Study on dynamic sealing performance of combined sealing structure of telescopic type of downhole robot by using HTHP coupling method

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Abstract
The sealing performance will directly affect the operation of downhole robot under HTHP condition. Traditional analysis methods of sealing performance are that the temperature and pressure is loaded respectively. This can not really evaluate the sealing performance. Besides, the simulation process is: Step 1: pre-compress O-ring to produce contact force. According to the contact pressure, select the compression ratio to calculate the displacement of the slip ring. Step 2: load fluid pressure on the O-ring. This simulation method is to directly load pressure on the undeformed O-ring. However, the O-ring will deform after pre-compression. Therefore, this simulation method is not accurate. In order to make the simulation data more accurate, calculate the data of shape, stress, and strain of O-ring caused by pre-compression caused by assembly. Then, import the deformation body containing the real data of shape, stress, and strain into a new model. On the basis, establish the numerical simulation model of piston, piston guide rod, O-ring, and FTS-ring with HTHP loads is. Finally, calculate and analyze. When the compression ratio of

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the O-ring is about 14%, the sealing performance is good. What's more, the distribution of contact stress and Von Mises of the O-ring at 8.3 mm/s of motion speed are analyzed. The results show that the foot shaped combined sealing structure can keep a good dynamic sealing performance under HTHP condition. This paper provides a theoretical basis for the analysis of the dynamic sealing performance by using HTHP coupling method. In the analysis of sealing performance: the hydraulic pressure is loaded to the real model with the real shape, stress, and strain produced by the O-ring assembly. This can more accurately evaluate the sealing performance under HTHP condition. It also provides a reference for the dynamic sealing structure design of downhole tools.

**Keywords**
Combined sealing performance, HTHP coupling method, dynamic sealing, telescopic type of downhole robot, oil and gas drilling

**Introduction**

Coiled tubing drilling technology has many advantages, such as lower cost, less pollution,\(^1\) easier to be automation and intelligence, etc. than the conventional drilling.\(^2\) Due to the axial friction force, the coiled tubing is prone to buckle and “lock up” in horizontal wells.\(^3\) In view of this situation, many scholars have studied the technical bottleneck of “locking up” caused by the coiled tubing buckling. Among them, friction force reducing tools and friction force reducing chemicals are the focuses of the investigation.\(^4,5\) Both of the friction force reduction technologies can increase the extension length of the coiled tubing in horizontal wells. However, the extension length of the coiled tubing is still limited.

In order to solve the problem of the coiled tubing buckling, Jørgen\(^6\) proposed a concept of using downhole robots to carry out downhole operations in 1987 for the first time. So far, all the downhole robots can carry out simple operations, such as transportation of logging tool, milling operation of bridge plug, etc. There is not a report about the drilling operation using the downhole robot. It is found that, among the downhole robots, only telescopic downhole robots can meet the drilling requirements.\(^7\) However, the telescopic type of downhole robot is still in the development stage. The high temperature and high pressure resistance of the sealing structure is one of the main reasons restricting the application of telescopic type of downhole robot.

In the aspect of the influence of temperature and pressure on the sealing performance research, Li et al.\(^8\) analyzed the deformation and force condition of the O-ring under different oil pressure conditions range from 0 to 10 MPa. Yamabe et al.\(^9\) presented the effects of hydrogen pressure, ambient temperature, and pressure cycle pattern on fracture behavior of O-rings. Zhou et al.\(^10\) analyzed the effects of wedge-ring, hydrogen pressure, and swelling on the sealing performance. Kömmling et al.\(^11\) studied the sealing performance of the casks designed for the storage or transportation. The test temperature was 60–150°C. Troufflard et al.\(^12\) simulated the thermo-mechanical behavior of O-ring during the temperature cycle ranging from 20°C to 34°C. As the above researches show, most researchers...
studied the effect of temperature and pressure, respectively. However, the downhole working condition can always reach to high temperature of 150°C and high differential pressure of 20 MPa. Therefore, it is necessary to analyze the sealing performance of O-ring with HTHP coupling method. In the aspect of dynamic sealing performance research, Huang et al.\textsuperscript{13} studied the dynamic sealing performance of the O-ring in the pressure self-adaptive equalizer. Zhang and Xie\textsuperscript{14} studied the effects of pre-compression amount, fluid pressure, and friction coefficient on the static and dynamic sealing performances of the O-ring when the pressure is 5 MPa. Tadayoshi and Koji\textsuperscript{15} proposed a model of viscoelastic behaviors of the material with Maxwell- hyperelasticity.

