Contact mechanical behaviors of radial metal seal for the interval control valve in intelligent well: Modeling and theoretical study

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Abstract
Intelligent wells equipped with the interval control valve (ICV), which have the potential applications to significantly improve oil production or control the water production of wells and fields, are more and more widely utilized in the oilfields. The radial metal seal of an ICV is intended to isolate the flow of produced fluids between the casing annulus and the well tubing. The contact mechanical behaviors of the radial metal seal are essential to the structural design and reliability of ICVs in the intelligent well. In this paper, the two-dimensional axisymmetric assembly of the radial metal seal and the closure member is abstracted as a combination of interference fit model and cantilever beam model. Based on that, the related theoretical model, between the radial metal seal’s contact stress and its independent variables, has been derived. Next, numerically simulations of contact stress between the radial metal seal and the closure member for different independent variables are conducted by the finite element method (FEM) to validate the theoretical model. It is found that the mean absolute value of relative error between the numerical and theoretical results of the contact stress is approximately 10%. Additionally, the structural parameters of the radial metal seal are proposed with the absolute value of relative error <10%. Furthermore, the value of the contact stress based on the cantilever beam model can be used to calculate an approximation theoretical solution. In summary, the proposed theoretical model provides a foundation for the design, improvement, and selection of radial metal seal under the known oilfield conditions.

KEYWORDS
contact mechanical behaviors, intelligent well, interval control valve, radial metal seal

1 | INTRODUCTION

Intelligent well technology is one of the most significant breakthroughs in oilfield production technologies in recent years. And it has been widely applied to enhance the production performance, realize the real-time control and production optimization, and maximize the ultimate recovery of the reservoirs in oilfields. An intelligent well completion system mainly consists of the interval control valves (ICVs), downhole gauges, and multi-port feed through
packers, which is used to segregate the sections of the wellbore. Furthermore, the packers and ICVs are installed inside the intelligent well as shown in Figure 1. Different from the other downhole tools, ICVs are required to work under some conditions, such as the high pressure loading or unloading. Conventional downhole sleeve seals are devised to handle a range of differential pressures between the well tubing and casing annulus, and the first-generation ICVs are evolved from the sliding sleeve technology. While an extra locking key mechanism and a boost piston are needed to reinforce the seal, the first-generation ICVs have been replaced by the second-generation ICVs. In addition, the second-generation ICV retains most components of the first-generation ICV with modifications made mainly in the upper and lower seat configurations. The metal seal in the second-generation ICVs is a radial metal seal, which is utilized to improve the seal capability.

Typically, the radial metal seal is a semi-dynamic seal and moves occasionally. Different from the reciprocating engine or bearing seal on a rotating shaft, the semi-dynamic seal provides a seal while translating sliding components within the oil-gas well production. From a view of seal function, the static seal is needed for an ICV. However, a relative motion between the components of an ICV is existing. Thus, the radial metal seal should be defined as semi-dynamic seal instead of the semi-static seal. Previous studies have demonstrated that the semi-dynamic seal is mainly made of the rubber, and few of them is the metal. The composition of the radial static metal seal is the same as that of radial semi-dynamic metal seal; therefore, the seal mechanism and research method of the radial static metal seal can be used as a reference for designing and exploring the radial semi-dynamic metal seal.

The metal seals can meet these requirements of safety working conditions in the downhole; thus, they are widely used in some oil-gas well operations. Specifically, the metal seals utilize a flexible metal seal as their external seal element to contact the internal sealed element, and then form a tight and durable seal, which can withstand the high pressure, high temperature, and corrosive-fluid conditions in oil-gas wells. Zhao et al. established the theoretical relationships between the contact stress of subsea X-tree wellhead connector sealer and its independent variables. The analytical results were consistent with the simulation results well via the FEM; however, the effect of plastic deformation is not taken into account. Peng et al. put forward a two-dimensional axisymmetric model of K-shaped seal used in the tubing hanger via the ABAQUS software, but the quantitative relationship of the factors affecting the contact stress of the K-type seals had not been performed. Wei et al. analyzed the seal mechanism of seal face for a mechanical connector of a submarine pipeline and obtained the critical conditions of the seal based on the superposition principle of elasticity. But it was not rigorous to judge whether the seal was formed by observing the seal indentation on the surface of the pipeline. Cui et al. investigated the seal principle of metal zonal isolation tools used in downhole by the FEM. The study only proved the feasibility of metal seal technology in seal the large annular, but the seal mechanism was not revealed. Qin et al. and Zhang et al. developed a finite element model for the performance of K-type metal seal rings on subsea Christmas tree tubing hanger under the practical working conditions with the ABAQUS software. These studies only qualitatively analyzed the effects of initial interference, working pressure, and working temperature on contact mechanical behavior. Chao et al. analyzed the seal mechanism and working principle of annular metal seal assembly used in the subsea wellhead system combining with the finite element analysis results, but the mathematics model was not put forward. As for the application of metal seals in other fields, Li et al. established the two-dimensional axisymmetric model of the metal seal U-ring by the ABAQUS software, but the effect of structural parameters of U-ring on the seal performance had not been studied. Many studies have investigated the seal performance of other special-shaped metal seals by the FEM, but no mathematical model was proposed.

