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TECHNICAL ARTICLE—PEER-REVIEWED

Static and Dynamic Sealing Performance Analysis of Rubber D-Ring Based on FEM

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Abstract In order to study the performance of the dynamic seal and static seal of D-ring, the FEM models of D-ring and O-ring were built, and the process of reciprocating seal of the D-ring was simulated. In addition, the effects of pre-compression, medium pressure, friction coefficient, and rubber hardness on sealing performance were discussed. The friction coefficients between rubber sealing ring and steel part, under different lubrication conditions, were measured through the experiment. The results show that, in static seal, the maximum contact stress of the D-ring increases with the increasing of pre-compression, friction coefficient, medium pressure, and rubber hardness. Besides, O-ring can be well replaced by D-ring in static seal. Since it can avoid distortion and twisting effectively in reciprocating dynamic seal and prolong its working life, D-ring has a better sealing performance in reciprocating seal than O-ring. As the friction coefficient grows, the amplitude of the maximum contact stress fluctuation increases as well as the fluctuation of the cross-section deformation which has a bad influence on sealing performance and fatigue life of the D-ring.

Keywords D-ring · Static seal · Reciprocating dynamic seal · Contact stress · Numerical simulation

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Introduction

For its great self-sealing performance and low cost, rubber sealing ring, especially O-ring, has been widely used in many fields, such as oil industry, food processing machinery, aviation, and aerospace. However, O-ring is also easily prone to distort and twist in reciprocating dynamic seal, especially when the friction in the circumferential direction of the O-ring is nonuniform and the coaxiality of groove is poor. Distortion and twisting make the sealing ring's working life shortened greatly. Compared with O-ring, D-ring, structure of which is based on O-ring, is more applicable for reciprocating dynamic seal. For its special structure, D-ring does not likely twist or distort in reciprocating dynamic seal. In the meantime, D-ring can achieve good self-sealing performance because of the semicircle structure of the inside of the D-ring which is an advantage inherited from O-ring.

Many researchers have studied the sealing performance of O-ring. For example, in 1990, Karaszkiewic had already studied on geometric distortion and the contact forces of O-ring [1]. Wang researched on the stress distribution of O-ring during the installation process and working process by computer simulation [2]. Tribological properties and seal abilities of sealing compensating mechanism were analyzed by Li [3]. Besides, Cui used numerical calculation to study the leakage and friction of O-ring [4]. Han and Zhang designed a new structure of sealing ring to reduce the failure probability [5]. But, so far, D-ring hardly shows in literatures. Therefore, finite element models of D-ring and O-ring were built, and the sealing performances of D-ring and O-ring in static seal and reciprocating dynamic seal were researched. In addition, the effects of pre-compression, medium pressure, and rubber hardness on sealing performance of D-ring were discussed.

Constitutive Model

Material Model of Rubber

Rubber can be modeled as a kind of incompressible hyperelastic material which deforms under the action of external force. Currently, a variety of constitutive models are used to describe nonlinear materials of rubber, such as Exponential–Hyperbolic rule, Mooney–Rivlin model, Klosenr–Segal model, and Ogden–Tschoegl model. In this paper, the Mooney–Rivlin model was selected to describe the mechanical characteristics of rubber linings. The function can be expressed as follows:

$$W = C_1(I_1 - 3) + C_2(I_2 - 3), \quad \text{(Eq 1)}$$

where W is the strain energy density, C_1, C_2 are Mooney–Rivlin coefficient, I_1, I_2 are the first and second strain tensor invariant.

The relationship of stress and strain can be expressed as follows:

$$\sigma = \partial W / \partial \varepsilon. \quad \text{(Eq 2)}$$

According to the rubber compression test, $C_1 = 1.87$, $C_2 = 0.47$ [6].

Experiment of Friction Coefficient

In order to measure the friction coefficient between rubber and steel part, MMW-1 (Jinan Caide Instrument Co., Ltd) was selected as friction testing machine. Test-pieces of rubber sealing ring were fixed on the steel plate by the process of vulcanization. The tensile strength of rubber is not less than 16 MPa, tensile elongation is not less than 200%, volume change rate is less than 15%, and the hardness is 80HA. Carbon steel, of which the yield strength is not less than 552 MPa and the tensile strength is not less than 689 MPa, was chosen as experimental material of steel sample. The steel sample was made into a cylindrical shape. The friction coefficients between rubber and steel under different lubricating conditions, such as no lubricant condition, water lubrication condition, oil lubrication condition, were examined with axial compressive force ($F = 20, 30, 40$ N) loaded on.