In conclusion, current researches on the sealing performance of the O-ring mainly focus on the effect of working temperature or working pressure respectively, and the studies on the dynamic sealing performance of the O-ring did not be considered. So, there are few researches on the effect of HTHP coupling on the sealing performance of the O-ring. Usually, the downhole temperature can reach to 150°C and the downhole differential pressure of the downhole robot can reach to 20 MPa. Aimed at the sealing structure under the downhole operation condition of the oil and gas drilling, the analysis of the sealing performance with HTHP coupling method is carried out. Furthermore, the dynamic sealing performance with the HTHP coupling method is studied.

**Model of combination O-ring under HTHP condition**

*Constitutive model of rubber material*

It can be seen from Figure 1 that the telescopic type of downhole robot consists of left motion unit and right motion unit. Among them, one motion unit consists of telescopic hydraulic cylinder, gripping block, and gripping hydraulic cylinder. The sealing structure is on the hydraulic cylinder. Considering the requirements of dust prevention and sealing, the foot shaped combined sealing structure (FSCSS) is
selected as the sliding sealing structure of the telescopic type of downhole robot. FSCSSs are assembled on the gripping hydraulic cylinder and telescopic hydraulic cylinder (Figure 2).

The O-ring in FSCSS is made of oil resistant fluororubber with a diameter of 3.5 mm, which is a high elastic polymer material. The material's biggest characteristics are incompressibility, high elasticity and high nonlinearity, namely geometric nonlinearity, material nonlinearity, and contact nonlinearity.

The constitutive models of rubber models can be mainly divided into two types: phenomenological model of strain energy and statistical model of molecular chain network. Because the polynomial model of phenomenological theory has high precision, the Mooney Rivlin constitutive model of phenomenological theory is used to describe the mechanical properties of Hyper-elastic Materials under the large deformation conditions. The second parameter function expression is as followed:

\[
W = C_{10} \left( I_1 - 3 \right) + C_{01} \left( I_2 - 3 \right)
\]

where \( W \) is strain energy density, \( C_{10}, C_{01} \) are Mooney Rivlin model coefficients of material. \( I_1, I_2 \) are the first and second strain tensor invariants.

Mooney-Rivlin model material coefficient of fluororubber at different temperatures can be obtained, as shown in Table 1 and the friction coefficient is 0.2.  

**Material properties**

As the structure of FSCSS is consistent in its circumferential direction. The 2D axisymmetric model is selected and the two-dimensional finite element model of piston guide rod, piston, O-ring, and foot type slip ring (FTS-ring) is established. The structure and size of the model refer to FSCSS in the sealing manual. The
diameter of the O-ring is 3.5 mm, and the size of the FTS-ring matches the size of the O-ring. PTFE is selected for the FTS-ring. By consulting the literature and material experimental standards, the elastic modulus is 1000 MPa and Poisson’s ratio is 0.29. According to the analysis of heat transfer, the thermodynamic related coefficient of the combined sealing material is obtained, as is shown in Table 2.19

Comparing with the O-ring of FSCSS, the elastic modulus of piston and piston guide rod in the model is far greater than that of the O-ring. Therefore, the analytical rigid body is used and the grid division is shown in Table 3. In Table 3, the Unit type of CAX4R is a meshing method for circles. And, the Unit type of CAX3 is for meshing the irregular shape.