There are also many other designs and studies for the dynamic seal of rotating shafts. Lee et al. explored the seal face width and contact pressure of the seal lip under the various interference fits between the shaft and the seal by the FEM and the numerical results are consistent with the experimental results well. It was found that the influence of the thermal deformation on the width of the contact zone, and the contact pressure was small and can be neglected from the

![FIGURE 1](image-url) The schematic diagram of intelligent well for four separate inflow zones and the details of zonal isolation (packers) and ICVs.
finite element analysis. Although the radial lip seal is a dynamic seal in radial direction, its research method and test device can be used as a reference for the study of semi-dynamic metal seal.

In this paper, the structure of radial metal seal of the second-generation ICV proposed by Well Dynamics Inc has been improved, and the related theoretical model of its contact mechanical behaviors also has been established. The theoretical and numerical analyses are carried out, which provides methods and means for the selection of application types and the improvement of structural parameters in intelligent completion operation. In addition, the related contents are organized as follows: in Section 2, the structure and principle of a radial metal seal are illustrated. In Section 3, the related theoretical model of the radial metal seal has been derived, and corresponding seal mechanism has been analyzed. In Section 4, the numerical verifications and error analyses are conducted. In Section 5, the conclusions and the further research in future work are summarized and discussed.

2 CONFIGURATION OF THE RADIAL METAL SEAL

In order to isolate the flow of produced fluids between the casing annulus and the well tubing, the radial metal seal, which includes a metal and nonmetal seal elements, is applied between the ICV housing and the closure member as shown in Figure 2. The nonmetal seal element can be used to energize the metal seal element in any directions in response to the differential pressure applied to the radial metal seal. Both the metal and nonmetal seal elements contact one of the ICV housing and the closure member, when the closure member blocks flow through the ICV housing. As the closure member is moved to relieve the differential pressure, the metal seal element will continue to seal against the differential pressure until the nonmetal seal element no longer seals between the ICV housing and the closure member.

As shown in Figure 3, only the micro-convex body contacts the seal face between the metal seal element and the closure member before the differential pressure is applied. If the deformation on the seal face is insufficient, two seal faces will not fully contact, which will result in generating the micro-leak path. The micro-leakage paths will be blocked by reducing the height of micro-peaks and the depth of micro-pits, or increasing the diameter of micro-peaks, which can realize a reliable seal.

The material properties of the radial metal seal are listed in Table 1. The metal seal element is made of Nickel-based Alloy Inconel 718. The alloy has been widely used in many fields, such as aviation, aerospace, and petrochemical, due to its excellent mechanical and processing properties. These properties include the good high-temperature strength, excellent creep resistance and fatigue resistance, excellent processing, and welding performance. Additionally, the nonmetal seal element is made of the fluorine rubber, with the merits of the high heat resistance, oil resistance, oxidation resistance, and corrosion resistance.

3 THEORETICAL MODEL OF THE RADIAL METAL SEAL

The nonmetal seal element exhibits the complicated nonlinear behaviors including the hysteresis, viscoelasticity, and stress softening effect. Thus, the loads applied by the nonmetal seal element on the metal seal element are unevenly distributed. And the influence of difference between the surface temperature and the bottom hole temperature on contact stress of the radial metal seal is ignored. For simplicity, the loads distributed on the metal seal element are assumed uniform. According to the structure, stress, and boundary conditions, the metal seal element is simplified as an axisymmetric structure with interference fit. As shown in Figure 4, the upper part of the structure can be abstracted as a mandrel and a sleeve model with the interference, δ, and the lower part is abstracted as a cantilever beam model.

When the influence of gravity is neglected, the equation of the interference fit between the closure member and the metal seal element is expressed by the elasticity mechanics method:

\[ \frac{\partial \sigma_r}{\partial r} + \frac{\partial \tau_{r\theta}}{\partial z} + \frac{\sigma_r - \sigma_\theta}{r} = 0 \]  

(1)

\[ \frac{\partial \sigma_\theta}{\partial z} + \frac{\partial \tau_{r\theta}}{\partial r} + \frac{\tau_{r\theta}}{r} = 0 \]  

(2)

where \( \sigma_r, \sigma_\theta, \sigma_z \), and \( \tau_{r\theta} \) are the stress components along \( r \) axis, \( \theta \) axis, \( z \) axis, and tangential direction in the cylindrical coordinate system, respectively. Furthermore, the four strain components (\( \varepsilon_r, \varepsilon_\theta, \varepsilon_z, \gamma_{r\theta} \)) can be expressed as:

\[ \varepsilon_r = \frac{\partial u_r}{\partial r} \]  

