

Original Paper

Stress characteristics and sealing mechanism of steel-titanium heterogeneous drill pipe joints



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ABSTRACT

Titanium alloy drill pipes exhibit promising potential for ultra-deep well drilling due to their high specific strength and superior corrosion resistance. However, the engagement surfaces of the pin and box joints feature distinct material stiffness and surface roughness, which influence both the load bearing characteristics and sealing performance. This study establishes a 3D elastoplastic finite element model for steel-titanium heterogeneous drill pipe joints and analyzes their stress distribution under complex loading conditions. Furthermore, a sealing performance evaluation model based on microscopic leakage mechanisms is proposed, and the sealing performance of single shoulder and double shoulder drill pipe joints is comparatively analyzed. The results indicate that, compared to the NC50 single-shoulder joint, the DS50 double-shoulder joint exhibits a 12.8% improvement in sealing performance under an axial tension of 2500 kN, and a 27.6% improvement under combined loading of 2500 kN axial tension and 8 kN·m bending moment. The synergistic action of the primary and secondary shoulders enhances the sealing stability of the drill pipe joint.

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1. Introduction

As drilling depths increase, drill strings face increasingly extreme loading conditions. For wells over 9000 m, the maximum axial tensile load on the upper steel drill pipes can exceed 3500 kN, placing high demands on both the drill string and the rig. These challenges, combined with high downhole temperatures, pressures, and in-situ stresses, create critical obstacles in ultra-deep well drilling. To meet these tough conditions, various specialized drill pipes have been developed worldwide, including high strength steel, aluminum alloy, titanium alloy, and carbon fiber drill pipes. Among these, titanium alloy drill pipes stand out because of their high specific strength, excellent resistance to high temperatures and corrosion, and long fatigue life. These benefits greatly reduce rig load and energy use while increasing the tensile margin and safety of the drilling tools. Additionally, titanium alloys

overcome the drawbacks of high-strength steel (moderate specific strength), aluminum alloys (limited high temperature resistance), and carbon fiber (high costs), making them highly suitable for ultra-deep well drilling.

In the late twentieth century, Smith et al. (2001) pioneered a titanium alloy pipe-steel joint combination approach to design a transition-joint type titanium-steel heterogeneous drill pipe (as shown in Fig. 1). The key innovation of this design lies in its elimination of the upset forging process on the titanium alloy end. Since the welded joint in the titanium-titanium region can maintain that of the base metal, omitting upset forging avoids the risk of microcracks induced by cold working and effectively reduces manufacturing costs. The application of this structure can reduce hook load and torque by approximately 40%. However, this design introduces two significant challenges (Kang et al., 2020): 1) The assembly between the steel joint and the titanium alloy pipe body is relatively complex, and the shoulder area is prone to developing residual stresses, thereby compromising the overall fatigue life of the drill pipe; 2) Galvanic corrosion is likely to occur between the steel joint and the titanium alloy pipe body, further reducing the overall fatigue life of the drill pipe.

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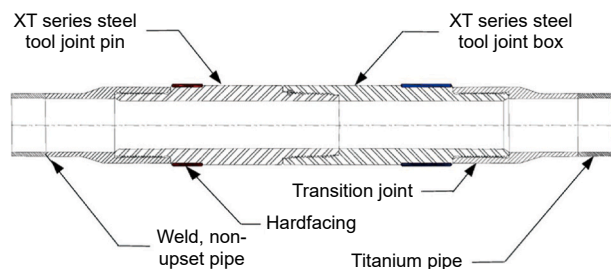


Fig. 1. Titanium drill pipe joint with transition fitting.

For an extended period, titanium alloy drill pipes have mainly used a design that combines a titanium alloy pipe body with steel joints. Although this setup offers high strength and flexibility, it presents two main challenges: 1) The connection between steel joints and the titanium alloy pipe body is complex and often causes residual stresses at the shoulder interface, which negatively impacts the drill pipe's fatigue life. 2) Galvanic corrosion easily occurs between the steel joints and the titanium alloy pipe body, further decreasing the drill pipe's fatigue life.

Recent advances in titanium alloy welding technology have led to significant progress. In 2017, Bohai NKK Drill Pipe Co., Ltd. successfully developed a friction-welded all-titanium alloy drill pipe, as shown in Fig. 2. Testing confirmed that the drill pipe meets all relevant performance and quality standards. This innovative design replaces the traditional cold-assembly process of titanium alloy tubes with steel joints by using titanium alloy for all components, including the joints. The friction welding technique effectively solves technical challenges such as upsetting at the tube ends, heat treatment, and joint friction welding (Cheng et al., 2020).

Titanium alloy drill pipes have seen considerable application in short-radius horizontal wells, yet research on their use in ultra-deep wells remains insufficient. While titanium alloy drill pipes themselves exhibit excellent performance characteristics, their practical application in composite drill strings inevitably results in heterogeneous titanium-steel joint combinations. Such combinations result in four distinct material pairing forms: steel-titanium, titanium-steel, steel-steel, and titanium-titanium, subjecting the joint areas to complex mechanical conditions. Under the extreme working environments of ultra-deep wells, significant axial loads, high torque, and elevated temperature and pressure can exacerbate uneven stress distribution at these heterogeneous joints. Furthermore, differences in stiffness and surface roughness between the materials may lead to issues such as galling, sealing failure, and fractures (Yan, 2016). This paper systematically investigates the load-bearing behavior and sealing mechanisms of titanium-steel heterogeneous joints under the specific conditions encountered in composite drill strings, aiming to provide a theoretical basis for the safe application of titanium alloy drill pipes in ultra-deep wells. Currently, research on the integrity of connections and sealing in heterogeneous drill pipe joints remain severely limited, significantly restricting the selection, design, and widespread adoption of titanium alloy drill pipes. Therefore, conducting an in-depth analysis of the load-bearing characteristics at the engagement interfaces of heterogeneous joints under ultra-

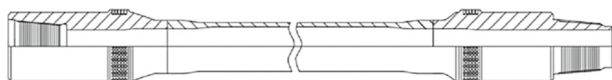


Fig. 2. Schematic diagram of the Bohai NKK all-titanium alloy drill pipe structure.

deep well conditions, and elucidating the mechanisms behind thread galling and sealing failure in dissimilar-material drill pipe joints, is crucial for enhancing the operational safety of composite drill strings incorporating titanium alloy components.

Schwind (1990) investigated the leak resistance of API round-thread joints and analyzed the effects of sealing pressure and contact width. They developed a mathematical model based on elastic theory to evaluate leak resistance under tensile loading. Xu et al. (2019) employed Hertz contact theory to develop a contact model for threaded joint sealing surfaces, examining the effects of sealing surface taper, applied torque, and material stiffness on contact width and stress. Their work provided a theoretical foundation for study of threaded joint sealing performance. Yu et al. (2023) developed a geometrically nonlinear model, coupled with empirical formulas derived from experimental data, to investigate the sealing capacity of premium threaded joints (PTJs) under practical working loads. Their results indicated that while internal pressure increases contact pressure along the sealing path, it concurrently reduces the joint's sealing capacity. Additionally, axial tensile loads significantly impair sealing performance.

Despite these advances, most discussions on joint sealing still focus on macroscopic perspectives. They often combine sealing parameters, such as contact pressure, with operational conditions to build models. Current evaluations rely heavily on indirect methods for analyzing sealing surface contact pressure. As a result, they fail to quantify how sealing performance relates to factors such as contact area, surface roughness, and surface treatment technologies.

Xu et al. (2014) proposed a microscopic gas leakage model for threaded joints with metal conical sealing structures, incorporating machining parameters and surface roughness to quantitatively evaluate sealing performance. Their findings revealed that gas leakage rates varied by an order of magnitude for different surface roughness levels, underscoring the need to consider machining precision in sealing design and performance assessment. Zhang et al. (2018) treated sealing interfaces as porous media and characterized their porosity using fractal theory. By applying Darcy's law, they developed a leakage model for sealing interfaces. The results showed good agreement between experimental data and model predictions when the sealing surfaces exhibited isotropic or fractal characteristics.

Theories of seepage and porous media have proven effective in investigating leakage in mechanical static seals. These theories are valuable for evaluating drill pipe joint sealing performance. However, they mainly address certain aspects. The microscopic contact state and stress distribution of sealing surfaces under contact stress are also critical for accurate assessment, but related research remains limited. Previous microscopic sealing models for threaded joints often used simplified roughness peak models. They neglected the influence of asperity arrangement on sealing performance (Liu et al., 2023; Yu et al., 2022). To fill these gaps, this study uses a digital characterization to analyze the microscopic morphology of rough surfaces and the sealing behavior of steel-titanium heterogeneous drill pipe joints under complex loading conditions. The aim is to provide novel methodologies and perspectives for evaluating the sealing performance of heterogeneous drill pipe joints.

2. Three-dimensional stress analysis of drill pipe joints

In ultra-deep drilling exceeding 8000 m, the application of titanium alloy drill pipes throughout the entire drill string is prohibitively expensive. Existing reports indicate that titanium alloy drill pipes are primarily used in the build-up sections of short-

Table 1
Structural parameters of NC50 drill pipe joint.

Parameter	Value	Parameter	Value
Outer diameter of joint, mm	168.3	Inner diameter of pin, mm	69.8
Inner diameter of box, mm	88.9	Bore diameter, mm	134.9
Pin length, mm	114.3	Pitch diameter of base, mm	128.1
Inner diameter of shoulder, mm	67.0	Outer diameter of shoulder, mm	77.0
Thread taper	1:6	Thread type	V-0.038R

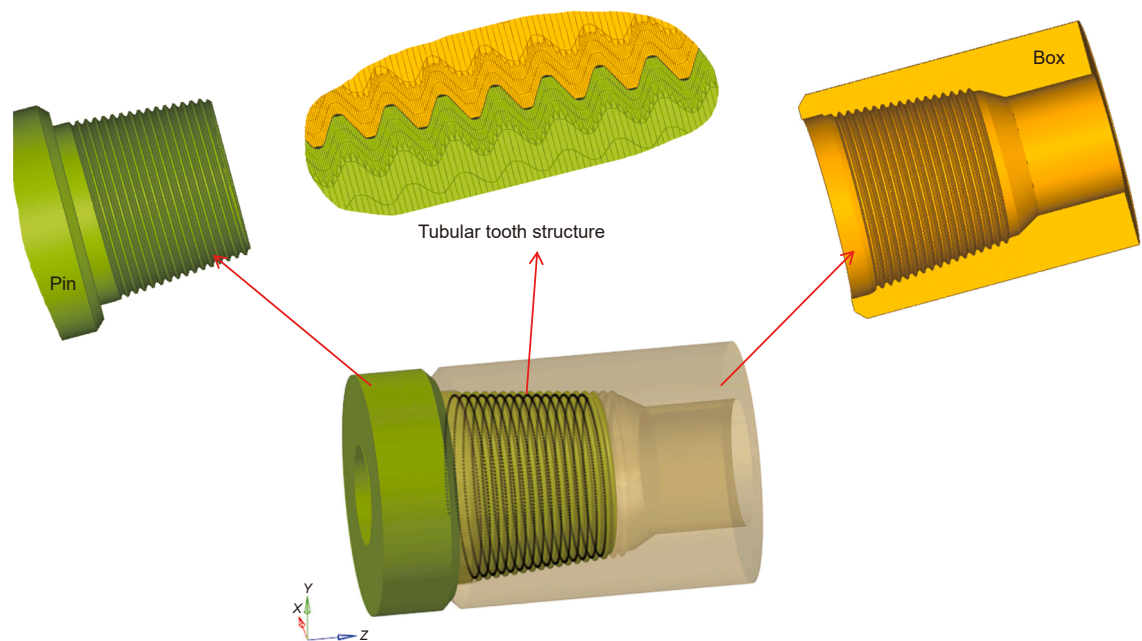


Fig. 3. 3D FE model of the NC50 drill pipe joint.

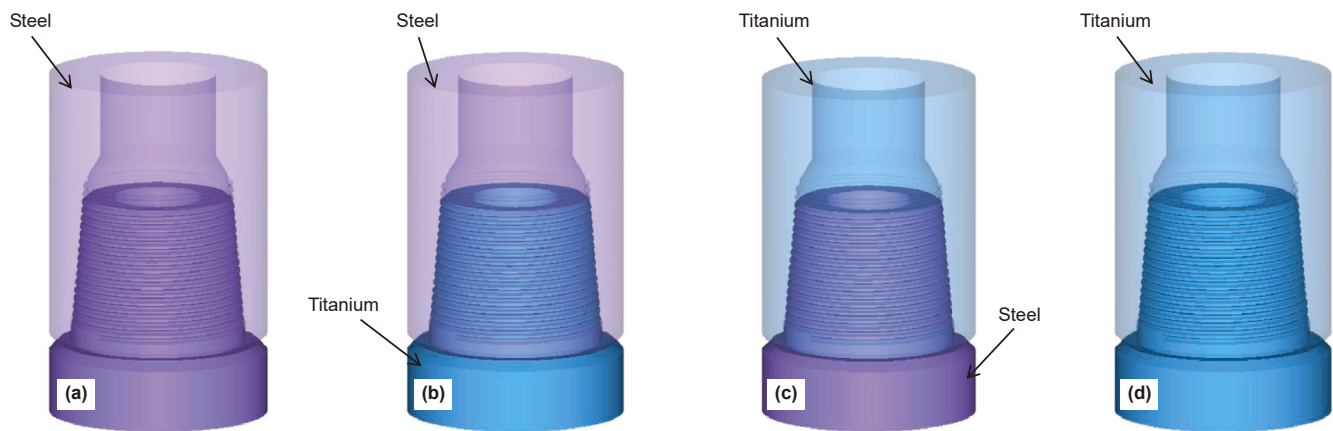


Fig. 4. Schematic diagram of drill pipe joints with different materials: (a) Steel, (b) titanium-steel, (c) steel-titanium, (d) titanium.

radius horizontal wells, suggesting that steel-titanium hybrid drill strings a more feasible solution (American Petroleum Institute, 1990). During the operation of heterogeneous joints, shoulder sealing failure can allow corrosive fluids to infiltrate the threaded region, increasing the risk of cracking under high-stress conditions. Therefore, analyzing the sealing performance of steel-titanium heterogeneous drill pipe joints and designing joints that meet performance requirements are crucial for ensuring the

operational safety of hybrid drill strings in ultra-deep and extreme-depth wells.