### Table 1. Material coefficient of fluororubber at different temperatures.

| Temperature (°C) | $C_{10}$ (MPa) | $C_{01}$ (MPa) |
|------------------|----------------|----------------|
| 20               | 1.87           | 0.47           |
| 150              | 0.63           | 0.158          |

### Table 2. Thermodynamic related coefficients of the combined sealing material.

|                   | FTS-ring | O-ring |
|-------------------|----------|--------|
| Temperature (°C)  |          |        |
| 20                | 0.00012  | 0.00023|
| 150               | 0.0002   | 0.0028 |
| Expansion coefficient | 0.34    | 0.25   |
| Heat conduction coefficient [W/(mm²·°C)] | $1.97 \times 10^3$ | $1.7 \times 10^3$ |
| Specific heat capacity [J/(T·kg)] |       |       |

### Table 3. Grid division of FSCSS.

| Part   | Unit number | Unit type | Grid type |
|--------|-------------|-----------|-----------|
| O-ring | 4548        | CAX4R     | Free mesh |
| FTS-ring | 5884      | CAX3      | Free mesh |

Pre-compression simulation model of FSCSS

The conventional simulation method has two steps: Step 1: pre-compress FSCSS to make the O-ring produce contact force. Then, apply radical displacement to the piston guide rod as the compression ratio of the O-ring range from 10% to 25%. Step 2: load pressure on the surface of O-ring and FSCSS, so as to simulate the pressure of the fluid on the O-ring (Figure 3).
Actually, when the O-ring is assembled, only a part of the O-ring is in contact with the fluid, and the other part of the O-ring is in contact with the piston guide rod and FTS-ring. Because the pressure load of the fluid can only be loaded at the position where the fluid contacts the O-ring. However, in the conventional numerical simulation method, the selection of the contact position between fluid and O-ring is strictly dependent on experience or estimation. Usually, the O-ring is divided into two semicircles, one of which is in contact with the fluid, and the pressure load of the fluid is only one semicircle. In other words, the pressure load of the fluid may not cover the contact area between the fluid and the O-ring completely, or the pressure load of the fluid may exceed the maximum contact area of the fluid. Therefore, the conventional analysis methods do not accurately simulate the real contact state between O-ring and fluid.

In order to make the simulation data more accurate, in this paper, the data of shape, stress, and strain of O-ring caused by pre-compression caused by assembly is calculated before the analysis of the dynamic sealing. Then, import the deformation body containing the real data of shape. This simulation method can accurately calculate the contact position between fluid and O-ring. The pressure load of the fluid can also be accurately loaded on the O-ring (Figure 4). Based on the simulation model, the calculated results are closer to the real state of O-ring. Therefore, the simulation method proposed in this paper is more accurate.

Under the condition of compression ratio of 10% and medium pressure of 10 MPa, the Mises stress distribution of O-ring and FTS-ring was obtained, as shown in Figures 5 and 6. It can be seen from the figure that the shape of O-ring in the conventional simulation model is very irregular and does not effectively contact with FTS-ring. The O-ring has penetrated into the gap between FTS-ring and the piston guide rod. It shows that the O-ring has been seriously damaged, which is not consistent with the actual use of O-ring. In the pre-compression simulation
model, the O-ring is in good contact with FTS-ring, and the O-ring is not damaged. Therefore, the pre-compression simulation model can more truly reflect the working state of O-ring. Then the calculation results will be more accurate.

**Effect of pre-compression ratio of O-ring on sealing performance**

The initial compression ratio is related to the section size and compression height, which can be expressed as follows:

\[
\Delta W = \frac{\Delta d_0}{d_0}
\]

(2)

where \(\Delta d_0\) is compression height of sealing ring. \(d_0\) is section diameter of sealing ring in natural state. According to the working condition of radial sealing, the pre-compression ratio is generally 10%–25%.
When the pre-compression ratio is large, the contact stress of the O-ring is larger and the sealing performance of FSCSS is better. But the wear will be more serious. When the pre-compression ratio is small, the contact stress is small and the sealing performance of FSCSS is relatively poor. But the wear is less. In this paper, the effect of pre-compression ratio at 10%, 14%, 18%, 22% on the sealing performance is studied.