The average friction coefficients between rubber and steel under different friction coefficients were shown in Fig. 1. The friction coefficient changes with lubricating conditions. The minimum friction coefficient appears under oil-base lubricant condition while the maximum friction coefficient appears under no lubricant condition. Besides, axial compressive force has a small effect on friction coefficient between rubber and steel. In this paper, default value for the friction coefficient between rubber sealing ring and steel part was set as 0.3.

Computation Model

Geometric Model

D-ring, O-ring, groove and slide bar were established by advanced finite element program based on the actual structure of the sealing system. As per related standard, the cross-section diameters of O-ring and the section length of D-ring are both 5.33 mm. The inner diameters of the O-ring and D-ring are both 45.20 mm. The material of groove and slide bar is medium carbon-hardened tempered steel whose density is 7800 kg/m^3, Poisson's ratio is 0.3, and modulus of elasticity is 210 GPa. Contact problem between sealing ring and slide bar is a functional extremum problem with constraints when it meets no penetration constraints condition. A contact penalty algorithm was employed to simulate the interactions between the ring and steel material. Two contact pairs, between rubber sealing and groove, and also between rubber sealing ring and slide bar, were established. As shown in Fig. 2, four-node quadrilateral bilinear axisymmetric elements (CAX4R) were used for meshing the whole finite element model.

Boundary Conditions and Load

In practice, rubber sealing ring has to face two different kinds of working conditions—static seal and dynamic seal. Static sealing process can be simulated by two steps:

Step 1. In order to simulate the installation process of sealing ring, pre-compression (0.3 mm) is loaded on.

Step 2. Medium pressure ($P = 3$ MPa) is loaded on the working surface of sealing ring to simulate the working condition.

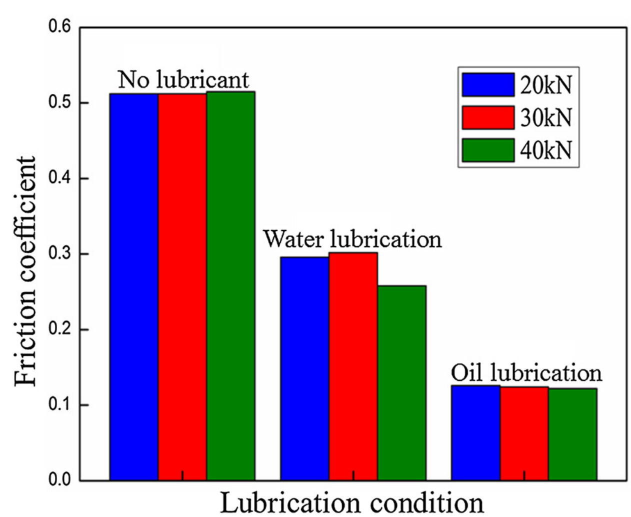
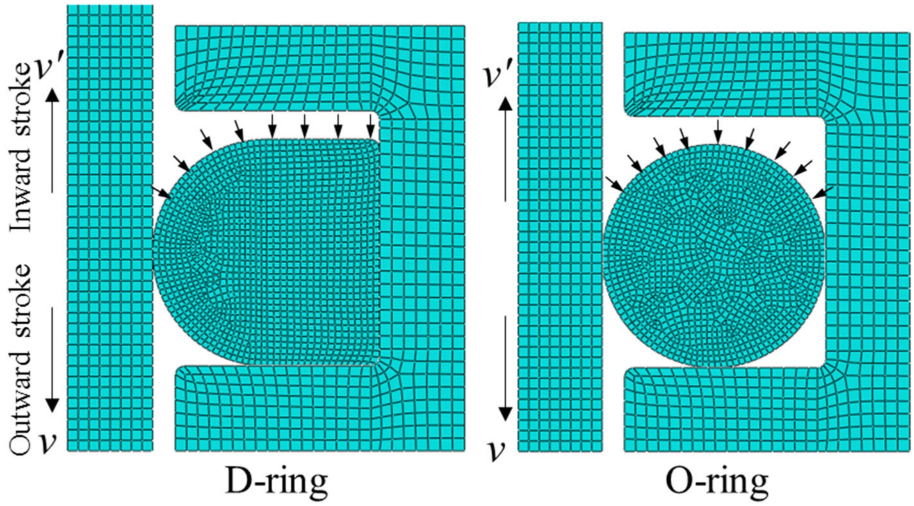