(3)

\[ \varepsilon_\theta = \frac{u_r}{r} \]  

(4)

\[ \varepsilon_z = \frac{\partial w}{\partial z} \]  

(5)

\[ \gamma_{r\theta} = \frac{\partial u_r}{\partial z} + \frac{\partial w}{\partial r} \]  

(6)

\[ \theta = \frac{\partial u_r}{\partial r} + \frac{u_r}{r} + \frac{\partial w}{\partial z} \]  

(7)
where \( u_r \) and \( w \) are the displacements in \( r \) axis and \( z \) axis of the cylindrical coordinate system, respectively. By substituting Equations (3)-(7) into Equations (1) and (2), the followings can be obtained:

\[
\frac{E}{2(1+\mu)} \left( \frac{\mu}{1-2\mu} \frac{\partial e}{\partial r} + \nabla^2 u_r - \frac{u_r}{r^2} \right) = 0
\]

(8)

\[
\frac{E}{2(1+\mu)} \left( \frac{\mu}{1-2\mu} \frac{\partial e}{\partial r} + \nabla^2 w \right) = 0
\]

(9)

where \( \nabla^2 = \frac{\partial^2}{\partial r^2} + \frac{1}{r} \frac{\partial}{\partial r} + \frac{\partial^2}{\partial z^2} \) is the Laplace operator. By using Love displacement function \( \zeta(r,z) \), the displacements in \( r \) axis and \( z \) axis directions are respectively expressed as:

\[
u_r = -\frac{1}{2G} \frac{\partial^2 \zeta}{\partial r \partial z}
\]

(10)

\[
w = \frac{1}{2G} \left[ 2(1-\mu)\nabla^2 - \frac{\partial^2}{\partial z^2} \right] \zeta
\]

(11)

Suppose \( \zeta(r,z) = A_1 z^3 + A_2 z^2 + A_3 \ln r \), where coefficients \( A_1 \), \( A_2 \) and \( A_3 \) can be solved from the boundary conditions of the closure member:

\[
r = a; \sigma_r = p_1, \quad \tau_{rz} = 0
\]

(12)

**TABLE 1** Material performance of the radial metal seal at 20°C

| Parameter               | Symbol | Metal seal element | Nonmetal seal element |
|-------------------------|--------|--------------------|-----------------------|
| Elasticity modulus (GPa)| \( E \) | 203                | 7.84 \times 10^{-3}   |
| Shear modulus (GPa)     | \( G \) | 78.1               | 0.49                  |
| Poisson ratio           | \( \mu \) | 0.3                |                       |
| Yield strength (MPa)    | \( \sigma_y \) | 1030              |                       |
| Ultimate strength (MPa) | \( \sigma_u \) | 1270              | 20                    |
| Density (kg·m⁻³)        | \( \rho \) | 8240              | 1900                  |

**FIGURE 2** The schematic diagram of the radial metal seal of an ICV in the intelligent well

**FIGURE 3** A, The micro-morphology of the seal face at the initial contact; B, micro-morphology of the seal face during the seal contact
where $a$ is the internal radius of the closure member, $b$ is the outer radius of the closure member, $p_1$ is the external load on the inner diameter, and $p$ is the contact stress on the seal face. According to Equations (12) and (13), the displacement in $r$ axis of the closure member can be expressed as:

$$ r = b; \sigma_r = p, \quad \tau_{rz} = 0 \quad (13) $$

$$ z = 0: z = z_c, \quad \sigma_z = q_1 \quad (14) $$

where $c$ and $E_2$ are the outer radius and elasticity modulus of the metal seal element, respectively. The conditions of deformation compatibility, between the closure member and the metal seal element, are as follows:

$$ |u_1^r|_{r=b} - |u_2^r|_{r=b} | = \delta \quad (17) $$

Substitute Equations (15) and (16) into Equation (17), the analytical solution of contact stress for the interference fit model between the closure member and the metal seal element can be obtained:

$$ p = \frac{b \left( \delta E_1 E_2 - b E_1 \mu q_2 + b E_2 \mu q_1 \right) \left( b^2 - a^2 \right) \left( c^2 - b^2 \right) - 2 b p_1 a^2 E_2 \left( c^2 - b^2 \right) - 2 b p_2 c^2 E_1 \left( b^2 - a^2 \right)}{b \left[ (1-\mu) b^2 + (1+\mu) a^2 \right] E_2 \left( c^2 - b^2 \right) + b \left[ (1-\mu) b^2 + (1+\mu) c^2 \right] E_1 \left( b^2 - a^2 \right)} \quad (18) $$