Figures 7 to 14 show the maximum Von Mises of FTS-ring and O-ring with different pre-compression ratios at 20°C and 150°C. It can be seen from Figures 5 to 12 that:

1. The contact stress of O-ring at 150°C is smaller than that at 20°C. Under different temperatures and pre-compression ratios, the stress of O-ring is mostly concentrated on the left and right sides, and two stress concentrations are formed. The two stress concentrations forms the sealing surface after pre-compression of O-ring.
Figure 8. Von Mises (compression ratio of 10%, 20°C).

Figure 9. Von Mises (compression ratio of 14%, 150°C).

Figure 10. Von Mises (compression ratio of 14%, 20°C).
Figure 11. Von Mises (compression ratio of 18%, 150°C).

Figure 12. Von Mises (compression ratio of 18%, 20°C).

Figure 13. Von Mises (compression ratio of 22%, 150°C).
(2) The maximum Von Mises of O-ring at 150°C is smaller than that at 20°C, and the difference is about 1 MPa. That is to say that the temperature has a great influence on the maximum Von Mises of O-ring. It shows that the damage effect of stress on O-ring is reduced at 150°C, which is conducive to the extension of O-ring life in FSCSS.

Figures 15 to 22 show the contact stress of FTS-ring and O-ring with different pre-compression ratio at 20°C and 150°C. It can be seen from Figures 15 to 20 that:

1. When the O-ring is pre-compressed, there is a closed contact surface in FSCSS. It can ensure the sealing performance in the early stage of FSCSS under the pressure of the working medium. With the increase of the pre-compression ratio, the contact surface become larger, and the contact stress increases. The sealing performance will be better.
2. The contact stress at 150°C is larger than that at 20°C. So, the sealing performance at high temperature of FSCSS is better than that at room temperature.

Figures 23 and 24 show the maximum Von Mises and maximum contact stress variation curves under different compression ratios of the O-ring. It can be seen from Figures 23 and 24 that:

It can be seen from Figure 26 that contact surface I, contact surface II, contact surface III, and contact surface IV interact to form contact stress. Next, the four contact surfaces are analyzed respectively.

1. The maximum Von Mises and the maximum contact stress of FSCSS increase with the increase of the pre-compression ratio, which shows a positive correlation.
Figure 15. Contact stress (of 10%, 150°C).

Figure 16. Contact stress (10%, 20°C).

Figure 17. Contact stress (14%, 150°C).
**Figure 18.** Contact stress (18%, 20°C).

**Figure 19.** Contact stress (18%, 150°C).

**Figure 20.** Contact stress (18%, 20°C).
When the pre-compression ratio is larger than 14%, the maximum Von Mises of FSCSS increases, and the damage effect of the combined seal on the O-ring seal increases rapidly. However, the contact stress increases slowly.

On the basis, the O-ring with pre-compression ratio of 14% is selected as the research object to analyze the sealing performance of FSCSS with HTHP coupling method.

**Discussion**

*Static sealing performance*

Because there is a hydraulic balance mechanism, the differential pressure between inside and outside the telescopic type of downhole robot is not larger than 20 MPa.
Besides, the temperature is about 150°C. So, when the temperature is 150°C, the influence of different pressure (0, 15, 20, 25, 30 MPa) on the sealing performance is analyzed.

Figures 26 to 29 show the distribution of contact stress in each contact surface under different pressure. Figure 26 shows the distribution of the contact stress of contact surface I. It can be seen from the figure that:

![Figure 23.](image1)  
**Figure 23.** Variation of Maximum Von Mises with pre-compression ratio of O-ring.

![Figure 24.](image2)  
**Figure 24.** Variation of Contact stress with pre-compression ratio of O-ring.
(1) The length of the contact surface I is about 5 mm. With the increase of pressure, the contact stress increases.