Fig. 1 Experiment results of friction coefficient

Fig. 2 Finite element models



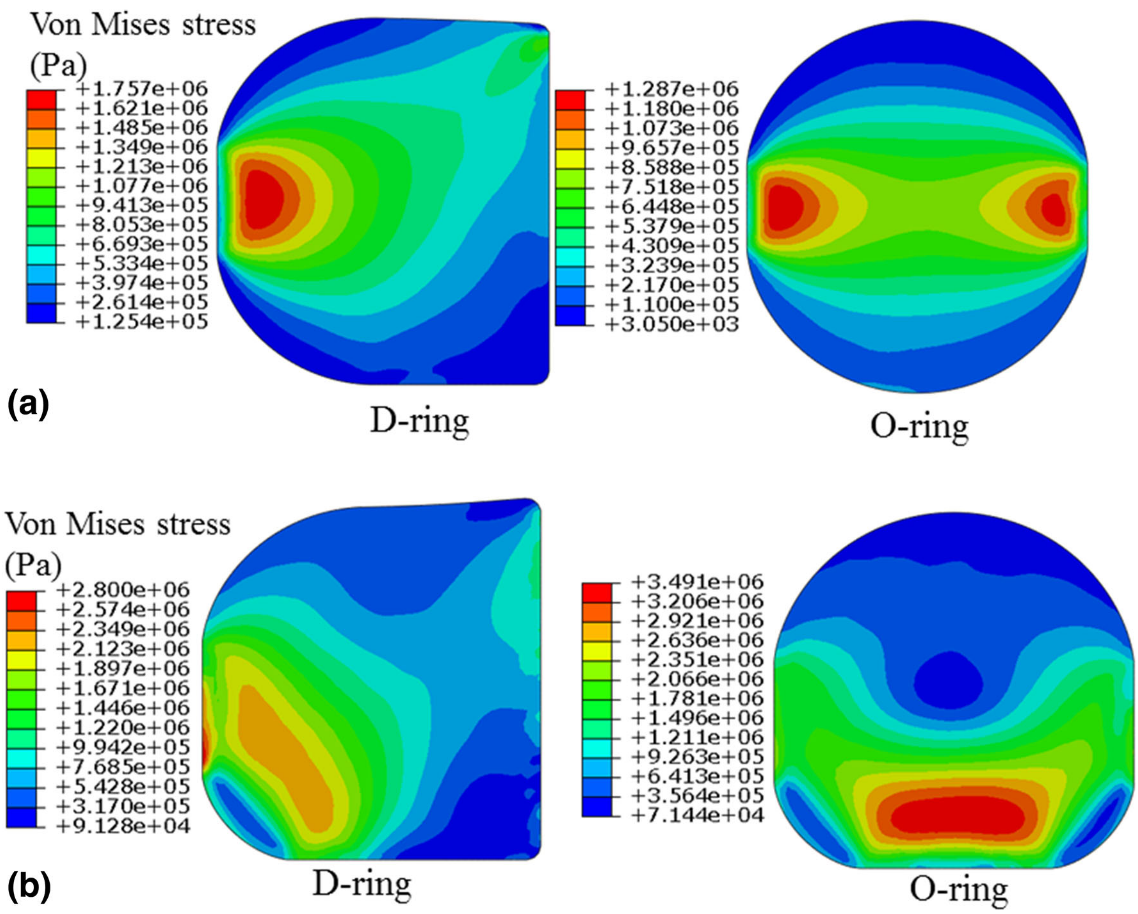


Fig. 3 (**a**) Von Mises stress distribution of sealing ring (no-pressure condition). (**b**) Von Mises stress distribution of sealing ring (medium pressure $P = 3$ MPa)