2.1. Finite element model of drill pipe joints

To investigate the influence of material properties on the mechanical behavior of drill pipe joints, a three-dimensional elasto-plastic finite element model was developed. The model is based on

Table 2
Stress-plastic strain for steel and titanium.

Steel				Titanium			
True stress, MPa	Plastic strain	True stress, MPa	Plastic strain	True stress, MPa	Plastic strain	True stress, MPa	Plastic strain
827.4	0.00000	976.5	0.02008	691.2	0.00000	831.9	0.00606
845.2	0.00010	1035.4	0.04007	710.6	0.00005	846.1	0.00909
855.3	0.00023	1061.5	0.05041	723.4	0.00010	859.5	0.01304
861.4	0.00032	1082.9	0.06065	739.7	0.00020	867.3	0.01602
875.6	0.00069	1099.8	0.07082	751.1	0.00031	875.7	0.02010
907.1	0.00251	1110.5	0.08000	765.1	0.00052	883.8	0.02523
927.3	0.00519			781.3	0.00096	890.2	0.03044
944.6	0.01033			800.5	0.00207	893.1	0.03354
961.0	0.01523			815.2	0.00359	896.4	0.04017

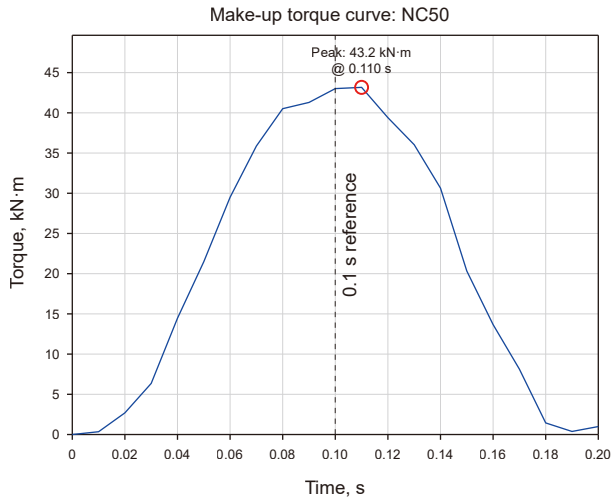


Fig. 5. Make-up torque distribution curve for NC50 joints.

the API NC50 drill pipe joint specifications (see Table 1), as illustrated in Fig. 3. The joint model comprises 567,840 elements and 649,560 nodes. Two types of heterogeneous drill pipe joint models were established based on the pin and box material combinations: titanium-steel (titanium pin, steel box) and steel-titanium (steel pin, titanium box), as shown in Fig. 4. In the figure, blue represents the titanium alloy material, while purple denotes the steel material.

Distributed coupling elements were applied at the free end of the pin to impose external loads, while kinematic coupling elements were placed at the free end of the box to apply constraints. The threaded engagement surfaces and shoulders were defined as surface-to-surface contacts, with tangential contact modeled using Coulomb friction and the normal contact set to hard contact. Considering the influence of thread compound containing 40%–60% zinc powder, the friction coefficient between the pin and box was set to 0.08 (American Petroleum Institute, 1990).

Based on mechanical tests, the elastic modulus of the titanium alloy was determined as 118.6 GPa, Poisson's ratio as 0.33, yield strength as 691.2 MPa, and ultimate tensile strength as 856.0 MPa. For steel, the corresponding values were 206.0 GPa, 0.29, 827.4 MPa, and 1020.6 MPa, respectively. The true stress and plastic strain data for both materials are detailed in Table 2.

2.2. Load-bearing characteristics of drill pipe joints under complex loads

To examine the load-bearing characteristics of threaded teeth in heterogeneous joints under complex loading, a node path along

the pin helix at the thread root was selected (Fig. 3). Identical external loads—a make-up torque of 43.1 kN·m, axial tension of 1600 kN, bending moment of 15.0 kN·m, and internal pressure of 30 MPa—were applied to different material configurations to compare their mechanical responses. The torque distribution curve is shown in Fig. 5.

The von Mises stress distribution under complex loading is illustrated in Fig. 6. Specifically, Fig. 6(a)–(c) depict the von Mises stress in the titanium section of the joints, while Fig. 6(d)–(f) shows the stress in the steel section. All four single-shoulder joint configurations exhibited similar overall stress levels, with peak stresses occurring at the first engaged thread near the shoulder. The maximum stresses in titanium and steel components remained below their respective ultimate strengths (856.0 and 1020.6 MPa), confirming the joints' structural integrity under the applied loads.

To study the load-bearing features of single-shoulder drill pipe joint threads made from different materials under complex loads, a path was selected along the base of the thread teeth on the pin spiral line, as shown in Fig. 7.

Fig. 8 shows von Mises stress distribution along the root node path of the thread teeth for various single-shoulder drill pipe joints under complex loading. The applied make-up torque causes a non-uniform stress distribution in the threads, with higher stress near the shoulder and lower stress farther away. When axial tension is used, the overall threaded section stress increases significantly. The degree of stress increase in the last few thread teeth differs by joint material combination: for example, the steel-titanium heterogeneous joint shows the greatest rise in these posterior threads, while other material combinations exhibit less pronounced increases. Consequently, for the steel-titanium joint, a stress pattern elevated at both ends and lower in the middle is observed.

When a bending moment is applied, one side of the joint experiences tension, while the other experiences compression. We extracted the maximum and minimum stress values for each thread tooth in the four types of single-shoulder drill pipe joints: Type A (60.4 MPa), Type B (56.4 MPa), Type C (97.1 MPa), and Type D (83.3 MPa), respectively, clarifying the specific material or joint types for each value.

Notably, the steel (pin)-titanium (box) single-shoulder drill pipe joint shows the greatest von Mises stress range in its threads, compared to steel-steel, titanium-steel, and titanium-titanium joints, which may weaken its bending fatigue resistance.

Given the steel-titanium heterogeneous joint's relatively low bending fatigue resistance, further analysis of contact characteristics at the shoulder surface is necessary to evaluate its sealing performance. The sealing capability of drill pipe joints is closely related to the contact pressure at the shoulder interface, where higher contact pressure within a certain range generally indicates

better sealing performance (Xue et al., 2022). Therefore, contact pressure serves as a critical parameter for assessing joint sealing performance. Under complex loading conditions, the changes in contact area and contact pressure on the shoulder mating surface of the steel-titanium heterogeneous joint are illustrated in Fig. 9(a) and (b), respectively. The calculated average contact pressure on the shoulder under stable working conditions is 424.72 MPa, a key metric for evaluating sealing performance.

3. Evaluation of sealing performance in steel-titanium heterogeneous drill pipe joints

Under downhole complex loading conditions and high-pressure fluid erosion, the coating on steel-titanium heterogeneous drill pipe joints is prone to failure. Once the coating deteriorates, the electrochemical potential difference between the titanium alloy (higher potential) and the carbon steel (lower potential) promotes galvanic corrosion at their interface. This localized corrosion creates narrow crevices in the threaded engagement area (Zhang, 2022). Consequently, compared with homogeneous steel drill pipe joints, heterogeneous steel-titanium joints exhibit a significantly increased risk of thread-sealing failure. In double-shoulder joint designs, the secondary shoulder serves as a protective barrier, preventing direct contact between drilling fluids and the pin/box threads and thereby reducing corrosion susceptibility and crack propagation. As illustrated in Fig. 10, double-shoulder joints demonstrate superior corrosion resistance compared to single-shoulder designs, making them more reliable under complex downhole loading conditions.

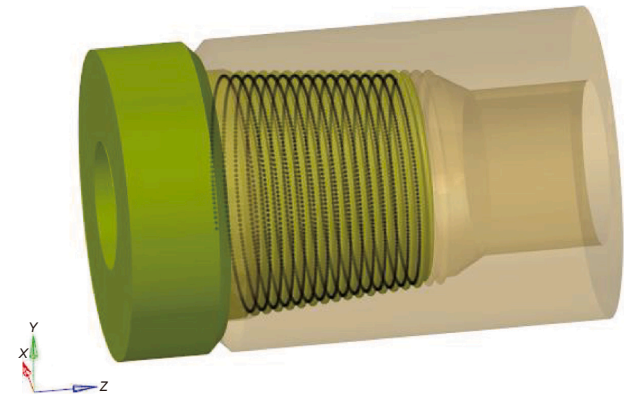


Fig. 7. Node path along the pin thread root.

For the secondary shoulder engagement surface of the double-shoulder drill pipe joint, the internal pressure equals the bore pressure, p_{si} inside the joint, while the external pressure is approximately 0.1 MPa. In contrast, the primary shoulder engagement surface is subjected to the external annular pressure p_{po} outside the joint, with its internal pressure being approximately 0.1 MPa.

Sealing effectiveness primarily depends on the contact pressure generated by axial interference fit at the shoulder interface. When subjected to downhole loads, drill pipe joints experience reduced shoulder contact pressure, potentially compromising or even eliminating sealing capability. Leakage rate serves as a critical comprehensive metric for evaluating seal performance (Silva and

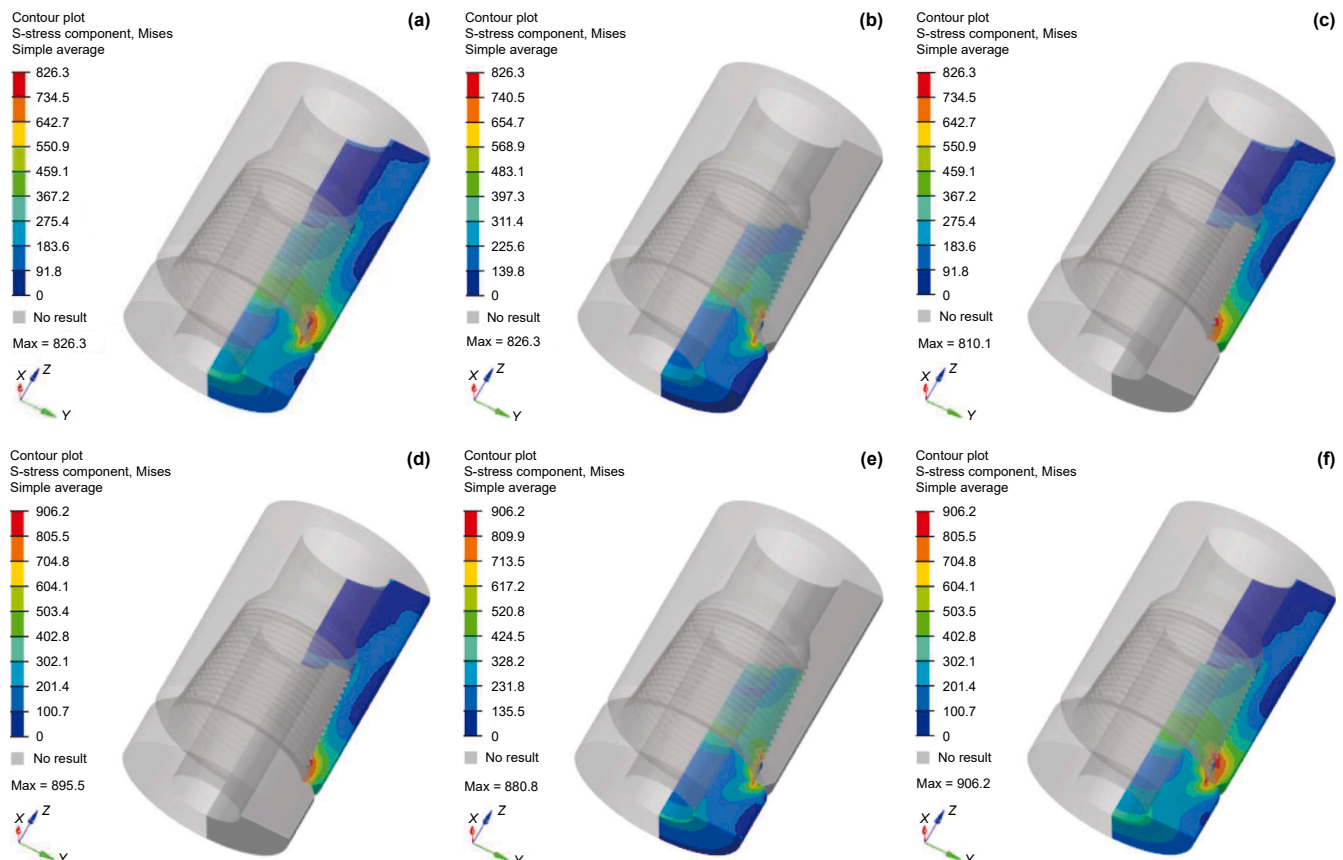


Fig. 6. von Mises stress distribution in drill pipe joints under complex loading: (a) Titanium, (b) titanium-steel, (c) steel-titanium, (d) titanium-steel, (e) steel-titanium, (f) steel.