(2) It can be seen from the figure that all the curves have a trough. The reason for the trough may be that the contact stress is small, resulting in the decrease of the contact stress.

(3) The distribution of the contact stress between 1 and 4 mm is stable, which indicates that the position between 1 and 4 mm is the sealing area. What’s more, the maximum contact stress is larger than the pressure. It is proved that the sealing performance between FTS-ring and piston is good.

Figure 27 shows the distribution of the contact stress of contact surface II. It can be seen from the figure that:
Figure 27. Distribution of contact stress of contact surface II.

1. The length of the contact surface II is about 6.5 mm. The lower position of the O-ring fluctuates more than the upper position. This is because the FTS-ring has a fillet at the lower part of the O-ring, which makes the deformation of the lower O-ring larger than that of the upper part, and further leads to the fluctuation of the contact stress distribution. Therefore, when the O-ring is pre-compressed, the fillet position of the FTS-ring should be closely fitted with the piston guide rod to avoid excessive deformation of the O-ring.

2. The distribution of contact stress between O-ring and piston guide rod is stable and fluctuates little. The maximum contact stress is larger than the pressure. It shows that the sealing performance between O-ring and piston guide rod is good and there is not a leakage problem.

Figure 28 shows the distribution of the contact stress of contact surface III. It can be seen from the figure that:

1. The length of the contact surface III is smaller than 3 mm. The distribution of contact stress between FTS-ring and piston guide rod decreases gradually from right to left. This is because the interaction between O-ring and FTS ring decreases with the change of contact position.

2. The contact stress of contact surface III is larger than the pressure, which indicates that the sealing performance between FTS-ring and piston guide rod is good.

Figure 29 shows the distribution of the contact stress of contact surface IV. It can be seen from the figure that:

1. The length of the contact surface III is smaller than 5.5 mm. The distribution of the contact stress increases gradually with the increase of the pressure. It can be seen that the deformation of O-ring is gradually increasing,
and the contact area at the upper left position has gradually decreased with the deformation of O-ring.

(2) There are two peaks in the contact stress, which correspond to the two sides of the contact between FTS-ring and O-ring, respectively. A trough corresponds to the fillet of the FTS-ring in contact with the O-ring. It can also be seen from Figures 15 to 22 that the fillet position is the last contact part of O-ring and foot type slip ring, so the contact stress is the minimum. The contact stress of the trough is greater than the pressure, which indicates that the sealing effect between O-ring and FTS-ring is good.

![Figure 28. Distribution of contact stress of contact surface III.](image)

![Figure 29. Distribution of contact stress of contact surface IV.](image)
From the above analysis, it can be seen that under the conditions of pre-compression ratio of 14%, high temperature of 150°C and high pressure of 10–30 MPa, FSCSS can keep good static sealing performance, thus ensuring the operation reliability of the telescopic type of downhole robot.

**Dynamic sealing performance**

Assuming that the penetration rate of the telescopic type of downhole robot is 30 m/h, the movement speed of the piston of the downhole robot is about 8.3 mm/s. The up stroke is the extension state of the telescopic hydraulic cylinder, and the down stroke is the retraction state of the telescopic hydraulic cylinder, as shown in Figure 1.

Figures 30 to 39 show the contact stress of FSCSS under different pressure when the piston of the hydraulic system reciprocates.
Figure 32. Change of contact stress in up stroke at 15 MPa.

Figure 33. Change of contact stress in down stroke at 15 MPa.

Figure 34. Change of contact stress in up stroke at 20 MPa.
Figure 35. Change of contact stress in down stroke at 20 MPa.

Figure 36. Change of contact stress in up stroke at 25 MPa.

Figure 37. Change of contact stress in down stroke at 25 MPa.
(1) FSCSS forms a closed contact surface because of the pre-compression O-ring, which ensures the sealing performance of FSCSS when the pressure is low. With the increase of the pre-compression ratio, the contact stress also increases, which can ensure the sealing performance.