To achieve the reciprocating dynamic sealing process, the third step has to be added. *Step 3*. Axial velocity is applied at the slide bar. Inward stroke is defined as the movement of the slide bar toward the same direction of the medium pressure. On the opposite, outward stroke is defined as the movement of the slide bar against the medium pressure. In order to meet the requirements of sealing, contact stress has to be larger than medium pressure.

$$\delta_{max} \geq P, \qquad \text{(Eq 3)}$$

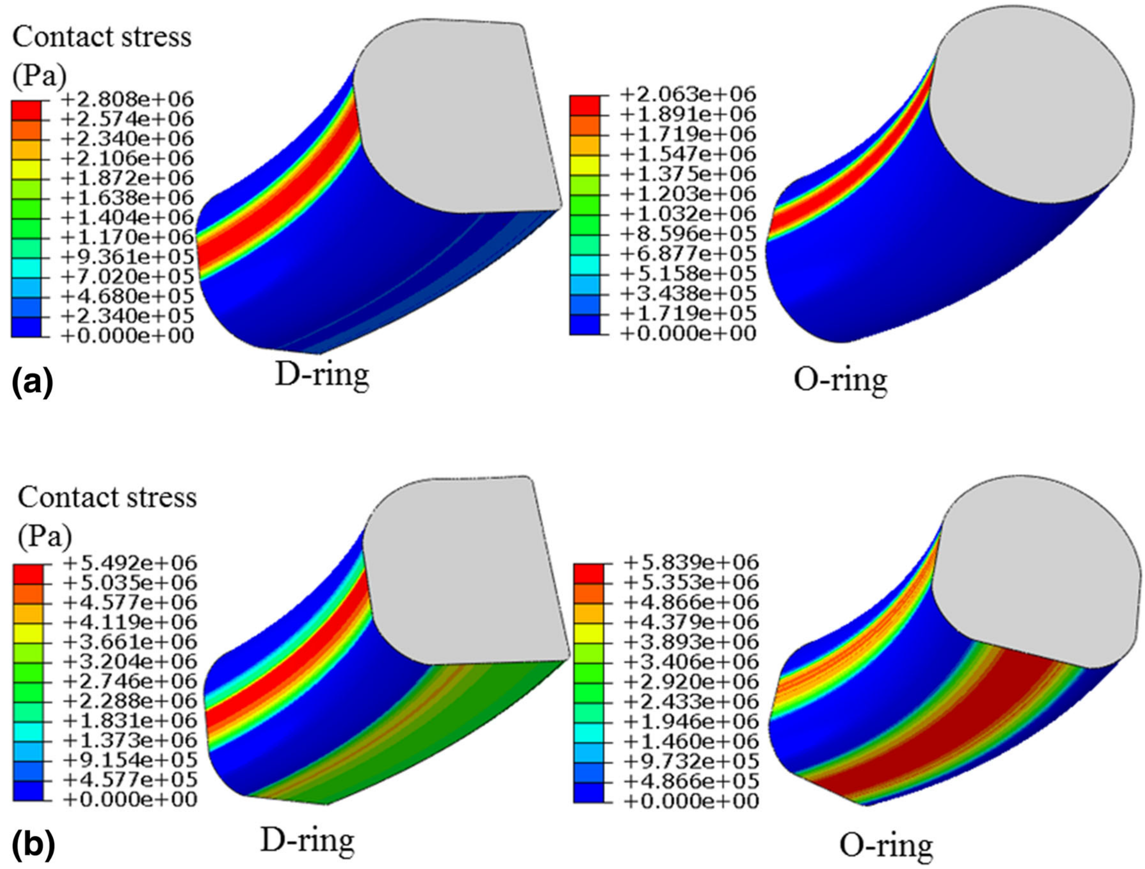


Fig. 4 (**a**) Contact stress distribution of sealing ring (no-pressure condition). (**b**) Contact stress distribution of sealing ring (medium pressure $P = 3$ MPa)

where $\delta_{\max}$ is the maximum contact stress of the main sealing surface. P is the medium pressure [7–9].