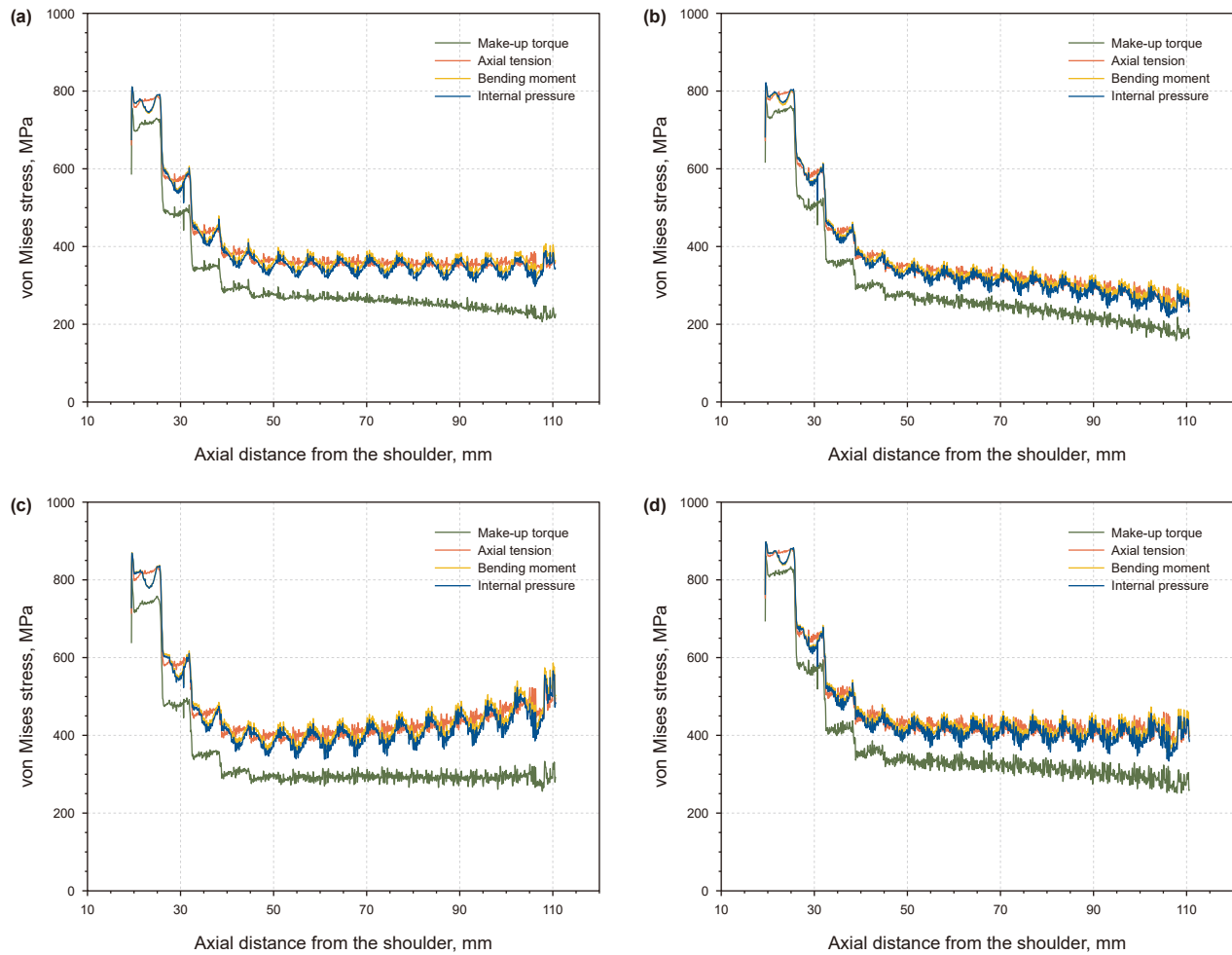


Fig. 8. von Mises stress distribution along the thread root path under different loading conditions: (a) Titanium, (b) titanium-steel, (c) steel-titanium, (d) steel.

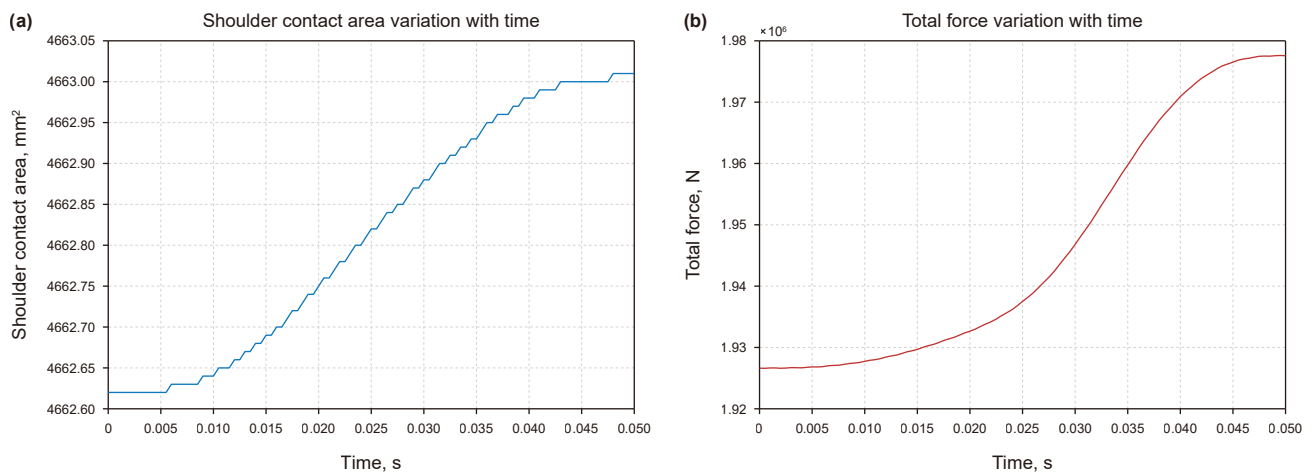


Fig. 9. Shoulder contact characteristics under operational loads: (a) Contact area variation on the shoulder surface; (b) contact pressure variation on the shoulder surface.

Netto, 2022), defined as the volumetric flow rate of test medium passing through the sealing interface per unit time under standardized conditions. Accurate prediction of leakage rates is essential for assessing the sealing performance of heterogeneous drill pipe joints.

The ISO/TR 10400 standard (ISO TR 10400, 2018) evaluates thread joint sealing performance from a “zero-leakage” perspective by analyzing contact pressure distribution on sealing surfaces, though it does not specify quantitative leakage rate criteria. In contrast, ISO 13679 (2002) defines effective sealing during internal

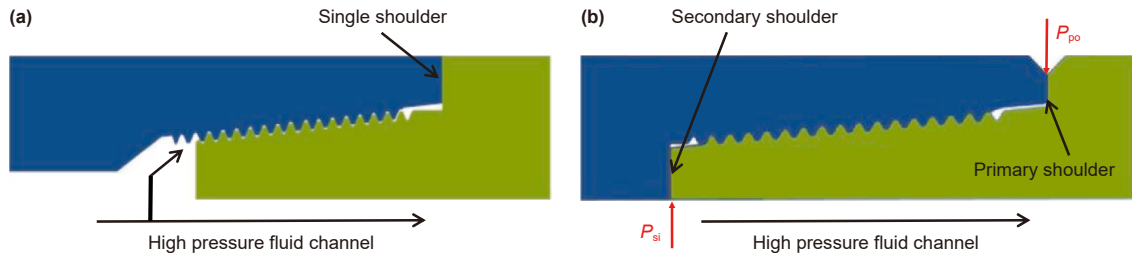


Fig. 10. Comparison of single-shoulder and double-shoulder structures: (a) Single shoulder, (b) double shoulder.

pressure testing as a gas leakage rate below $0.9 \text{ cm}^3/15 \text{ min}$ ($1 \times 10^{-9} \text{ m}^3/\text{s}$), but does not define standards for liquid leakage. Because liquids possess larger molecular sizes and stronger intermolecular forces compared to gases, they have lower permeability through microscopic metal pores, thereby presenting less leakage risk. Thus, referencing gas leakage criteria to assess liquid sealing performance in drill pipe joint shoulder models is justified by these physical differences.

3.1. Development of the shoulder sealing model

The make-up process of drill pipe joints, where pin and box shoulders transition from non-contact to full sealing, can be microscopically simplified as the contact between two rough surfaces. For metal seals, sealing analysis focuses on two key

aspects: (1) characterizing metal surface roughness to calculate interfacial contact loads, and (2) modeling complex leakage channels before and after contact to determine leakage rates under specific pressure conditions.

To address these challenges, this study establishes a multiscale shoulder sealing model for drill pipe joints, with data transfer achieved through the following steps (Fig. 11(a) and (b)):

First, finite element analysis outputs—including local contact pressure $p(x, y)$ and nodal displacements $\bar{u}_z$ are extracted from shoulder contact simulations.

A weighted average pressure $\bar{p} = \frac{\sum_{i=1}^n p_i A_i}{\sum_{i=1}^n A_i}$ is calculated over the contact area, where p_i represents elemental pressure and A_i denotes elemental contact area. The maximum displacement difference $\lambda = \max(u_z) - \min(u_z)$ within the contact zone is then determined.

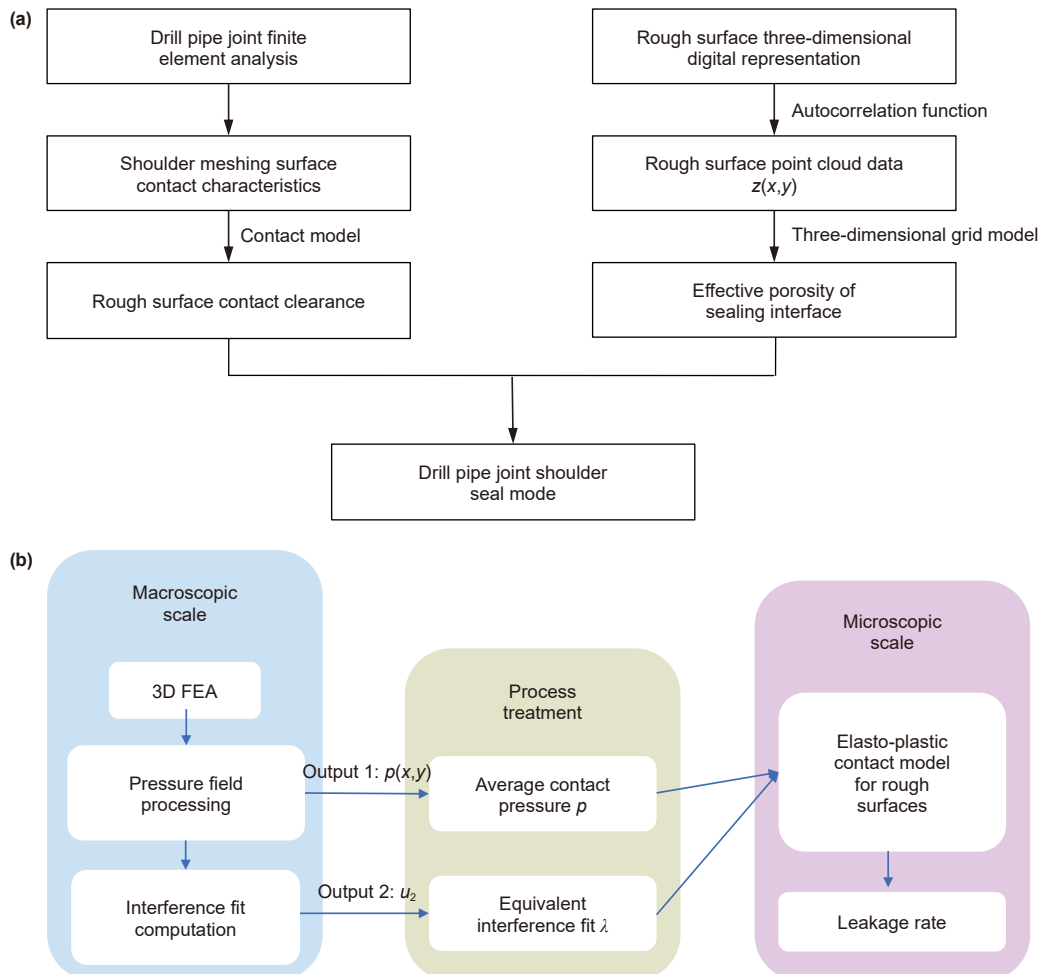


Fig. 11. Flowchart of the shoulder sealing model: (a) Flowchart of the multiscale modeling approach, (b) data transfer process between macro and micro models.

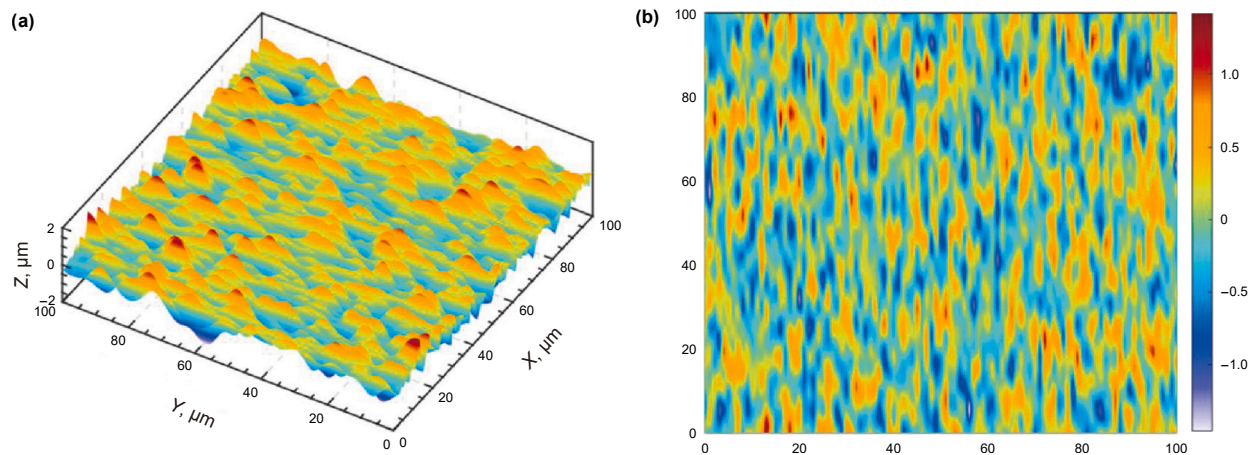


Fig. 12. Microscopic morphology of rough surfaces: (a) Rough point cloud data ($\beta_x = 1, \beta_y = 50$), (b) Rough surface cloud map ($\beta_x = 1, \beta_y = 50$).

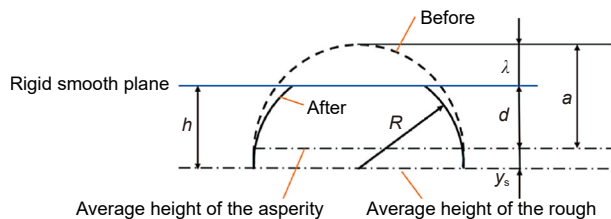


Fig. 13. Schematic of a single asperity in contact with an ideal rigid plane.

These parameters ($\bar{p}$ and λ) serve as inputs for the contact model, which integrates macroscopic geometric features and microscopic surface topography to evaluate sealing performance.

At the microscale, rough surfaces are digitally reconstructed using statistical modeling principles, accounting for specific roughness parameters and surface texture orientations. An elastoplastic contact model is developed to establish the functional relationship between average contact pressure and interfacial gap distribution. A three-dimensional grid model is subsequently used to calculate the porosity of the shoulder mating interface. Macroscopically, full-scale finite element analysis of drill pipe joints computes the average contact pressure on shoulder interfaces under operational loads. By combining microscale porosity data with macroscale pressure distributions, the model applies seepage mechanics theory to predict leakage rates across the shoulder interface under designated pressure conditions.