When the length–height ratio of the cantilever beam is more than 5, the cantilever beam can be regarded as an Euler–Bernoulli beam, regardless of the shear deformation:

$$ \frac{d^2 w}{dx^2} = \frac{M(x)}{EI} \quad (19) $$

where $M(x) = -M'' - 1/2 q_2 (l-x)^2 + F'' (l-x) \sin \beta$, $\beta$ and $l$ are the inclination angle and length of the cantilever beam, respectively. Substitute the boundary conditions $w(0) = 0$, $\varphi(0) = 0$ and $\varphi(l) = 0$ into Equation (19), the displacement in $z$ axis direction, $w$ can be obtained:

$$ w = \frac{-q_2 (l-x)^4 + 4 F'' (l-x)^3 \sin \beta - 6 (F'' \sin \beta - 8 q_2 l^2) x^2 + \left( \frac{l^2}{2} F'' \sin \beta - \frac{q_2 l}{6} \right) x + q_2 l^4 - 4 F'' l^3 \sin \beta}{24 E I} \quad (20) $$

**FIGURE 4** The structural and mechanical analysis of the radial metal seal of an ICV
where \( I = \frac{b_0 l^3}{12} \) and \( b_0 = 1 \) are the inertia moment and the cross-section width of the cantilever beam, respectively. Thus, the horizontal displacement of the cantilever beam end face, \( u_1 \), is:

\[
u_1 = \frac{w(l)}{\sin \beta} = \frac{q_2 l^4 - 4F'' l^3 \sin \beta - 6(F'' l \sin \beta - 8q_2 l^3)l^2 + (12F'' l^2 \sin \beta - 4q_2 l^4)l}{24EI \sin \beta}
\]

Equation (22) can be derived from Equation (21) as follows:

\[
F'' = \frac{q_2 l^4}{2 \sin \beta} + E u_1 \left( \frac{l}{l} \right)^3
\]

Because the cross-section height of the metal seal element is much less than its radial dimension, \( F'' \) is considered to be approximately equal to the contact force, \( F_{En} \) on the seal face, namely, \( F_{En} = F'' \). Then, the analytical solution of contact stress at the seal face is as follows:

\[
\sigma_{En} = \frac{F_{En}}{h}
\]

where \( h \) is the width of actual seal face. With the length–height ratio of the cantilever beam <5, the Timoshenko beam model is applied to evaluate the influence of shear deformation on the deflection of the cantilever beam.⁴⁸

\[
dw = \frac{-6M'' + q(l-x)^3 + 3F''(l-x)^3 \sin \beta}{6EI} + \frac{\alpha_2 x - a_2 F''}{GA} + C_1
\]

where \( \alpha_2 \) is the shear coefficient and \( G \) is the shear modulus of the metal seal element. Substitute the boundary conditions \( w(0) = 0, \varphi(0) = 0, \) and \( \varphi(l) = 0 \) into Equation (24), the following results can be obtained:

\[
F_{Tm} = \frac{q_2}{2 \sin \beta} + u_1 \left( \frac{l}{l} \right)^3
\]

The horizontal displacement of the cantilever beam end face can also be deduced:

\[
u_1 = \frac{w(l)}{\sin \beta} = \frac{q_2 l^4 - 3F'' l \sin \beta}{12EI \sin \beta} + \frac{q_2 l^4 + 3 \sin \beta F'' l^3}{6EI \sin \beta} + \frac{\alpha_2 F'' l}{GA} + \frac{q_2 l^4 - 4 \sin \beta F'' l^3}{24EI \sin \beta}
\]

Considering the influence of shear deformation, the analytical solutions of contact force and contact stress at the seal face are as follows:

\[
\sigma_{En} = \frac{F_{En}}{h}
\]

\[
\sigma_{Tm} = \frac{F_{Tm}}{h}
\]

Then, the total analytical solution of contact stress is as follows:

\[
\sigma_1 = \varphi + \sigma_{En}
\]

\[
\sigma_2 = \varphi + \sigma_{Tm}
\]

It can be inferred from Equations (29) and (30) that there is a positive correlation between the contact stress and the working pressure, the initial interference, the horizontal displacement of the cantilever beam end face, and the cross-section height of the metal seal element. And there is a negative correlation among the contact stress, the inclination angle of the cantilever beam, and the actual seal face width.
4 | NUMERICAL ANALYSES AND VALIDATION