Fundamental Assumption

It is essential for research on rubber sealing ring to make some assumptions for rubber as the material nonlinearity, geometrical nonlinearity, and contact nonlinearity. In order to study the mechanical property and sealing performance of sealing ring, some assumptions were made as follows.

(1) Rubber material used by sealing ring has a determined Poisson's ratio and elasticity modulus.
(2) Medium will not make corrosion to rubber sealing ring.
(3) Rubber sealing ring would not be affected by medium temperature.
(4) Volume of sealing ring does not change with creep deformation.
(5) Compression and tension of rubber material have the same creeping property.

Analysis of Static Sealing Performance

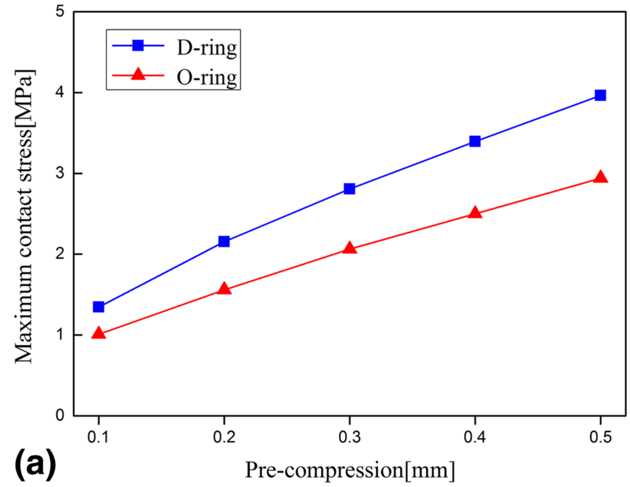
Stress Analysis

When pre-compression is 0.3 mm, von Mises stress distributions of D-ring and O-ring are shown in Fig. 3. As shown in Fig. 3a , high stress area appears on the left end of the D-ring (in the semicircular convex) when D-ring is under no-pressure working condition and the maximum von Mises stress of the D-ring is 1.757 MPa which appears inside the sealing ring instead of on the surface of the sealing ring. Meanwhile, von Mises stress of the rectangular part of the D-ring is relatively small. Von Mises stress distribution of the D-ring is not a symmetrical distribution and the high stress area is more concentrated on the top inside the D-ring. High stress area of O-ring is distributed into "dumbbell shape" and more than one high stress area appears in the O-ring. The maximum von Mises stress of the O-ring is 1.287 MPa, 26.7% smaller than D-ring's. When medium pressure is 3 MPa, von Mises stress distribution of O-ring is shown in Fig. 3b. There is a big difference between von Mises stress of the sealing ring under no-pressure working condition and under pressured working condition. The maximum von Mises stress of the D-ring appears on the surface which contacted with slide bar, but high stress area still appears in the left part of the D-ring. Under the action of medium pressure, high stress area of O-ring gradually transferred from the center to the underside and it is distributed in rectangle shape. However, high stress area in the center of the O-ring is replaced by low stress area. Under pressured working condition, the maximum von Mises stress of O-ring is 3.491 MPa while of D-ring it is 19.7% lower (about 2.8 MPa). Compared with D-ring, O-ring is more likely to fail.

Figure 4 shows the contact stress distributions of the D-ring and the O-ring. While the sealing rings are under no-pressure condition, the maximum contact stress appears at the main sealing surface which contacts slide bar. And the contact width of D-ring is larger than of O-ring (as seen in Fig. 4a). The maximum contact stress of D-ring is 2.8 MPa while that of O-ring is 2 MPa. When medium pressure is 3 MPa, the contact stress distributions of these two kinds of sealing rings are shown in Fig. 4b. Under the action of medium pressure, the rubber sealing rings come in contact with slide bar and groove to achieve self-sealing. The contact stress of the main surface of the D-ring is the maximum while it is relatively small on the other side. Both of the main surface and the other side, the maximum contact stresses are larger than 3 MPa which means D-ring can achieve good self-sealing. As for O-ring, it can achieve good sealing despite the maximum contact stress appearing at the other side instead of the main surface. Since the maximum contact stresses of these two kinds of rubber sealing rings have no great difference, O-ring could be replaced well by D-ring in static sealing. In addition, when considering the function of the main sealing surface, D-ring is also much more reliable than O-ring.