3.2. Digital characterization of rough surface morphology

During manufacturing, cutting tools impart distinct machining marks on thread flanks and shoulder surfaces of drill pipe joints,

creating characteristic microtopographies that directly influence contact behavior and sealing performance. While surface roughness can be measured experimentally, raw data often requires filtering and modification for analytical use. In contrast, digital characterization of rough surfaces not only provides efficient discrete datasets but also enables parametric studies of surface features.

This study employs stochastic process theory (Nowinka and Dall'Acqua, 2011) for three-dimensional surface characterization, treating rough surfaces as stationary random processes described by autocorrelation functions and statistical parameters. Typically, turned shoulder surfaces exhibit strong anisotropy, with periodic tool marks aligned along the machining direction (Greenwood and Williamson, 1966). The microscopic topography is digitally reconstructed using an exponential autocorrelation function, $g(x_s, y_s)$, to simulate interdependencies between critical surface points, thereby enabling the generation of three-dimensional microsurface profiles.

The exponential autocorrelation function is expressed as:

$$g(x_s, y_s) = \psi^2 \exp \left\{ -2.3 \times \left[\left(\frac{x_s}{\beta_x} \right)^2 + \left(\frac{y_s}{\beta_y} \right)^2 \right]^{\frac{1}{2}} \right\} \quad (1)$$

where ψ represents the root mean square (RMS) roughness of the surface, β_x and β_y denote the correlation lengths in the X and Y directions, respectively, x_s and y_s are the spatial coordinates on the surface.

Greenwood's extensive measurements of engineering surfaces revealed that most surface asperity heights follow Gaussian distribution characteristics (Wang, 2014). The procedure for generating Gaussian-distributed rough surfaces comprises the following steps:

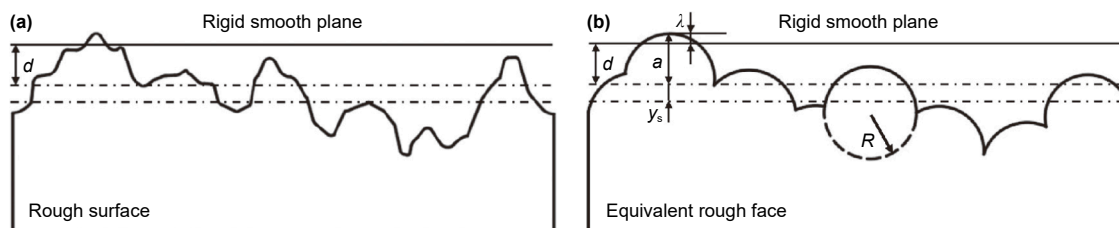


Fig. 14. Schematic diagram of rough surface contact: (a) Actual rough face, (b) equivalent rough face.

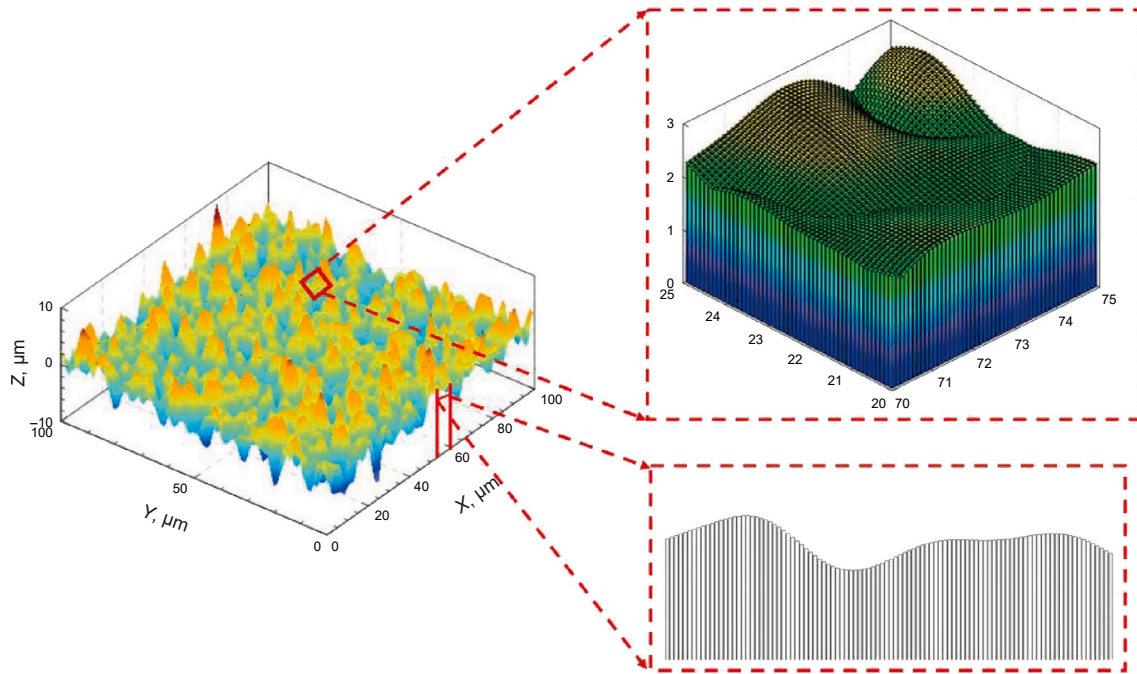


Fig. 15. Discretized rough surface represented by point cloud data.

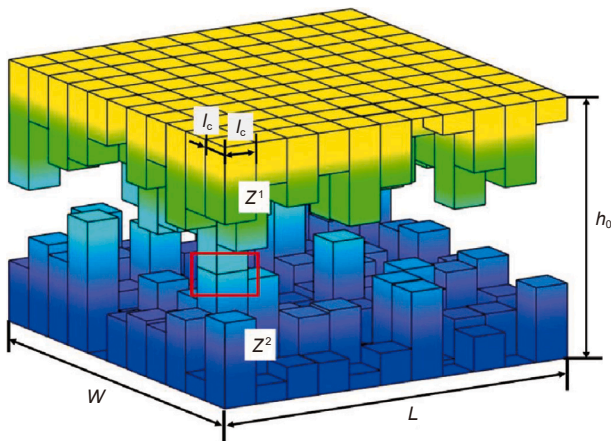


Fig. 16. Three-dimensional grid model of the sealing interface.

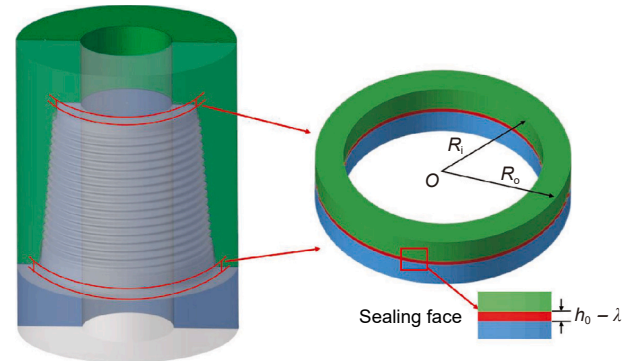


Fig. 17. Schematic diagram of shoulder sealing.

- (1) Generate a two-dimensional Gaussian white noise random sequence $l(x_s, y_s)$ and apply Fourier transformation to obtain its spectral representation $L(\omega_x, \omega_y)$, where the input sequence's power spectral density is denoted as γ .
- (2) Perform Fourier transformation on the exponential auto-correlation function $g(x_s, y_s)$ to derive the power spectral density function of the filtered output signal $G(\omega_x, \omega_y)$.
- (3) Define the transfer function $T(\omega_x, \omega_y)$ of the filter as:

$$T(\omega_x, \omega_y) = (G(\omega_x, \omega_y) / \gamma)^{1/2} \quad (2)$$

- (4) Process the input 2D random sequence through the filter to obtain the output sequence, representing the height distribution in Fourier space:

$$Z(\omega_x, \omega_y) = T(\omega_x, \omega_y) \times L(\omega_x, \omega_y) \quad (3)$$

- (5) Apply inverse Fourier transformation to derive the height distribution function $Z(x_s, y_s)$ of the rough surface.

The arithmetic average deviation (R_a) of machined surfaces is typically relatively low. For Gaussian-distributed surfaces, the relationship between root mean square (RMS) roughness and R_a can be approximated as $RMS \approx 1.11 R_a$. Taking into account that the surface roughness of tool joints tends to increase beyond initial machined values due to installation and downhole wear, and considering the idealized nature of asperity equivalence methods, a root mean square (RMS) roughness of $2.0 \mu m$ (corresponding to $R_a \approx 1.8 \mu m$) was selected, which falls within the typical range for machined surfaces under realistic drilling conditions.

Surface dimensions = $100 \mu m \times 100 \mu m$, and autocorrelation coefficients $\beta_x = 1$, $\beta_y = 50$. This anisotropic surface generation produces directional textures aligned with the larger autocorrelation coefficient (Fig. 12(a) and (b)), effectively simulating turned shoulder surfaces with distinct Y-parallel grooves. While simulated surfaces cannot replicate all real machining features, they

effectively represent texture characteristics crucial for engineering applications.

3.3. Elastoplastic contact model for rough surfaces

Sealing performance depends not only on surface topography but also on the contact mechanics between two rough surfaces. During compression, initial contact occurs at the highest asperity peaks. Analyzing single-asperity contact behavior enables accurate prediction of:

- (1) Actual contact pressure; (2) Normal deformation; (3) Post-loading porosity evolution.

The Zhao-Maietta-Chang (ZMC) model (Greenwood and Tripp, 1970), based on deformation continuity principles, was employed under these assumptions:

- (1) Asperity summits are spherical
- (2) Asperities are sufficiently spaced to neglect interactions

This model is widely recognized and applied in the analysis of rough surface contact due to its robust framework, which bridges the elastic and fully plastic deformation regimes. Based on deformation continuity principles, it provides a smooth, continuous transition function between these stages, offering more accurate predictions for the contact area and pressure during the elastoplastic transition—a critical phase for metal sealing surfaces under downhole loading conditions.

3.3.1. Single asperity contact model

The contact between two rough surfaces can be simplified as an equivalent rough surface interacting with a rigid plane (Fig. 13) (Timoshenko and Goodier, 1951). Key parameters include:

In the figure: λ is the compression depth (μm), a is the asperity height (μm), y_s is the distance between asperity mean plane and surface mean plane (μm), d is the separation rigid plane and asperity mean plane (μm), R is the equivalent asperity radius (μm).

The compression amount λ generated by the asperity under normal load is given by:

$$\lambda = a - d \quad (4)$$

Under normal loading, asperities undergo three deformation stages:

1. Elastic deformation ($\lambda < \lambda_1$)

Per Hertzian contact theory (Tabor, 2000):
Contact area:

$$A_e = \pi R \lambda \quad (5)$$

Mean pressure:

$$p_e = \frac{4E}{3\pi} \left(\frac{\lambda}{R} \right)^{1/2} \quad (6)$$

$$E = E_1 / (1 - \nu_1^2) + E_2 / (1 - \nu_2^2) \quad (7)$$

where equivalent modulus E , with E_1 , E_2 and ν_1 , ν_2 being elastic moduli and Poisson's ratios of steel and titanium alloy, respectively.

2. Yield initiation ($p_e = 0.6 H_m$)

Material hardness $H_m = 2.84\sigma_f$ (σ_f : yield strength).
Critical compression λ_1 :

$$\lambda_1 = \left(\frac{0.9\pi H_m}{2E} \right)^2 R \quad (8)$$

3. Full plasticity ($\lambda \geq \lambda_2$)

Based on Tabor's indentation tests (McCool, 1986; Tabor, 2000):

$$\lambda_2 = 110\lambda_1 \quad (9)$$

Contact area:

$$A_p = 2\pi R \lambda \quad (10)$$

Mean pressure:

$$p_p = H_m \quad (11)$$

3. Elastoplastic transition ($\lambda_1 \leq \lambda < \lambda_2$)

The ZMC model introduces a bridging function $f(\lambda)$ to ensure smooth transitions:

$$f(\lambda) = -2 \left(\frac{\lambda - \lambda_1}{\lambda_2 - \lambda_1} \right)^3 + 3 \left(\frac{\lambda - \lambda_1}{\lambda_2 - \lambda_1} \right)^2 \quad (12)$$

During this stage, the relationship between the contact area, contact pressure, and compression amount becomes highly complex. To ensure smooth and continuous transitions between the elastic, elastoplastic, and fully plastic stages, the ZMC contact model introduces a continuity template function (Greenwood and Tripp, 1970) to smoothly connect the elastic and plastic deformation stages of the asperity.

Satisfying $f(\lambda_1) = 0$, $f(\lambda_2) = 1$ and $f'(\lambda_1) = 0$, $f'(\lambda_2) = 0$, and continuous derivatives. For any function $y = U(\lambda)$, $0 \leq \lambda \leq \lambda_1$ (elastic) and $y = V(\lambda)$, $\lambda_2 \leq \lambda \leq +\infty$ (plastic), the transitional form is:

$$y = U(\lambda) + [V(\lambda) - U(\lambda)] f(\lambda), \lambda_1 \leq \lambda \leq \lambda_2 \quad (13)$$

when $U(\lambda) = p_e$, $V(\lambda) = p_p$, using the template function $f(\lambda)$, the average contact pressure p_{ep} during the elastoplastic stage can be derived as:

$$p_{ep} = 0.6 H_m \left(\frac{\lambda}{\lambda_1} \right)^{1/2} + \left[H_m - 0.6 H_m \left(\frac{\lambda}{\lambda_1} \right)^{1/2} \right] f(\lambda) \quad (14)$$

Similarly, when $U(\lambda) = A_e$, $V(\lambda) = A_p$, the contact area A_{ep} during the elastoplastic stage is:

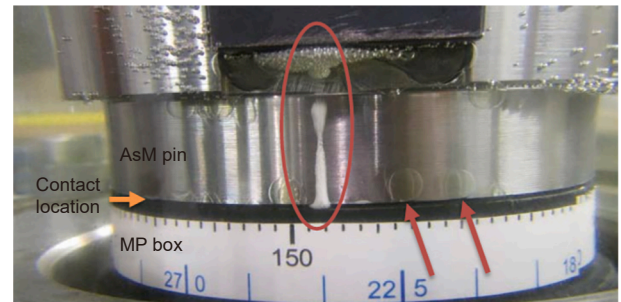


Fig. 18. Sealing experiment of reference material (Ernens et al., 2019).