Finite element method is a procedure for obtaining numerical solution of boundary value problems, which are sets of ordinary differential equations whose solution is subject to certain boundary conditions. The FEM process is shown in Figure 5. The structure of the metal seal element and the closure member is axisymmetric, and a two-dimensional finite element model is established. The outer surface of metal seal element is fully constrained by the fixed constraints. The working pressure is imposed on both the upper and lower parts of the metal seal element. The displacement in r axis direction is applied to the closure member according to the initial interference. Two analysis steps are established in FEM: the first step is to apply the displacement in r axis direction to the closure member, and the second step is to apply the working pressure to the metal seal element. Furthermore, the mesh element type of the metal seal element is CAX4R, and the approximate global size is 0.5 mm. The elements on the seal face are refined to 0.01 mm to improve the accuracy of results. The mesh size of the closure member is equal to 285.3, 356.3, 416.8, 461.7, 508.1, and 547.6 MPa, respectively. It can be found that the maximum value of contact stress, the average value of contact stress, the horizontal displacement of the cantilever beam end face $u_1$, and the seal face width $h$ from the FEM results. For instance, when $q_2=40$ MPa, $\delta=0.05$ mm, $\beta=79^\circ$, $l=10$ mm, $t=2$ mm, $a=39$ mm, $b=49$ mm and $c=53.3$ mm, $\sigma_{\text{max}}$ and $u_1$ of the metal seal face can be obtained in the visualization module of ABAQUS, which are 461.7 MPa and 0.054 mm, respectively (Table 2). By collecting the contact stress value of each node on the seal face, the contact stress distribution along the true seal face can be obtained. And so $\sigma_{\text{ave}}$ is equal to 352.63 MPa and $h$ is equal to 0.84 mm. By substituting the above parameter and material parameter values into Equations (18), (23), (28), (29), and (30), $p$, $\sigma_{\text{ave}}$, $\sigma_{\text{ave}}$, $\sigma_1$, and $\sigma_2$ can be calculated, which are 12.48, 345.48, 307.95, 357.97, and 320.43 MPa, respectively. The theoretical solution values of the contact stress can be obtained by changing an independent variable in turn. Then, the relationship curves between the contact stress's theoretical solutions ($\sigma_1$ and $\sigma_2$) and independent variables, between the contact stress's numerical solutions ($\sigma_{\text{max}}$ and $\sigma_{\text{ave}}$) and independent variables, are drawn, and the absolute values of relative error between the theoretical and numerical solutions of the contact stress are calculated based on Equations (31) and (32).

$$RE_1 = \frac{\sigma_{\text{ave}} - \sigma_1}{\sigma_{\text{ave}}} \times 100\%$$  \hspace{1cm} (31)

$$RE_2 = \frac{\sigma_{\text{ave}} - \sigma_2}{\sigma_{\text{ave}}} \times 100\%$$  \hspace{1cm} (32)

4.1 | The relationship between contact stress and working pressure

With the aim to study the relationship between the contact stress and the working pressure $q_2$, the structural parameters of the closure member and the metal seal element remain unchanged, and the working pressure $q_2$ is changed from 0 to 60 MPa. As shown in Figure 6A, when the working pressure is changed from 10 to 60 MPa with the increment of 10 MPa, $\sigma_{\text{max}}$ is equal to 285.3, 356.3, 416.8, 461.7, 508.1, and 547.6 MPa, respectively. It can be found that $\sigma_{\text{max}}$ increases with the increase working pressure $q_2$. This demonstrates that the metal seal element has the self-tightening function, which can ensure the reliability of the radial mental seal of the ICV. As observed from Figure 6A when the working pressure is changed from 10 to 60 MPa, the seal face width is 0.50, 0.64, 0.74, 0.84, 0.93, and 1.00 mm, respectively. It can be noticed that the width of the seal face is increasing with the increase of the working pressure $q_2$. The increasing seal face width $h$ can effectively reduce the leakage on the seal face. As shown in Figure 6B, $\sigma_{\text{max}}$, $\sigma_{\text{ave}}$, $\sigma_1$, and $\sigma_2$ increase with the rise in the working pressure $q_2$. The mean absolute value of relative error between $\sigma_1$ and $\sigma_{\text{ave}}$ is 7.09%. It is evident that the results of the theoretical model are in good agreement with the numerical solution of contact stress. Thus, the relationship between the contact stress and the working pressure is numerical validated.

| Parameter (unit) | $q_2$ (MPa) | $\delta$ (mm) | $\beta$ (°) | $h$ (mm) | $l$ (mm) | $t$ (mm) | $a$ (m) | $b$ (mm) | $c$ (mm) |
|----------------|-------------|-------------|-------------|---------|---------|---------|---------|---------|---------|
| #1            | 0-60        | 0.05        | 79          | 0.30-1.00| 10.00   | 2.0     | 39      | 49      | 52.9    |
| #2            | 40          | 0.01-0.14   | 79          | 0.73-0.81| 10.76   | 2.0     | 39      | 49      | 52.9    |
| #3            | 60          | 0.05        | 68-88       | 0.96-1.01| 10.00-10.76| 2.0     | 39      | 49      | 52.9    |
| #4            | 60          | 0.1         | 79          | 0.58-0.84| 10.00   | 2.0     | 39      | 49      | 52.9    |
| #5            | 50          | 0.1         | 79          | 0.94-1.05| 7.10-14.30| 2.0     | 39      | 49      | 52.6-53.3|
| #6            | 50          | 0.1         | 79          | 1.03-1.06| 11.20   | 1.5-2.6| 39      | 49      | 52.9    |
| #7            | 60          | 0.05        | 79          | 1.00     | 10.00   | 2.0     | 30-39   | 49      | 52.9    |
| #8            | 60          | 0.05        | 79          | 1.07     | 10.00   | 2.0     | 39      | 47-51   | 52.9    |
| #9            | 60          | 0.05        | 79          | 0.98-1.11| 10.00   | 2.0     | 39      | 49      | 51.5-54.6|
4.2 The relationships between contact stress and initial interference, between contact stress and horizontal displacement of the cantilever beam end face