Effect of Pre-compression

Appropriate pre-compression can maintain stable and reliable self-sealing of rubber sealing ring. Figure 5a shows the curves of the maximum contact stress of the main sealing surface of D-ring and O-ring under different pre-compressions when medium pressure $P = 0$ MPa. The maximum contact stresses of the main sealing surfaces, of both the D-ring and the O-ring, increase with the increasing of the pre-compression. Besides, the contact stress of D-ring is always higher than it of O-ring. The maximum contact stress-compression curves of the main sealing surfaces of D-ring and O-ring under different pre-compressions when medium pressure $P = 3$ MPa are shown in Fig. 5b. The maximum contact stress of both O-ring and D-ring is larger than the medium pressure $P = 3$ MPa which



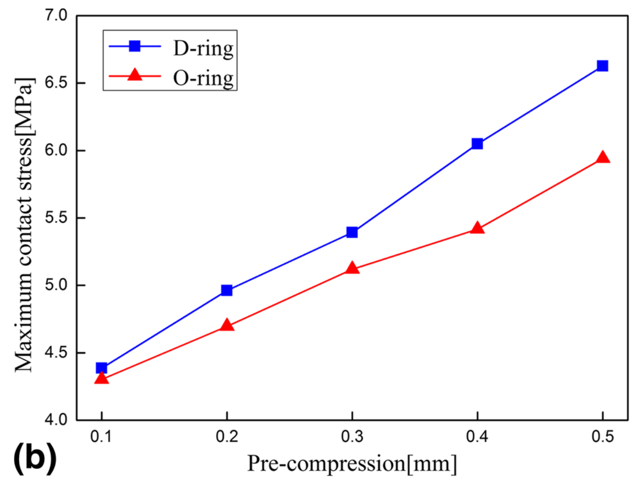


Fig. 5 (**a**) Contact stress of sealing ring under different pre-compressions (no-pressure condition). (**b**) Contact stress of sealing ring under different pre-compressions (medium pressure $P = 3$ MPa)

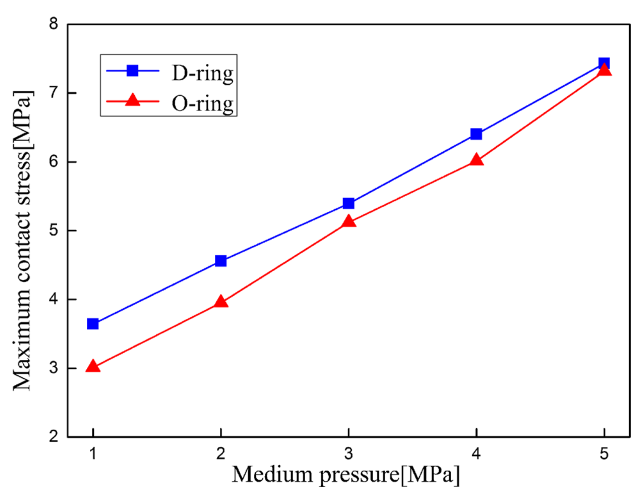


Fig. 6 Contact stress of sealing ring under different pressures

means they can well achieve the expected sealing performance. Under the condition where medium pressure exists, the maximum contact stress of both D-ring and O-ring increases with the increasing of pre-compression. However, compared with O-ring, D-ring has a higher growth rate of contact stress with the change of pre-compression. It ultimately causes the difference value of contact stress between D-ring and O-ring to get more and more giant as pre-compression increases.

Effect of Medium Pressure

When pre-compression is set as 0.3 mm, the maximum contact stress curves, under different medium pressures, of the main sealing surface of both D-ring and O-ring are shown in Fig. 6. With the increasing of medium pressure, the maximum contact stress of both rubber sealing rings, which is also larger than its corresponding medium pressure, increases gradually. Hence, well-performing sealing can be ensured. The contact stress difference value between D-ring and O-ring decreases as medium pressure increases. D-ring's static sealing performance excelled O-ring, on account of having larger maximum contact stress.