Table 3
Summary of face seal experimental conditions (Ernens et al., 2019).

Geometry	Value
Round-off radius pin, mm	80
Contact line mean diameter, mm	70
Maximum normal load, kN	120
Contact intensity, N/mm	546
Hertzian contact stress(max), GPa	0.5
Hertzian contact width(max), mm	1.4
Internal pressure(liquid), MPa	70
Viscosity, Pa·s	0.0528
Density, g/cm ³	0.96

$$A_{ep} = \pi R \lambda + (2\pi R \lambda - \pi R \lambda) f(\lambda) \quad (15)$$

3.3.2. Contact model for two rough surfaces

Assuming uniform asperity radius R (Fig. 14), the number of contacting asperities n over nominal area A_n is (Greenwood and Tripp, 1970):

$$n = N \int_d^\infty \varphi(a) da = \eta A_n \int_d^\infty \varphi(a) da \quad (16)$$

where η is the areal density of asperities (mm⁻²), A_n is the nominal contact area (mm²), d is surface separation distance, $\varphi(a)$ is the probability density function of asperity heights.

The variable d represents the separation between the rigid smooth plane and the mean plane of the asperity heights (as illustrated in Figs. 13 and 14). It is the critical distance that determines which asperities are in contact: only asperities with a height greater than d will make contact with the rigid plane. The $\varphi(a)$ is the probability density function of the asperity heights, describing the statistical distribution of the heights of the surface asperities. It is commonly assumed to follow a Gaussian distribution for many engineered surfaces.

Assuming the asperity heights follow a Gaussian distribution, the probability density function is:

$$\varphi(a) = \frac{1}{\sqrt{2\pi}\psi_s} e^{-a^2/(2\psi_s^2)} \quad (17)$$

where ψ_s is the standard deviation of asperity heights (μm).

The number of asperities undergoing elastic deformation n_e , elastoplastic deformation n_{ep} , and fully plastic deformation n_p on the contact surface are respectively:

$$n_e = \eta A_n \int_d^{d+\lambda_1} \varphi(a) da \quad (18)$$

$$n_{ep} = \eta A_n \int_{d+\lambda_1}^{d+\lambda_2} \varphi(a) da \quad (19)$$

$$n_p = \eta A_n \int_{d+\lambda_2}^\infty \varphi(a) da \quad (20)$$

The total contact load F_c between the two surfaces is the sum of the loads on all contacting asperities. For a given surface separation d , the total contact force F_c can be expressed as:

$$F_c(d) = n_e \times p_e A_e + n_{ep} \times p_{ep} A_{ep} + n_p \times p_p A_p \quad (21)$$

Metal-to-metal seal leakage characteristics
Comparison of experimental data, model predictions
and SPE literature

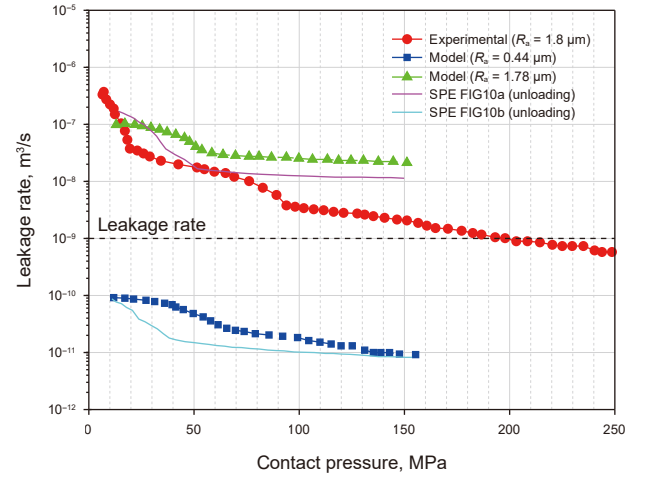


Fig. 19. Comparison between sealing model and experimental results.

Table 4
Parameters of the double-shoulder drill pipe joint.

Parameter	Value	Parameter	Value
External diameter, mm	168.3	Pin inner diameter, mm	88.9
Box inner diameter, mm	88.9	Bored hole diameter, mm	134.9
Pin length, mm	130.0	Pitch diameter (gauge point), mm	128.1
Thread taper	1:6	Thread profile	V-0.038R

$$F_c(d) = \frac{4}{3} \eta A_n E R^{1/2} \int_d^{d+\lambda_1} \lambda^{3/2} \varphi(a) da + 2\pi \eta A_n H R \int_{d+\lambda_2}^\infty \lambda \varphi(a) da + \pi R H \eta A_n \int_{d+\lambda_1}^{d+\lambda_2} \left\{ 0.6 \left(\frac{\lambda}{\lambda_1} \right)^{1/2} [1 - f(\lambda)] + f(\lambda) \right\} [1 + f(\lambda)] \lambda \varphi(a) da \quad (22)$$

The average contact pressure p_t on the contact surface is:

$$p_t(d) = \frac{F_c(d)}{A_n} \quad (23)$$

In this study, the standard deviation of asperity heights $\psi_s = 2.4 \mu\text{m}$, the areal density of asperities $\eta = 3.98 \times 10^3 \text{ mm}^{-2}$, and the equivalent radius of curvature of asperities $R = 6.52 \times 10^{-2} \text{ mm}$ (Yu, 2022).

3.4. Calculation of effective porosity at the sealing interface

To account for the three-dimensional microscopic characteristics of the sealing interface, this section describes the interface using point cloud data and a three-dimensional grid model based on rough surfaces (Zheng et al., 2013). The purpose of this modeling approach is to enable evaluation of the sealing interface's effective porosity, which provides theoretical support for constructing the shoulder sealing model for drill pipe joints.

Using the conceptual model of the equivalent rough surface shown in Fig. 14(b), the actual rough surface is simplified as a contact between spherical asperities and a rigid plane. Based on

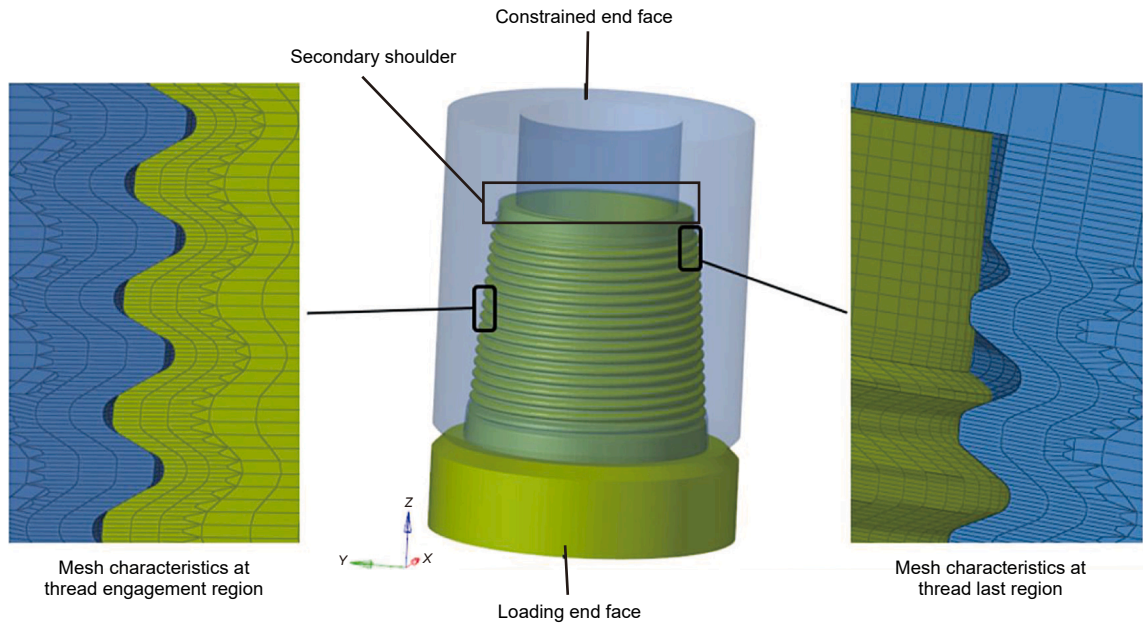


Fig. 20. 3D FE model of the double-shoulder drill pipe joint.

Table 5
Loading conditions.

Step	Load type	NC50 joint	DS50 joint
1	Make-up torque, kN·m	43.1	64.2
2	Axial tensile, kN	1600	1600
3	Bending moment, kN·m	15	15
4	Internal pressure, MPa	30	30

this numerical model, three-dimensional point cloud data are generated to discretize the rough surface, as shown in Fig. 15. In this step, the discretization method treats each point as a cylindrical element, with the height corresponding to the Z-coordinate in the point cloud data. The number of cylindrical elements matches the number of points in the point cloud. Next, the two

discretized rough surfaces are combined to form a three-dimensional grid model of the sealing interface, as illustrated in Fig. 16. This grid model represents a discretized representation of the equivalent rough surface from Fig. 15, converting the continuous curved asperity morphology into discrete cuboid units. The core objective is to simplify the quantitative calculation of the sealing gap through regularized modeling, as higher discretization accuracy produces more precise gap values.

The point cloud of the upper rough surface is denoted as Z^1 , and that of the lower rough surface as Z^2 . A rectangular sealing interface with length L (μm), width W (μm), and cylinder height u_{ij} (μm) is studied. Here, u_{ij} is the height coordinate of each point, and the length and width of each cylinder are l_c (μm). The indices i and j denote the planar coordinates of the points in the X and Y directions, respectively. The initial contact state, where the two

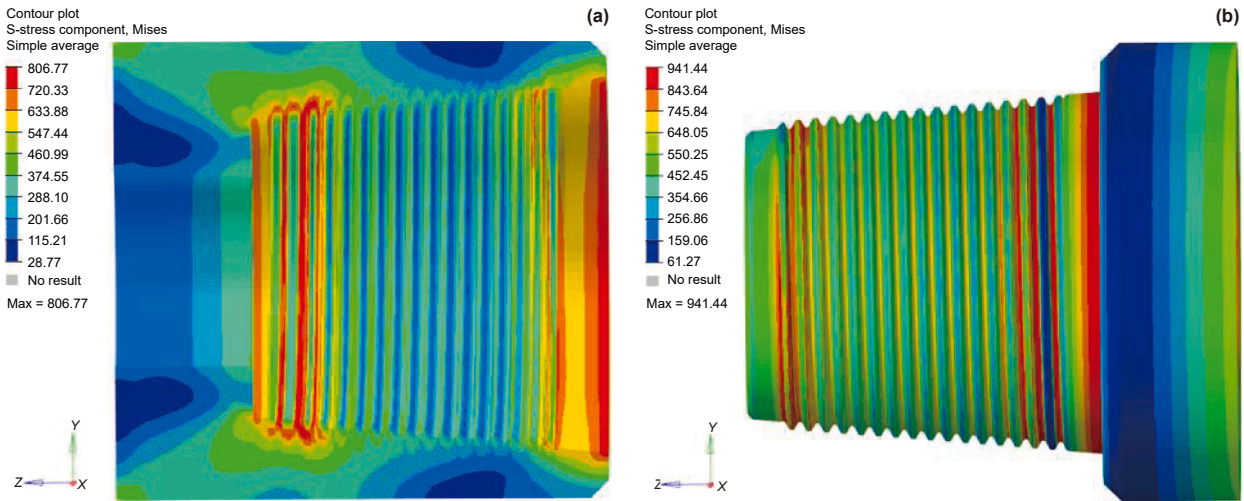


Fig. 21. von Mises stress analysis of the double-shoulder drill pipe joint under complex loads: (a) Box (titanium), (b) pin (steel).

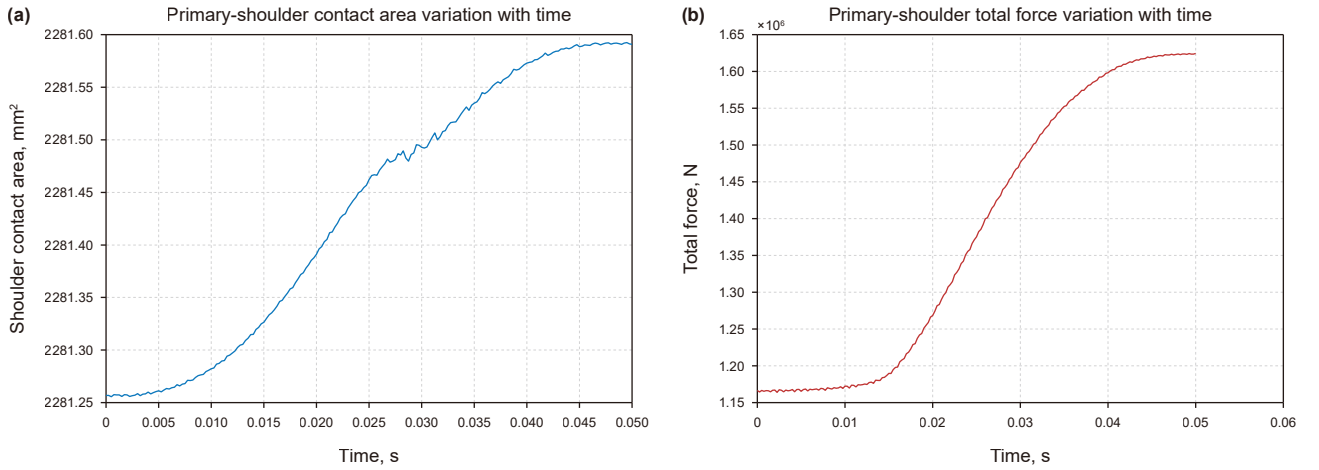


Fig. 22. Primary shoulder engagement surface: (a) Variation in contact area, (b) variation in contact force.