Keeping other independent variables constant, the initial interference between the closure member and the metal seal element is changed from 0.01 to 0.14 mm with the increment of 0.01 mm to investigate the relationship of contact stress and initial interference, and the relationship of contact stress and horizontal displacement of the metal seal face. In Figure 7A, when the initial interference is changed from 0.11 to 0.14 mm with the increment of 0.01 mm, $\sigma_{\text{max}}$ is equal to 556.7, 604.4, 956.5, and 1278 MPa respectively. And it can be found that with the increase of the initial interference, the actual seal face location moves to one end of the metal seal face. When the initial interference is not <0.12 mm, the stress concentration appears at one end of the seal face. And the greater the initial interference is, the more significant the stress concentration is, which indicates that the initial interference should be reasonably designed to avoid the stress concentration. As shown in Figure 7B that $\sigma_{\text{max}}$, $\sigma_{\text{ave}}$, $\sigma_1$, and $\sigma_2$ increase as the initial interference increases. The mean absolute value of relative error between $\sigma_1$ and $\sigma_{\text{ave}}$ is 6.09%. Therefore, the validation between the contact stress-initial interference relationship is positive and acceptable.

4.3 The relationship between contact stress and sine of the cantilever beam inclination angle

The relationship between the contact stress and the sine of the cantilever beam inclination angle is investigated by changing the cantilever beam inclination angle from 68° to 88° with the increment of 1° and keeping other parameters constant. In Figure 8A, it can be found that with the increase of the cantilever beam inclination angle, the seal face location changes. From Figure 8B, it can be found that $\sigma_{\text{max}}$, $\sigma_{\text{ave}}$, $\sigma_1$, and $\sigma_2$ decrease with the increasing sine of the cantilever beam inclination angle. Moreover, the deviation between the $\sigma_1 - \sin\beta$ and $\sigma_{\text{ave}} - \sin\beta$ curves is very small and the mean absolute value of relative error between $\sigma_1$ and $\sigma_{\text{ave}}$ is 3.08%, which shows that the theoretical solution of the contact stress is in good agreement with the numerical solution of the contact stress.

4.4 The relationship between contact stress and seal face width

To explore the relationship between the contact stress and the seal face width, only the seal face width is changed and other parameter values are maintained. In Figure 9A, when the seal face width is changed from 0.61 to 0.84 mm, the corresponding $\sigma_{\text{max}}$ are 586, 553.1, 531.3, 502, 486.3, 464.4, and 449.3 MPa respectively. It can be found that with the increase of the seal face width, the position and magnitude of $\sigma_{\text{max}}$ on the seal face change gradually. Figure 9B indicates that $\sigma_{\text{max}}$, $\sigma_{\text{ave}}$, $\sigma_1$, and $\sigma_2$ decrease almost linearly with the increase of the seal face width. The mean absolute value of relative error between $\sigma_1$ and $\sigma_{\text{ave}}$ is 4.69%. Thus, it can be concluded that the validation with the contact stress and the seal face width relationship is confirmed.

4.5 The relationships between contact stress and cantilever beam length, between contact stress and length–height ratio of the cantilever beam

The relationships between contact stress and cantilever beam length, between contact stress and length–height ratio of the
A cantilever beam can be established by changing the cantilever beam length from 7.1 to 14.3 mm and keeping other parameters constant. It can be found in Figure 10A that when the cantilever beam length is changed from 7.1 to 14.3 mm, $\sigma_{\text{max}}$ equals to 545.8, 524.7, 524.7, 529.6, 542.2, 559.4, 563.9, 576.7, 584.3, 593.6, 601.6, 614.4, 634.7, and 654.8 MPa, respectively. And it can be drawn that with the increase in the cantilever beam length, the location and magnitude of $\sigma_{\text{max}}$ on the seal face change gradually, and the variation of seal face width is not obvious. It can be seen from Figure 10B that $\sigma_{\text{max}}$, $\sigma_{\text{ave}}$, $\sigma_1$, and $\sigma_2$ decrease firstly and then increase with the increase of the cantilever beam length. As can be seen from Equation (22), the contact force is inversely proportional to the cube of the cantilever beam length and is proportional to the first power of the cantilever beam length. So when the length of cantilever beam is relatively short, the cubic term of cantilever beam length dominates in the contact stress, and with the increase of cantilever beam length, the first power of the cantilever beam length dominates. The mean absolute value of relative error between $\sigma_1$ and $\sigma_{\text{ave}}$ is 10.59% and the $\sigma_1-l$ and $\sigma_{\text{ave}}-l$ curves show good following performance, which shows that the validation with the contact stress-the length of cantilever beam relationship is acceptable.