Effect of Rubber Material

The Shore hardness of rubber sealing rings which are generally applied in multiple fields is from 70 to 90 HA. Liu derived physical parameters (C_1 and C_2) which are well consistent with engineering practice of rubber materials under different hardness.

The curves of the maximum contact stress of the D-ring's main sealing surface and the O-ring's with different rubber hardness are shown in Fig. 7a, when pre-compression is 0.3 mm and medium pressure does not exist. With the increasing of rubber hardness, main sealing surfaces' maximum contact stresses of both O-ring and D-ring increase nonlinearly. In addition, when medium pressure $P = 3$ MPa, the maximum contact stresses of these two kinds of sealing rings increase nonlinearly as well (as shown in Fig. 7b). No matter under no-pressure condition or pressured condition, the maximum contact stress of D-ring is larger than O-ring with corresponding rubber hardness which ensures D-ring has a better and stable sealing performance.

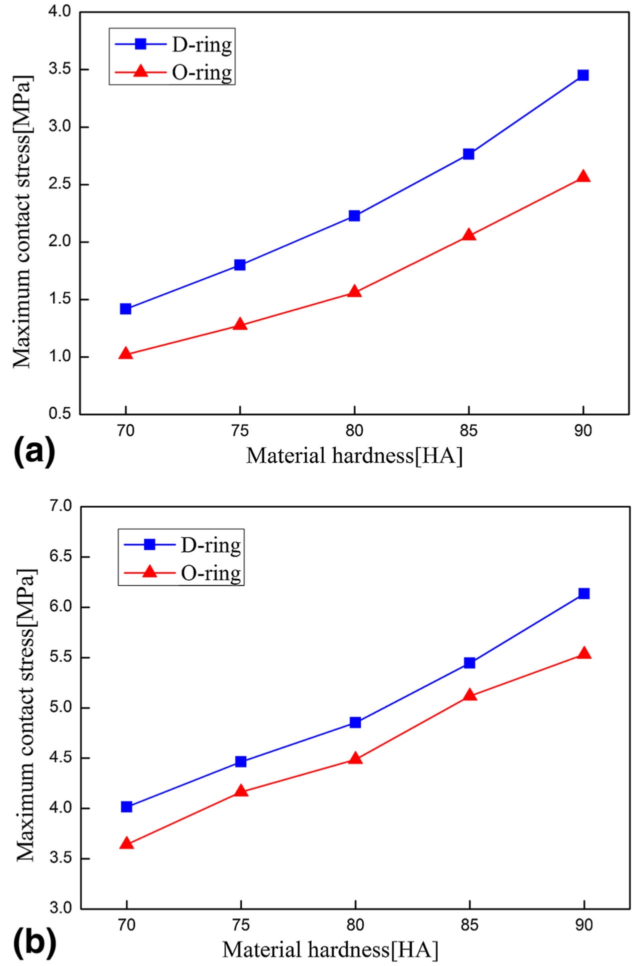


Fig. 7 (**a**) Contact stress of sealing ring under different rubber hardness (no-pressure condition). (**b**) Contact stress of sealing ring under different rubber hardness (medium pressure $P = 3$ MPa)

Analysis of Reciprocating Dynamic Sealing Performance

In addition to being used in static seal, rubber sealing rings are also wildly applied in reciprocating dynamic seal. In order to compare dynamic sealing performance of the two types of sealing rings, the whole process of reciprocating dynamic seal was simulated. The maximum contact stresses of D-ring and O-ring in reciprocating dynamic seal are shown in Fig. 8, when axial velocity $v = 0.2$ m/s, pre-compression is 3 mm, and medium pressure $P = 3$ MPa. For its own viscoelasticity, rubber sealing rings' maximum contact stresses fluctuate wildly when reciprocating motion begins. By contrast, stress fluctuation of D-ring is much more wide at this stage of the reciprocating motion. But when reciprocating motion reaches the stable stage in outward or inward stroke, the maximum contact stress of D-ring is always larger than O-ring's. Besides, contact stress fluctuation of D-ring is relatively small and sealing performance is more stable which makes D-ring meet the sealing requirements satisfactorily. Since the semicircular structure of the D-ring's inner side inherits good self-sealing function from O-ring and the structure of D-ring's outer side takes advantage of rectangular shape, D-ring can effectively avoid the damages of distortion and twisting, which in addition, prolongs its working life.