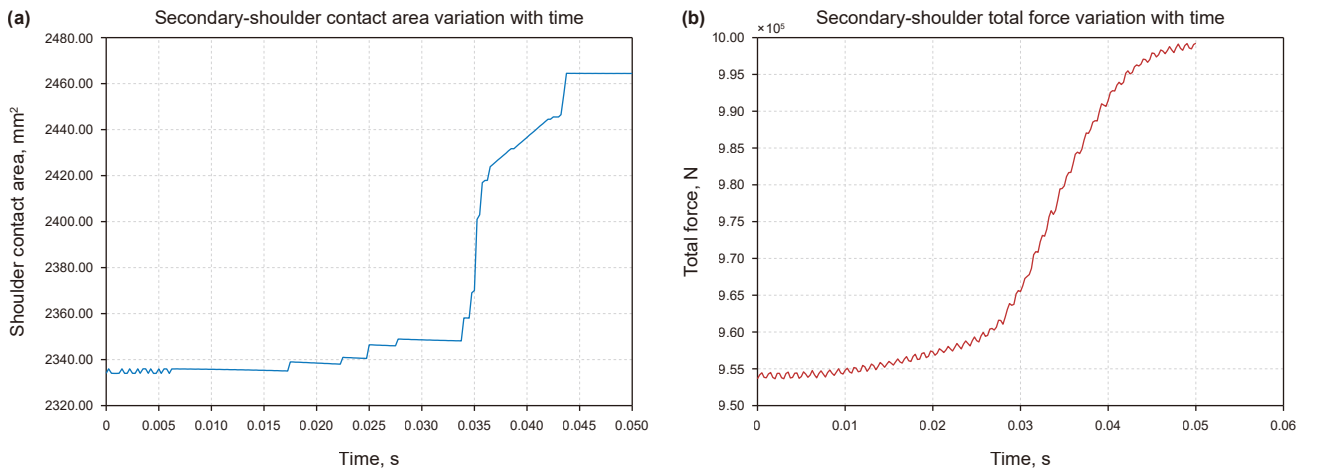


Fig. 23. Secondary shoulder engagement surface: (a) Variation in contact area, (b) variation in contact force.

rough surfaces first touch without asperity deformation, is defined by the critical height h_0 (μm) of the sealing interface.

Based on the three-dimensional grid model of the sealing interface, the proportion of the solid part of the sealing interface in the initial contact state is given by:

$$\frac{V_{\text{solid}}(0)}{V_0(0)} = \frac{l_c^2 \sum_{i=1}^n \sum_{j=1}^n (u_{ij}^1 + u_{ij}^2)}{LWh_0} \quad (24)$$

where n is the number of points in the X or Y direction of the rough surface point cloud data, $V_{\text{solid}}(0)$ is the initial volume of the solid part of the sealing interface (μm^3), $V_0(0)$ is the initial volume of the sealing interface (μm^3). The initial porosity $\phi(0)$ of the sealing interface is:

$$\phi(0) = 1 - \frac{V_{\text{solid}}(0)}{V_0(0)} = 1 - \frac{l_c^2 \sum_{i=1}^n \sum_{j=1}^n (u_{ij}^1 + u_{ij}^2)}{LWh_0} \quad (25)$$

where $h_0 = \max(u_{ij}^1 + u_{ij}^2)$ is the critical height of the two rough surfaces, which increases with surface roughness.

When the upper shoulder surface moves downward, the interference amount at the shoulder mating surface is λ , and the

height of the sealing interface decreases from h_0 to $(h_0 - \lambda)$. Consequently, V_0 becomes $LW(h_0 - \lambda)$. Assuming that the deformation of a single asperity has minimal impact on adjacent asperities (Liu et al., 2023) and that the solid volume V_{solid} remains unchanged before and after deformation, the corresponding porosity $\phi(\lambda)$ is:

$$\phi(\lambda) = \frac{LW(h_0 - \lambda) - [1 - \phi(0)]LWh_0}{LW(h_0 - \lambda)} = 1 - \frac{[1 - \phi(0)]h_0}{h_0 - \lambda} \quad (26)$$

3.5. Establishment of the shoulder sealing model

Metal-to-metal seals in drill pipe joints rely on axial preload to create interfacial contact pressure. The annular sealing zone (red area in Fig. 17) experiences:

Due to its geometric shape, the shoulder sealing forms an annular closed region, as shown in Fig. 17. The red area in Fig. 17 represents the sealing interface, with an inner diameter R_i (m) and an outer diameter R_o (m).

For the primary shoulder:

Internal pressure (p_{pi}) corresponds to the chamfer clearance pressure (P_a), external pressure (p_{po}) equals the joint's annular pressure (P_a).

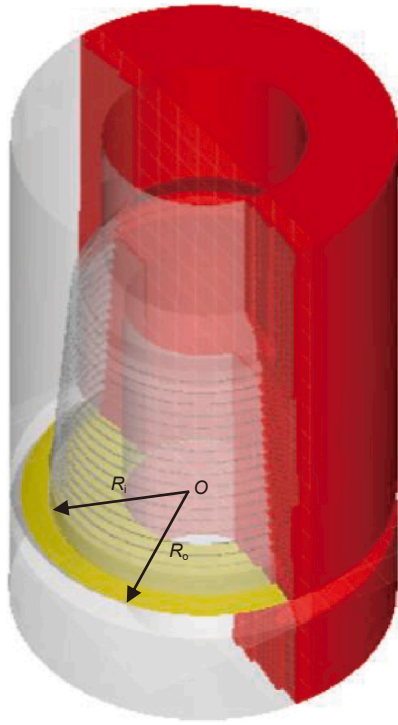


Fig. 24. Schematic of shoulder sealing in the NC50 steel-titanium heterogeneous drill pipe joint.

For the secondary shoulder:

Internal pressure (p_{si}) represents the tool joint bore pressure (Pa), external pressure (p_{so}) matches the chamfer clearance pressure (Pa).

Under these conditions, fluid leakage occurs across the sealing interface driven by the pressure differential. According to the principle of mass conservation, the mass of fluid entering and leaving the shoulder mating surface per unit time during leakage is equal:

$$\frac{\partial \rho}{\partial t} = -\nabla \cdot [\rho v(\lambda)] \quad (27)$$

where v is the seepage velocity (m/s), ρ is the drilling fluid density (kg/m^3), $\nabla \cdot$ is the divergence operator. From Darcy's law:

$$v(\lambda) = \frac{K(\lambda)}{\mu} (\nabla p_l) \quad (28)$$

where μ is the fluid viscosity (Pa·s), K is the permeability of the shoulder (m^2), p_l is the environmental pressure (Pa), ∇ is the gradient operator.

Studies on the flow of single-phase Newtonian fluids in porous media suggest that permeability is dependent solely on the geometric structure of the porous medium and is independent of fluid properties. The empirical formula for permeability considering the tortuosity of the sealing interface is as follows (Chen, 2020):

$$K(\lambda) = \frac{c\phi(\lambda)^3}{m(\lambda) \left(\frac{1.896}{R} \right)^2} \quad (29)$$

$$m(\lambda) = \frac{1}{2} \left[1 + \frac{1}{2} \sqrt{1 - \phi(\lambda)} + \sqrt{1 - \phi(\lambda)} \frac{\sqrt{\left(\frac{1}{\sqrt{1 - \phi(\lambda)}} - 1 \right)^2 + \frac{1}{4}}}{1 - \sqrt{1 - \phi(\lambda)}} \right] \quad (30)$$

where m is the tortuosity, c is the Knudsen empirical coefficient, typically ranging from 5 to 6, combining Eqs. (28) and (27) yields:

$$\frac{\partial \rho}{\partial t} = -\nabla \cdot \left[\rho \frac{K(\lambda)}{\mu} (\nabla p_l) \right] \quad (31)$$

Assuming the fluid density is time-independent, the left-hand side of Eq. (31) becomes zero, leading to:

$$\nabla^2 p_l = \frac{\partial^2 p_l}{\partial x^2} + \frac{\partial^2 p_l}{\partial y^2} + \frac{\partial^2 p_l}{\partial z^2} = 0 \quad (32)$$

Transforming Eq. (32) into cylindrical coordinates gives:

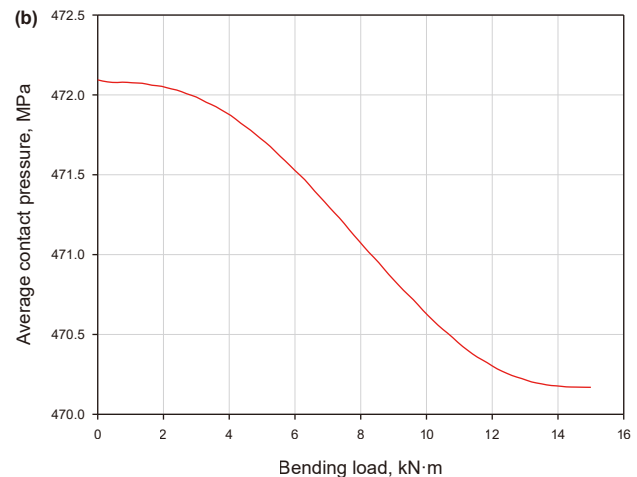
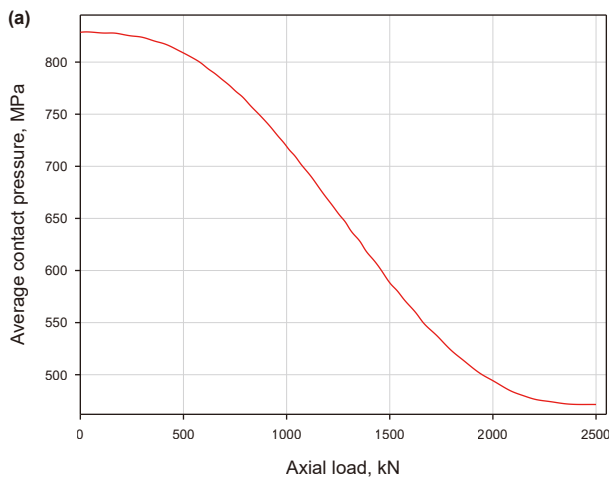


Fig. 25. Average contact pressure on the shoulder mating surface under varying external loads: (a) Average contact pressure under axial tensile force, (b) average contact pressure under bending moment.

$$\frac{d^2 p_1}{dr^2} + \frac{1}{r} \frac{dp_1}{dr} = \frac{d}{dr} \left(r \frac{dp_1}{dr} \right) = 0 \quad (33)$$

The boundary conditions are:

$$p_1(R_i) = p_i, p_1(R_o) = p_o \quad (34)$$

Combining Eqs. (33) and (34), the radial pressure gradient is expressed as:

$$\frac{dp_1}{dr} = \frac{p_o - p_i}{r \ln \frac{R_o}{R_i}} \quad (35)$$

Substituting Eq. (35) into Darcy's law Eq. (28), the seepage velocity is:

$$v(\lambda) = \frac{K(\lambda)}{\mu} \frac{dp_1}{dr} = \frac{K(\lambda)(p_o - p_i)}{\mu r \ln \frac{R_o}{R_i}} \quad (36)$$

The shoulder leakage rate $Q(\lambda)$ is the product of the seepage velocity and the shoulder cross-sectional area:

$$Q(\lambda) = 2\pi r(h_0 - \lambda)v(\lambda) = \frac{2\pi(h_0 - \lambda)K(\lambda)(p_o - p_i)}{\mu \ln \frac{R_o}{R_i}} \quad (37)$$

where $Q(\lambda)$ is the shoulder leakage rate (m^3/s).

Based on the boundary conditions of the primary and secondary shoulders, the respective leakage rates can be calculated using Eq. (37).

3.6. Comparison between sealing model and experimental results

In the study of metal-to-metal sealing, Ernens et al. (2019) investigated the effects of surface topography, thread compounds, and coatings on sealing through experiments and simulations. As shown in Fig. 18, the Shell Sealing Mock-up Rig (SSMUR) used annular seal specimens from P110 tubing. A liquid pressure of 70 MPa was applied, and the contact load was gradually reduced to detect leakage onset. Results showed surface micro-asperity features greatly impact sealing. Specimens with distinct micro-asperity structures (e.g., FRS11) exhibited lower initial leakage due to the formation of localized high contact stresses that created flow barriers. In contrast, those with higher surface roughness (e.g., FRS08) had higher leakage rates because they lacked effective flow-blocking structures. Thread compounds improved sealing by filling microscopic gaps and reducing wear.

To validate the theoretical leakage rate model (Eq. (37)) proposed in this study, a comparative analysis was conducted

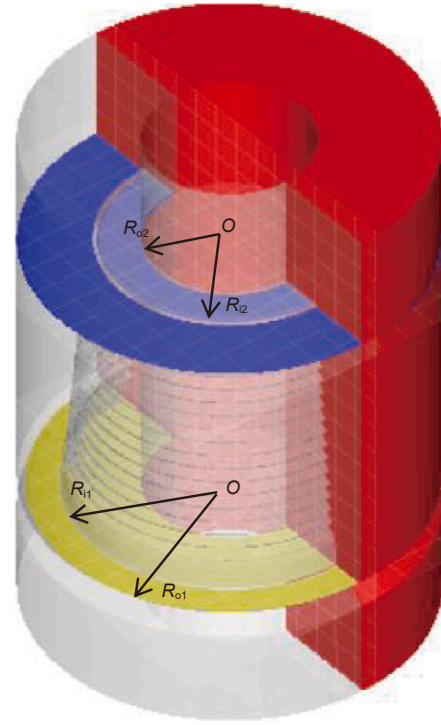


Fig. 27. Schematic of double-shoulder sealing.

between the experimental data from the literature (Table 3) and the model predictions, as shown in Fig. 19.

Since Ernens et al. did not provide specific surface roughness parameters, parameter adjustments were conducted to achieve an optimal fit between the model and experimental data. It was found that when the roughness was set to $1.78 \mu\text{m}$, the model fitting curve (triangles–green curve) aligned well with the unloading curve of FRS08 (Fig. 10(a) (Ernens et al., 2019), showing a significant increase in leakage rate with decreasing contact intensity. When the roughness was set to $0.44 \mu\text{m}$, the model results (squares–blue curve) matched the trend of the unloading curve of FRS11 (Fig. 10(b)) (Ernens et al., 2019), characterized by a consistently low leakage rate even under low contact intensity. Although differences in experimental conditions (such as pressure medium and material properties) exist, the predicted trends in leakage rate from the model align with the experimental data. Particularly near the critical contact intensity corresponding to the onset of leakage,

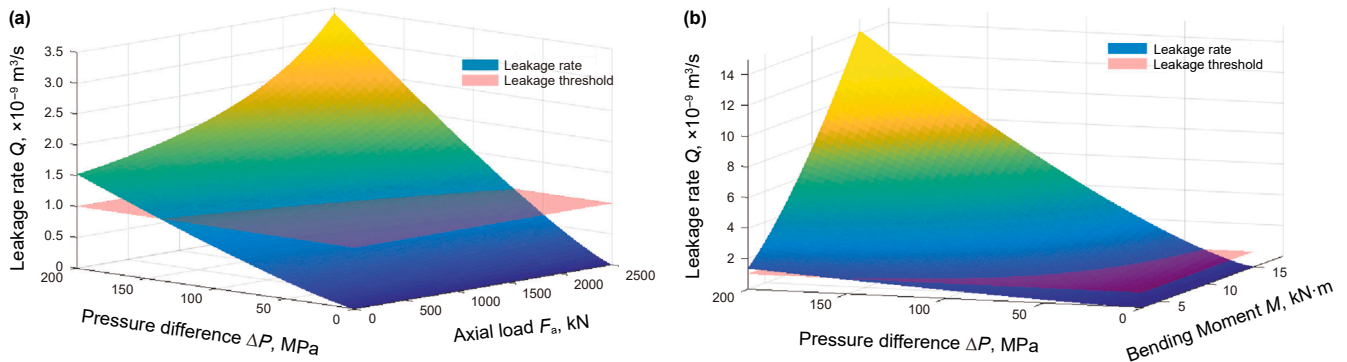


Fig. 26. Leakage rate trends under different working loads: (a) Leakage rate under varying pressure differential and axial tensions, (b) leakage rate under varying pressure differential and bending moments.