4.6 The relationships between contact stress and cross-section height of the metal seal element, between contact stress and height-length ratio of the cantilever beam

To study the relationships between contact stress and cross-section height of the metal seal element, between contact stress
and height–length ratio of the cantilever beam, the cross-section height of the metal seal element is changed between 1.5 and 2.6 mm with the increment of 0.1 mm and other parameters are maintained constant. In Figure 11A, it can be found that with the increase of the cross-section height of the metal seal element and the height–length ratio of the cantilever beam, \( \sigma_{\text{max}} \) increases gently, and the width and position of the seal face are nearly constant. From Figure 11B, it can be deduced that \( \sigma_{\text{max}} \), \( \sigma_{\text{ave}} \), \( \sigma_1 \), and \( \sigma_2 \) increase with the increasing cross-section height of the metal seal element and the height–length ratio of the cantilever beam. When the height–length ratio of the cantilever beam increases to 0.1958 the deviation between \( \sigma_{\text{ave}} \) and \( \sigma_1 \) is bigger than the deviation between \( \sigma_{\text{ave}} \) and \( \sigma_2 \). It shows that when the effect of shear deformation on the deflection of the cantilever beam is unable to be neglected the theoretical solution derived from the Timoshenko beam model is closer to the numerical solution. The mean absolute value of relative error between \( \sigma_{\text{ave}} \) and \( \sigma_1 \) is 9.54%. It can be concluded that the simulated and theoretical results are in good agreement with each other and the theoretical relationship between the cross-section height of the metal seal element and the contact stress is verified.

### 4.7 The relationship between contact stress and internal radius of the closure member

Changing the internal radius of the closure member from 36 to 42 mm with the increment of 0.5 mm and keeping other parameters constant, the relationship between the contact stress and the internal radius of the closure member is investigated. In Figure 12A, it can be drawn that with the increase of the internal radius of the closure member, \( \sigma_{\text{max}} \), the width and position of the seal face remain constant. From Figure 12B, it
can be found that $\sigma_{\text{max}}$, $\sigma_{\text{ave}}$, $\sigma_1$, and $\sigma_2$ almost remain constant with the increasing internal radius of the closure member, which shows that the internal radius of the closure member has little effect on the contact stress. The mean absolute value of relative error between $\sigma_1$ and $\sigma_{\text{ave}}$ is 4.78%, which shows that the validation with the contact stress-internal radius of the closure member relationship is positive and acceptable.

4.8 The relationship between contact stress and outer radius of the closure member

Changing the outer radius of the closure member from 47 to 51 mm and keeping other parameters unchanged to study the relationship between the contact stress and the outer radius of the closure member. In Figure 13A, it can be drawn that with the increase of the closure member outer radius, $\sigma_{\text{max}}$, the width and position of the seal face remain constant. From Figure 13B, it can be found that $\sigma_{\text{max}}$, $\sigma_{\text{ave}}$, $\sigma_1$, and $\sigma_2$ almost remain constant with the increasing outer radius of the closure member. It shows that the outer radius of the closure member has little effect on the contact stress. The mean absolute value of relative error between $\sigma_1$ and $\sigma_{\text{ave}}$ is 7.23%, which shows that the relationship between the outer radius of the closure member and the contact stress is verified.

4.9 The relationship between contact stress and outer radius of the metal seal element

Changing the outer radius of the metal seal element from 51.5 to 54.1 mm and keeping all other parameters the same to explore the relationship between the contact stress and the metal seal element outer radius. In Figure 14A, it can be found that with the increase in the outer radius of the metal seal element, $\sigma_{\text{max}}$ decreases slowly, and the seal face width is almost unchanged. From Figure 14B, it can be drawn that $\sigma_{\text{max}}$, $\sigma_{\text{ave}}$, $\sigma_1$, and $\sigma_2$ slightly decrease with the increasing outer radius of the metal seal element. The mean absolute value of relative error between $\sigma_1$ and $\sigma_{\text{ave}}$ is 6.4%, which shows that the validation with the contact stress-outer radius of the metal seal element relationship is positive and acceptable.

