Nat. Gas Ind.* **2015**, *35*, 103–109. [\[CrossRef\]](#)
32. Lu, G.C.; Chu, G.N. Research of Directional Stress Distribution on the Fully Plastic Bulged Pressure Hull. *Adv. Mater. Res.* **2015**, *1082*, 412–415. [\[CrossRef\]](#)