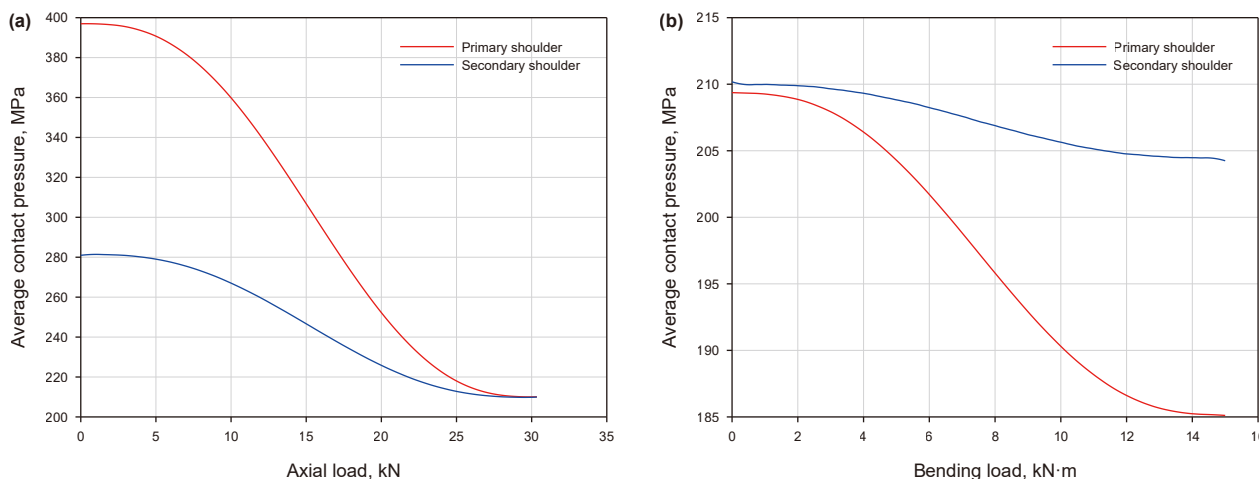


Fig. 28. Average contact pressure on double-shoulder mating surfaces under varying external loads: (a) Average contact pressure under axial tension, (b) average contact pressure under bending moment.

both the simulation and experimental results exhibit a rapid increase in leakage.

This comparison validates the effectiveness of the proposed model in capturing the flow behavior at the sealing interface, especially its sensitivity to surface topography. Furthermore, under the experimental conditions of this study, with $R_a = 1.8 \mu\text{m}$, the predicted leakage rate at the secondary shoulder of the double-shoulder joint, as a function of contact pressure is shown by the red curve in the figure. The leakage rate gradually decreases as the contact pressure increases.

4. Analysis of stress characteristics in steel-titanium heterogeneous double-shoulder drill pipe joints

4.1. Design of the 3D model for double-shoulder joints

According to API RP 5A3 standards, thread compound should be applied to the threaded surfaces of drill pipe joints before make-up. This serves primarily as a lubricant but also fills thread gaps, providing auxiliary sealing. However, in the complex downhole loading environments of ultra-deep and extreme-depth wells, the thread compound is subjected to extrusion and redistribution, and high temperatures can cause partial evaporation, diminishing or even eliminating its auxiliary sealing function. Once the thread compound loses its sealing capability, high-pressure drilling fluid can infiltrate the threaded engagement surfaces. Under the combined action of corrosive media and tensile stress, stress corrosion cracks may initiate at the thread roots due to stress concentration. These cracks can propagate, eventually leading to thread leakage.

Double-shoulder drill pipe joints incorporate a secondary shoulder in addition to the primary shoulder, forming a dual-shoulder structure. The secondary shoulder helps distribute part of the load during operation, enhancing both the sealing and torsional resistance of the joint. Moreover, the presence of the secondary shoulder disperses the load, reducing stress concentration at the thread roots and delaying the initiation and propagation of stress corrosion cracks.

The double-shoulder drill pipe joint selected for this study features a smooth inner diameter, which promotes stable fluid flow and minimizes erosion. Designated as DS50, its structural parameters are listed in Table 4.

4.2. FE model of the steel-titanium heterogeneous double-shoulder drill pipe joint

Taking a double-shoulder drill pipe joint as the analysis object, a three-dimensional finite element model of the joint was established, as shown in Fig. 20. The DS50 drill pipe joint model comprises a total of 737,880 elements and 825,360 nodes. The element type selected for the simulation was C3D8I. Previous analysis has shown that in the single-shoulder design, the joint with a steel pin and a titanium box exhibits the largest fluctuation in stress along the thread tooth path, which can lead to weakened bending fatigue resistance. To further investigate its performance, the double-shoulder drill pipe joint was designed as a heterogeneous steel-titanium assembly—specifically, the box is made of titanium alloy while the pin is made of steel. Detailed material parameters are presented in Section 2.1.

4.3. Comparison of load-bearing performance between single- and double-shoulder steel-titanium heterogeneous drill pipe joints under complex loads

To investigate the influence of the secondary shoulder structure on joint performance and load distribution uniformity, the mechanical characteristics of single- and double-shoulder drill pipe joints under combined loading were analyzed and compared. All joints were pre-tightened to the recommended makeup torque, with identical downhole loading conditions applied (Table 5).

The results in Fig. 21 show that after pre-tightening, peak stresses in both the pin and box of the double-shoulder joint under axial tension are concentrated at the first engaged thread near the shoulder. The peak stresses in the titanium (box) and steel (pin) components remain below their respective strength limits of 856.0 and 1020.6 MPa, confirming that the double-shoulder design meets the load requirements for this scenario.

Under complex loading, the contact pressure results for the steel-titanium dissimilar double-shoulder joint are presented in Fig. 22(a) and (b) and Fig. 23(a) and (b). Calculations show that the average contact pressure on the primary shoulder interface is 779.52 MPa with a contact area of 2087.56 mm², while the secondary shoulder exhibits an average contact pressure of 467.41 MPa and a contact area of 2464.28 mm².

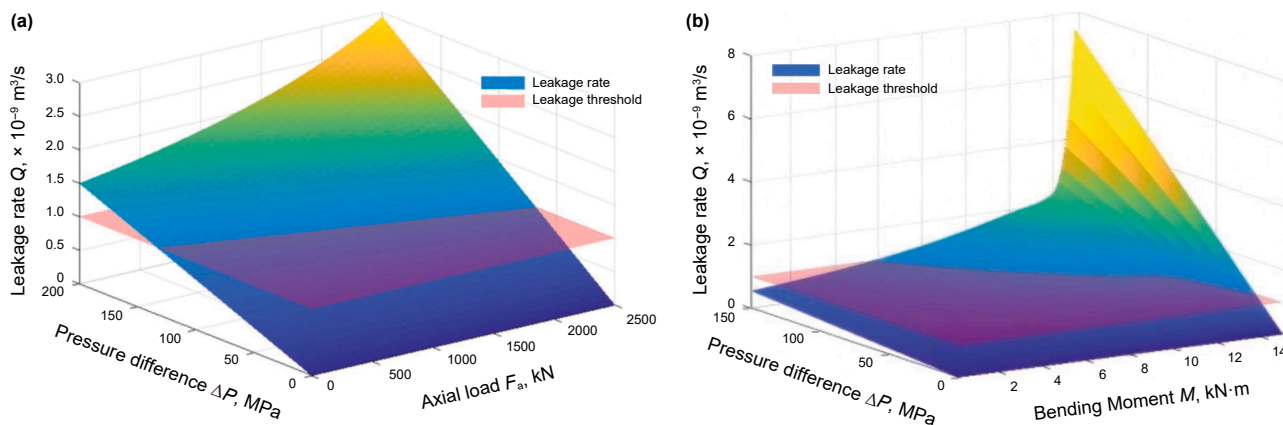


Fig. 29. Leakage rate trends for double-shoulder joints under varying pressure differentials, axial tensions, and bending moments: **(a)** Leakage rate under varying pressure differential and axial tensions on secondary shoulder, **(b)** leakage rate under varying pressure differential and bending moments on primary shoulder.

5. Sealing performance evaluation of steel-titanium heterogeneous drill pipe joints under complex loads

During downhole operations, drill pipe joints are subjected to a combination of complex loading conditions, these include internal and external pressure, axial force, bending moment, and torque. Together, they create an extremely harsh working environment. When exposed to tensile and bending loads, the contact pressure on the shoulder surfaces decreases significantly, this leads to a notable reduction or even complete loss of sealing capacity. Furthermore, excessive differential pressure between the inside and outside of the joint also adversely affects its sealing performance.

As noted in Section 2.2, the bending fatigue resistance of the thread teeth in steel (pin)-titanium (box) heterogeneous joints is comparatively weaker. In engineering practice, structural failure often initiates at such vulnerable locations. Therefore, investigating the sealing performance of these joints can directly reveal their risk of sealing failure under complex loads and provide a targeted theoretical basis for design optimization. In light of the above, this section focuses on steel (pin)-titanium (box) heterogeneous joints. To address this, the characteristics of contact pressure distribution across the engaged shoulder surfaces are analyzed, and, together with a leakage rate model for the sealing interface, an evaluation method is established for assessing the sealing performance of steel (pin)-titanium (box) heterogeneous drill pipe joints under complex loading conditions.

5.1. Sealing performance evaluation of steel-titanium heterogeneous drill pipe joints

The NC50 steel-titanium heterogeneous drill pipe joint was subjected to specific external loads: a make-up torque of 43.1 kN·m, an axial tension of 2500 kN, and a bending moment of 15.0 kN·m. The shoulder surface roughness was set to 2.0 μm , with autocorrelation coefficients $\beta_x = 1$ and $\beta_y = 50$, and a fluid viscosity of 0.05 Pa·s (Chen et al., 2022). The shoulder mating surface had an inner diameter $R_i = 135.0$ mm and an outer diameter $R_o = 154.0$ mm (Fig. 24). Under bending, the contact pressure increased on the compression side and decreased on the tension side (highlighted in yellow in Fig. 24). Since reduced contact pressure compromises sealing, this study focuses on the tension side. Fig. 25 shows the relationship between average contact pressure and external loads on the tension side.

In Section 3, the leakage rate $Q(\lambda)$ for the shoulder was derived using the contact model and sealing theory, as given in Eq. (37). The leakage rate is influenced by the pressure differential ($\Delta p = p_o - p_i$), asperity compression λ , and contact surface properties. In practice, external loads like axial tensile forces and bending moments alter the contact pressure, affecting leakage channel height. Thus, the leakage rate depends on both the pressure differential and external loads.

Building on the contact pressure results in Fig. 25, the leakage rate formula was refined using curve fitting. Here, leakage rate $Q(\lambda)$ is the dependent variable, where Q is the volumetric leakage rate and λ is the nondimensional parameter. Pressure differential Δp and axial tensile force F_a , where Δp is the pressure difference across the seal and F_a is the applied axial force, are independent variables. Known parameters, results of Fig. 25, and Eq. (37) were combined. The leakage rate became a function of Δp and F_a . To improve analysis and applicability, the equation was non-dimensionalized, removing unit dependencies and increasing generalizability.

The characteristic leakage rate (Q_1) was set at $1 \times 10^{-9} \text{ m}^3/\text{s}$ for scaling. The characteristic pressure differential (Δp_i) was set at 1 MPa, representing typical system conditions. The characteristic axial tensile force (F_{a0}) was set at 1000 kN. Using these values, the nondimensional variables were defined as shown in Eq. (38):

$$\pi_1 = \frac{Q_1}{Q_i}, \pi_2 = \frac{\Delta p}{\Delta p_i}, \pi_3 = \frac{F_a}{F_{a0}} \quad (38)$$

where π_1 is the nondimensional leakage rate, π_2 is the nondimensional pressure differential, and π_3 is the nondimensional axial tensile force.

Substituting the original variables in Eq. (37) with their nondimensional counterparts and incorporating the characteristic relationships yields Eq. (39):

$$Q_1 = Q_i \cdot \pi_1, \Delta p = \Delta p_i \cdot \pi_2, F_a = F_{a0} \cdot \pi_3 \quad (39)$$

First, the curve between average contact pressure and axial tensile load, shown in Fig. 25(a), was fitted to show average contact pressure p as a function of axial tensile load. Next, this expression for contact pressure was substituted into Eqs. (27)–(37) in Section 3.5 to obtain a formula relating the leakage rate to the axial tensile load. Finally, using Eq. (39), a dimensionless form of the leakage rate as a function of axial tensile load—given by Eq. (40)—was reached.

$$\begin{aligned}\pi_1 = & -2.46 \times 10^{-7} \pi_2^2 + 1.75 \times 10^{-2} \pi_2 - 2.01 \times 10^{-10} \pi_3^2 \\ & - 1.05 \times 10^{-6} \pi_3 + 1.97 \times 10^{-8} \pi_2 \cdot \pi_3 + 3.44 \times 10^{-4}\end{aligned}\quad (40)$$

By substituting specific values for the pressure differential and axial tensile force into Eqs. (38)–(40), the corresponding leakage rates were calculated under varying conditions. Comparing these results with the leakage threshold determined whether the shoulder was prone to leakage.