(2) At high temperature, the contact stress is greater than that at the normal temperature, which is caused by the expansion and contraction of the sealing materials. It also shows that the sealing performance of FSCSS at high temperature is better than that at normal temperature.

Figures 40 to 43 show the maximum contact stress of the O-ring when the piston moves reciprocally.

(1) It can be seen from Figures 40 and 41 that the maximum contact stress of the contact surface I, contact surface III varies greatly when the piston
reciprocally moves. The maximum contact stress between FTS-ring and piston in up strokes is larger than that in down strokes. The maximum contact stress between FTS-ring and piston guide rod in up strokes is smaller than in down strokes. The reason is that when the piston is in up strokes, the FTS-ring is subjected to upward friction. The maximum contact stress between FTS-ring and piston increases. However, the maximum contact stress between FTS-ring and piston guide rod decreases. When the piston is in down strokes, the maximum contact stress between FTS-ring and piston decreases. Also, the maximum contact stress between FTS-ring
and the piston guide rod increases. It can be seen from the figure that the maximum contact stress is larger than the pressure when the piston reciprocates, which indicates that there is not a leakage phenomenon. The sealing performance is good.

(2) It can be seen from Figures 42 and 43 that the maximum contact stress of contact surface II and contact surface IV change little when the piston reciprocates. It indicates that the maximum contact stress of the O-ring is little affected by the reciprocating motion of the piston. What’s more, the maximum contact stress on O-ring is larger than the pressure. Therefore, there

Figure 42. Maximum contact stress of contact surface III.

Figure 43. Maximum contact stress of contact surface IV.
is not leakage phenomenon on the contact surface. The sealing performance is good.

In order to explore the influence of the velocity of the combined sealing structure of telescopic type of downhole on sealing performance, the variation of maximum contact stress of O-ring with velocity under 20 MPa, 150°C and pre-compression ratio of 14% was analyzed. It can be seen from Figure 44 that the maximum contact stress drops sharply from 0 to 2.5 mm/s, and the static seal pressure is far greater than the dynamic seal under the same working conditions. The seal pressure of dynamic seal decreases with the increase of velocity, but the change range is small.

From the above analysis, it can be seen that FSCSS can keep good dynamic sealing performance when the piston moves at the reciprocating speed of 8.3 mm/s under the condition of pre-compression ratio of 14%, high temperature of 150°C, and high pressure of 10–30 MPa.

Conclusion

Considering the requirements of dust prevention and sealing of the telescopic type of downhole robot, FSCSS is selected. In order to make the simulation data more accurate, HTHP coupling numerical simulation model of piston, piston sleeve, O-ring, and FTS-ring is established, which introduces the real data of shape, stress, and strain of O-ring caused by pre-compression caused by assembly. It is found that the sealing performance is very excellent when the compression ratio of the O-ring is about 14%. What’s more, according to the limit motion law of the downhole robot, the distribution of contact stress, and Von Mises of the O-ring at 8.3 mm/s of motion speed are analyzed. The results show that FSCSS can keep a good dynamic sealing performance under HTHP condition. At the same time, it is found that under the same pressure, the sealing performance of the telescopic type
of the downhole robot under high temperature is better than that under low temperature. This is due to the expansion and contraction of the material. This paper provides a theoretical basis for the analysis of the dynamic sealing performance by using HTHP coupling method. What’s more, it also provides a reference for the dynamic sealing structure design of downhole tools under HTHP condition.

Author contributions
J.G. Zhao put forward the idea of numerical simulation research on the sealing performance of drilling tools under the condition of HTHP coupling. H.X. Peng, S. Han, S.J. Fang, and K.P. Wang calculate the numerical model based on the drilling condition. S. Han and Y. Zhang analyze the sealing performance. Z.X. Zhu and C. Tu are responsible for literature research and text collation.