4.10 Error analysis

On the one hand, when the working pressure is applied on the nonmetal seal element, the contact stress distribution along the interface between the nonmetal seal element and metal seal element is shown in Figure 15A. It can be seen that the contact stress distribution along the interface is approximately uniform. While when the working pressure is directly applied on the metal seal element, the pressure distribution along the metal seal element is completely uniform. When the initial interference is 0.1 mm, the average relative error between the two situations is shown in Figure 15B, which indicates that the average relative error decreases exponentially with the increase of the working pressure. Therefore, in Section 3, the assumption that the working pressure on the metal seal element is uniformly distributed is reasonable.

On the other hand, when the working pressure from 0 to 20 MPa is applied on the metal seal element and nonmetal seal element, respectively, the contact stress distributions along the seal face are shown in Figure 16A. It can be seen that the change
FIGURE 12  A, The distribution of the contact stress along the seal face for various internal radiuses of the closure member; B, the relationship between contact stress and internal radius of the closure member

FIGURE 13  A, The distribution of the contact stress along the seal face for different outer radiuses of the closure member; B, the relationship between contact stress and outer radius of the closure member

FIGURE 14  A, The distribution of the contact stress along the seal face for different outer radiuses of the metal seal element; B, the relationship between contact stress and outer radius of the metal seal element
The trend of the contact stress along the seal face, the position of the maximum contact stress and the width of the seal face are very similar for both cases under different working pressures. The absolute values of relative error of the contact stress between the two cases for different working pressures are all <4%.

Figure 17 summarizes the mean absolute values of relative error between the theoretical and numerical solutions of the contact stress for different independent variables. It can be drawn that the mean absolute values of relative error between $\sigma_1$ and $\sigma_{ave}$ caused by all parameters except the cantilever beam length are smaller than 10%. And the mean absolute values of relative error between $\sigma_2$ and $\sigma_{ave}$ caused by all parameters are approximately 15%. It may therefore be concluded that the theoretical relationship between different independent variables, and the contact stress is in good agreement with the relationship between different independent variables and the numerical solution of the contact stress.
When the relative error between the theoretical and numerical solutions of the contact stress is <10%, the independent variable values of the metal seal element are proposed in Table 3, which supply the constraints for the optimum design of the metal seal element. Under the condition of given closure structure, mathematical model of the optimum design of the metal seal element can be expressed as follows:

$$\max \sigma_{Ex}(q_2, u_1, \sin \beta, h, l, t)$$

(33)

where the implication is that the contact stress is maximized by taking appropriate values of independent variables. And the constraints are $0.013 \leq u_1 \leq 0.101, 68 \leq \beta \leq 88, 0.58 \leq h \leq 0.84, 9.7 \leq l \leq 14.3, 1.8 \leq t \leq 2.3$ and $0 \leq \sigma_{Ed} \leq \sigma_y$.

Under the conditions of given working pressure and the closure member structure, the optimum structural parameters of the metal seal element can be obtained based on the optimization algorithm Equation (33) and the constraints.

The arithmetic average values of the theoretical solution of the contact stress under the above nine conditions are obtained, and the three-dimensional histogram and the absolute errors between the theoretical and numerical solutions of the contact stress are shown in Figure 18. The theoretical solution of the contact stress given by the interference fit model between the closure member and the metal seal element is much smaller than that given by the cantilever beam model. And the ratio of contact stress value given by interference fit to the total contact stress value is approximately 5%. Therefore, the contact stress value based on the cantilever beam model can be used as an approximation of the theoretical solution.

### 5 CONCLUSIONS

The theoretical and numerical analyses are conducted to analyze the radial metal seal for the ICV in intelligent well. A theoretical model of the contact stress of radial metal seal has been derived, and corresponding numerical analyses via the ABAQUS software are conducted to make a comparison with the theoretical results. The results demonstrate that the theoretical and numerical solutions agree with each other well. Additionally, the influence of the independent variables on the contact stress is analyzed. Based on all of these, the following conclusions can be drawn:

1. The influence level of independent variables on contact stress from high to low is as follows: working pressure, seal face width, initial interference, inclination angle, and...
the closure member structure parameters. The influence of the sectional height and length of metal seal element on the contact stress is complex, which is related to the square or cube of their ratio. Therefore, the influence of the sectional height and length of metal seal element on the contact stress should be considered comprehensively in the design of radial metal seal.

2. When the mean absolute value of relative error between the analytical and numerical solutions of contact stress is <10%, the independent variable values are proposed, and these parameter values can be used as a reference for designing the radial metal seal. Combined with working pressure and material parameters, the structural parameters of metal seal element can be given preliminarily by optimum design method.

3. The ratio of contact stress value given by interference fit to total contact stress value is approximately 5%, which indicates that the interference fit between the closure member and the metal seal element has little contribution to the total contact stress. Thus, the contact stress value based on the cantilever beam model can be used as an approximation of the theoretical solution.

The experimental setup bench has been set up and the experimental verification of the theoretical model is expected to carry out in future research.

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CONFLICT OF INTEREST
The authors declare that they have no conflicts of interest.

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