Using the same methodology, the leakage rate Q_2 , which depends on the pressure difference (Δp) and the bending moment (M), is fitted, by taking the characteristic bending moment value, new dimensionless variables are defined. This process results in Eq. (41):

$$\gamma_1 = \frac{Q_2}{Q_i}, \gamma_2 = \frac{\Delta p}{\Delta p_i}, \gamma_3 = \frac{M}{M_0} \quad (41)$$

where γ_1 is the nondimensional leakage rate, γ_2 is the nondimensional pressure differential, and γ_3 is the nondimensional bending moment.

The relationship curve between average contact pressure and bending moment in Fig. 25(b) was fitted to express the average contact pressure p as a function of the bending moment. The resulting expression for contact pressure was then substituted into Eqs. (27)–(37) in section 3.5 to derive an analytical formula relating leakage rate to bending moment. Finally, using Eq. (41), the dimensionless form of the leakage rate as a function of bending moment—given by Eq. (42)—was obtained.

$$Q_2 = Q_i \cdot \gamma_1, \Delta p = \Delta p_i \cdot \gamma_2, M = M_0 \cdot \gamma_3 \quad (42)$$

Further simplification yielded the nondimensional form shown in Eq. (43):

$$\begin{aligned}\gamma_1 = & -2.25 \times 10^{-8} \gamma_2^2 + 7.02 \times 10^{-6} \gamma_2 - 1.17 \times 10^{-11} \gamma_3^2 - 1.12 \\ & \times 10^{-9} \gamma_3 + 2.08 \times 10^{-9} \gamma_2 \cdot \gamma_3 - 2.79 \times 10^{-3}\end{aligned}\quad (43)$$

It should be noted that the application of the bending moment is based on the precondition of an axial tension of 2500 kN; thus, the results for the bending moment are influenced by the preceding axial tension data.

By substituting different values of bending moment into Eqs. (41)–(43), the corresponding leakage rates under various pressure differences and bending moments can be obtained. Comparing these results with the leakage threshold determines whether the joint shoulder leaks under these conditions.

Based on Fig. 25 and Eqs. (38)–(43), the leakage rate at the shoulder engagement surface of the NC50 steel-titanium heterogeneous drill pipe joint under various loads and pressure differentials was analyzed using an interpolation method. The results are presented in Fig. 26, which illustrates the variation of leakage rate under different pressure differentials, axial tensile loads, and bending moments.

As indicated by Eq. (40) or Fig. 26(a), when the axial tensile load reaches 2500 kN, the critical pressure differential at which leakage initiates is 68.5 MPa. Subsequently, with the additional application of a bending moment, Eq. (43) or Fig. 26(b) shows that under an axial load of 2500 kN and a bending moment of 8 kN·m, the critical pressure differential for leakage occurrence decreases to 57.9 MPa.

5.2. Influence of double structure on sealing performance of steel-titanium heterogeneous drill pipe joints

An external load was applied to the steel-titanium heterogeneous double-shoulder drill pipe joint. The load consisted of a make-up torque of 64.2 kN·m, an axial tension of 2500 kN, and a bending moment of 15.0 kN·m. These load conditions were used to analyze the influence of different external loading conditions on the sealing performance of the shoulders. The primary shoulder surface was assigned a roughness of 2.0 μm , autocorrelation coefficients of $\beta_x = 1$, $\beta_y = 50$, and the fluid viscosity was set to 0.05 Pa·s. The inner contact radius R_{i1} of the primary shoulder is 135.0 mm, and its outer contact radius R_{o1} is 154.0 mm; the inner contact radius R_{i2} of the secondary shoulder is 104.0 mm, and its outer contact radius R_{o2} is 70.0 mm. A schematic diagram of the specific structure is shown in Fig. 27. Due to the bending moment, the contact pressure on the tension side of the shoulders is lower than that on the compression side. Consequently, the tension side represents the weak point for sealing performance. In Fig. 27, the tension side of the primary shoulder is marked in yellow, and that of the secondary shoulder is marked in blue. To assess sealing integrity, the subsequent analysis focuses on the tension sides of both shoulders. The resulting average contact pressures on their engagement surfaces are presented in Fig. 27.

The double-shoulder joint has a more complex shoulder structure than the standard steel-titanium drill pipe joint, its sealing performance depends on both the primary and secondary shoulders. As shown in Fig. 28(a) and (b), under axial tension, the contact force on both engagement surfaces gradually decreases. This means axial tension lowers the shoulders' load-bearing capacity and the secondary shoulder has a lower average contact pressure. As the bending moment increases, the average contact pressure on the tension side decreases for both shoulders, with the primary shoulder's contact force dropping faster. In this structure, the primary shoulder primarily provides sealing resistance against axial tension, while the secondary shoulder is more adaptable to bending moments.

In summary, for the contact condition of the secondary shoulder engagement surface subjected to axial tension on the tension side, a quantitative relationship model between leakage rate, pressure difference, and axial tension is established based on the geometric parameters, material properties, and experimentally measured data of the secondary shoulder. In this study, the leakage rate $Q_{\text{secondary}}(\lambda)$ is treated as the response variable, with the pressure difference (Δp) and axial tension (F_a) as independent variables. Incorporating the variation characteristics of the contact pressure on the secondary shoulder, the leakage rate is modeled as a function of contact pressure and pressure difference. The fundamental formula for leakage rate remains Eq. (37).

Based on experimental data and the theoretical model, the leakage rate of the secondary shoulder is derived as a function of pressure difference and axial tension. Dimensionless processing is then performed, as shown in Eq. (44):

$$\alpha_1 = \frac{Q_{\text{secondary}}}{Q_i}, \alpha_2 = \frac{\Delta p}{\Delta p_i}, \alpha_3 = \frac{F_a}{F_{a0}} \quad (44)$$

where α_1 is the nondimensional leakage rate, α_2 is the nondimensional pressure differential, and α_3 is the nondimensional axial tensile force.

The original variables $Q_{\text{secondary}}$, Δp , and F_a in the formula are replaced with dimensionless variables, and characteristic value relations are introduced, as in Eq. (45):

$$Q_{\text{secondary}} = Q_i \cdot \alpha_1, \Delta p = \Delta p_i \cdot \alpha_2, F_a = F_{a0} \cdot \alpha_3 \quad (45)$$

Similarly, the relationship between the average contact pressure at the secondary shoulder and the axial tensile load, as shown in Fig. 28(a), was fitted to represent the average contact pressure p as a function of the axial tensile load. This expression was incorporated into Eqs. (27)–(37) from section 3.5 to establish the leakage rate as a function of axial tensile load. Combining the result with Eq. (45) yielded the corresponding dimensionless leakage rate equation, Eq. (46), in terms of axial tensile load.

$$\alpha_1 = 654\alpha_2^2 - 20.3\alpha_2 + 0.0207\alpha_3^2 - 0.106\alpha_3 + 7.65\alpha_2 \cdot \alpha_3 + 0.16 \quad (46)$$

Using Eqs. (44)–(46), the leakage rates corresponding to different pressure differences and axial tensions can be obtained. Comparing these results with the leakage threshold determines whether the shoulder leaks under these conditions.

For the contact condition of the primary shoulder subjected to a bending moment on the tension side, a relationship model between leakage rate, pressure difference, and bending moment is established in the same manner. Here, the leakage rate $Q_{\text{primary}}(\lambda)$ is the response variable, with pressure difference ($p_o - p_i = \Delta p$) and bending moment (M) as independent variables. Incorporating the variation characteristics of the contact pressure on the primary shoulder and based on experimental data and the theoretical model, the leakage rate of the primary shoulder is derived as a function of pressure difference and bending moment. Following the same method, the leakage rate $Q_{\text{primary}}(\lambda)$, as a function of pressure difference Δp and bending moment M , is fitted, and new dimensionless variables are defined, resulting in Eq. (47):

$$\beta_1 = \frac{Q_{\text{primary}}}{Q_i}, \beta_2 = \frac{\Delta p}{\Delta p_i}, \beta_3 = \frac{M}{M_0} \quad (47)$$

where β_1 is the nondimensional leakage rate, β_2 is the nondimensional pressure differential, and β_3 is the nondimensional bending moment.

The original variables Q_{primary} , Δp , and M are replaced with dimensionless variables, and characteristic value relations are introduced, as in Eq. (48):

$$Q_{\text{primary}} = Q_i \cdot \beta_1, \Delta p = \Delta p_i \cdot \beta_2, M = M_0 \cdot \beta_3 \quad (48)$$

For the relationship between average contact pressure and bending moment illustrated in Fig. 28(b), a functional fit was performed to express p in terms of the bending moment. This contact pressure expression was again substituted into Eqs. (27)–(37), resulting in a leakage rate formula that depends on the bending moment. By applying Eq. (48), the original variables Q_{primary} , Δp , and M were replaced with their dimensionless counterparts, and characteristic scaling relations were introduced, resulting in the dimensionless leakage rate equation, Eq. (49), expressed as a function of bending moment.

$$\beta_1 = -1.25 \times 10^{-8}\beta_2^2 + 2.29 \times 10^{-8}\beta_2 + 3.65 \times 10^{-10}\beta_3^2 - 4.7 \times 10^{-9}\beta_3 + 1.47 \times 10^{-7}\beta_2^2\beta_3 \quad (49)$$

Using Eqs. (47)–(49), the leakage rates corresponding to different pressure differences and bending moments can be obtained. Comparing these results with the leakage threshold determines whether the shoulder leaks under these conditions.

Based on Fig. 28 and Eqs. (44)–(49), the leakage rates of the double-shoulder joint under various loads and pressure differentials were analyzed, with the results shown in Fig. 29. According to Eq. (46) or Fig. 29(a), when leakage occurs under an axial tensile load of 2500 kN, the secondary shoulder can withstand a pressure differential of 77.3 MPa. With the additional application of a bending moment, Eq. (49) or Fig. 29(b) indicates that under an axial load of 2500 kN and a bending moment of 8 kN·m at leakage initiation, the primary shoulder can sustain a pressure differential of 73.9 MPa.

Compared to the NC50 drill pipe joint, under critical leakage conditions and an axial tensile load of 2500 kN, the NC50 joint withstands a pressure differential of 68.5 MPa at the shoulder, while the DS50-H secondary shoulder sustains 77.3 MPa—an improvement of 12.8% in sealing performance.

Under an applied bending moment of 8 kN·m, the NC50 joint tolerates a pressure differential of 57.9 MPa, whereas the DS50 primary shoulder maintains 73.9 MPa, representing a 27.6% enhancement in sealing capability.

When the leakage rate $Q(\lambda) \leq 1 \times 10^{-9} \text{ m}^3/\text{s}$, the sealing performance of the shoulder is classified as satisfactory. Compared to the ideal sealing scenario, the sealing effectiveness of the single-shoulder design exhibits notable limitations. Additionally, the structural design of the single-shoulder joint allows drilling fluid in the drill string bore to directly contact the pin and box threads. This direct contact exposes the threads to increased risk of damage from the drilling fluid and mechanical loads, especially in the heterogeneous joint under complex loading conditions. The double-shoulder design demonstrates improved sealing performance compared to the single-shoulder joint, suggesting that the design meets performance objectives.

6. Conclusions

This study employs a three-dimensional elastoplastic finite element method to analyze the stress characteristics of drill pipe joints with different material combinations (titanium-steel and steel-titanium) under complex loading. Building on this, a double-shoulder drill pipe joint was designed to enhance performance. The sealing properties of the improved joint's shoulders were evaluated under various downhole loads using microscopic leakage mechanisms. The study identified how material hardness, surface roughness, and shoulder structural parameters influence leakage rates under differing contact pressures. These findings offer a theoretical foundation for evaluating and improving the sealing performance of heterogeneous drill pipe joints. The main conclusions are as follows:

- 1) Under complex loads (make-up torque, axial tension, bending moment, internal pressure), the peak stress in heterogeneous drill pipe joints is concentrated in the first engaged thread tooth adjacent to the shoulder. The peak stress values in both the titanium alloy and steel remain below their respective strength limits, meeting the design strength requirements. Among these, the steel-titanium heterogeneous single-shoulder joint exhibits the largest fluctuation range in thread tooth stress and possesses relatively weaker bending fatigue resistance.
- 2) A sealing model for the shoulders of heterogeneous drill pipe joints was built using numerical simulation of rough surfaces, an elastoplastic contact model, a three-dimensional grid model, and seepage mechanics formulas. Contact pressure and displacement are obtained via macroscopic three-dimensional (3D) elastoplastic finite element analysis, which numerically solves how surfaces deform under load. This model enables quantitative assessment of shoulder sealing performance under

a range of downhole conditions, such as different loads and pressure differences, offering a novel methodology for sealing performance research.

- 3) For the steel-titanium heterogeneous single-shoulder drill pipe joint, the critical pressure difference causing leakage at the shoulder is 68.5 MPa under an axial tension of 2500 kN. After applying a bending moment of 8 kN·m, this critical pressure difference decreases to 57.9 MPa. In the double-shoulder structure, however, the secondary shoulder has a critical leakage pressure difference of 77.3 MPa under 2500 kN axial tension, and the primary shoulder maintains a critical pressure difference of 73.9 MPa under the same axial tension combined with an 8 kN·m bending moment. Compared to the single-shoulder design, the double-shoulder joint demonstrates a 12.8% improvement in sealing performance under axial tension and a 27.6% improvement under bending moment. The secondary shoulder shows greater adaptability to bending moments, while the primary shoulder primarily provides sealing against axial tension. Their synergistic interaction enhances sealing stability under complex loads (make-up torque, axial tension, bending moment, internal pressure).
- 4) The study found that seal failure in the drill pipe joint occurs prior to strength failure, indicating that sealing integrity is a critical prerequisite for the safe design of drill pipe joints. Combined with the results from the sealing model, it is concluded that the double-shoulder structure enhances sealing performance through the cooperative action of the primary and secondary shoulders, providing theoretical support for the safe application of steel-titanium composite drill strings in ultra-deep wells.

CRedit authorship contribution statement

Feng Chen: Project administration, Data curation. **Jia-Qing Liu:** Writing – original draft. **Shang-Ying Yang:** Methodology. **Su-Na Zhang:** Methodology. **Jia-Nan Zhang:** Methodology, Investigation. **Wen-Chang Wang:** Formal analysis, Data curation. **Qin-Feng Di:** Project administration.

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Conflict of interest statement

The authors declare no conflict of interest in the submission of this manuscript. All authors have approved the manuscript for publication.

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