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References
1. Roehrlich M. Coiled tubing drilling - a concept for directional drilling in pressure depleted reservoirs. Oil Gas Eur Mag 2016; 42(3): 121–130.
2. Liu Q. Smart drilling system in future. CAI Trans Intell Syst 2009; 4(1): 16–20.
3. Liu QY, Dong R, Geng K, et al. The status of current research on downhole robots and their multiple applications. Pet Drill Tech 2019; 47(3): 50–55.
4. Bybee K. New downhole tool extends coiled-tubing reach. J Pet Technol 2015; 52(6): 32–34.
5. Livescu S and Craig S. A critical review of the coiled tubing friction-reducing technologies in extended-reach wells. Part 1: Lubricants. J Pet Sci Eng 2011; 157: 747–759.
6. Hallundbæk J. The story of welltec, http://www.welltec.com/company/(1995-1-3) (accessed 11 December 2020).
7. Au G, Sheirtov T, Elstrop S, et al. Next-generation release device: strong, safer, efficient, and rigorously qualified. In: SPE/CoTA coiled tubing and well intervention conference and exhibition, 27 March 2018. Texas, USA: Society of Petroleum Engineers.
8. Li DX, Zhao HL, Zhang SM, et al. Finite element analysis on the influence of back-up ring on the sealing performance of rubber O-ring. *Adv Mater Res* 2011; 199–200: 1595–1599.

9. Yamabe J, Koga A and Nishimura S. Failure behavior of rubber O-ring under cyclic exposure to high-pressure hydrogen gas. *Eng Fail Anal* 2013; 35: 193–205.

10. Zhou CL, Zheng IY, Gu CH, et al. Sealing performance analysis of rubber O-ring in high-pressure gaseous hydrogen based on finite element method. *Int J Hydrogen Energy* 2017; 42(16): 11996–12004.

11. Kömmling A, Jaunich M, Pourmand P, et al. Influence of ageing on sealability of elastomeric O-rings. *Macromol Symp* 2017; 373(1): 1600157.

12. Troufflard J, Laurent H, Rio G, et al. Temperature-dependent modelling of a HNBR O-ring seal above and below the glass transition temperature. *Mater Des* 2018; 156: 1–15.

13. Huang HC, Ye YY, Yang CJ, et al. Study of the sealing characteristic of polytetrafluoroethylene-coated O-ring applied in gas-tight deep-sea water sampler. *Adv Mater Res* 2011; 295–297: 3–10.

14. Zhang J and Xie J. Investigation of static and dynamic seal performances of a rubber O-ring. *J Tribol* 2018; 140(4): 1–46.

15. Tadayoshi S and Koji F. Analytical prediction of dynamic properties of O-ring with hydrostatic pressure distribution. *J Appl Mech* 2018; 12(85): 121001.

16. Mooney M. A theory of large elastic deformation. *J Appl Phys* 1940; 9(11): 582–592.

17. Liu QY, Yang YQ and Zhu HY. Sealing performance of C-sliding ring combined seals. *Lubr Eng* 2017; 42(8): 36–41.

18. Che HD. Che’s seal manual, https://max.book118.com/html/2018/1023/5113233013001323.shtm (2018-10-23) (accessed 11 December 2020).

19. Yu JY, Gao JY, Lin W, et al. Experimental study and finite element analysis of the performance of reciprocating seal in rapping device of gasifier. *Appl Mech Mater* 2011; 42: 48–53.

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Hanxiu Peng is a senior engineer and mainly engaged in the research of intelligent completion and complex structure well completion. In 1999, he graduated from China University of Petroleum (East China) in mechanical manufacturing. In 2004, he obtained a master’s degree in oil and gas storage and transportation from China University of Petroleum (East China). He published 3 papers and authorized 4 patents.

Shiji Fang is studying for a bachelor’s degree in mechanical engineering from Southwest Petroleum University. From July 2019 to Sep 219, he participated in production practice in Fanyu Li Neng Co., Ltd in Mianzhu. During the undergraduate time, he has passed the CET6. From 2019 to 2020, he took part in the 7th National College Students’ engineering
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