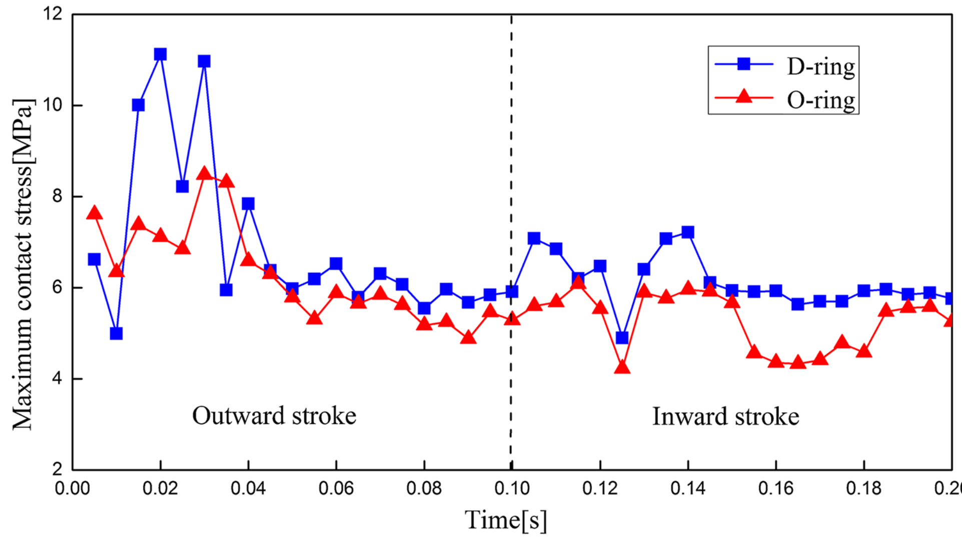


Fig. 8 Contact stress of sealing ring in reciprocating motion

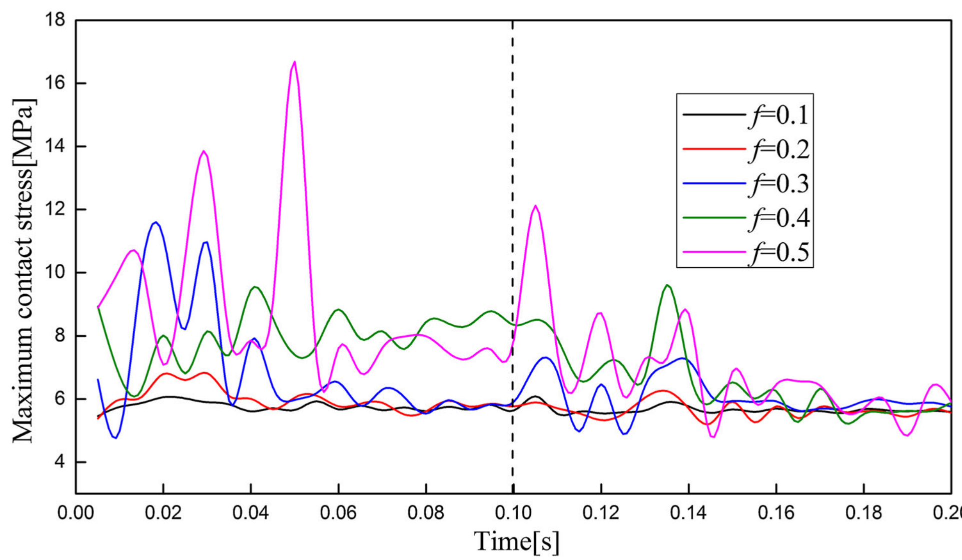


Fig. 9 Contact stress of sealing ring under different friction coefficients

Friction coefficient between rubber sealing ring and steel part plays an important role in reciprocating dynamic seal for it has a significant impact on sealing performance, operation stability, and frictional wear. According to the experimental test, friction coefficients under oil lubricating condition, water lubricating condition, and no lubricant condition are around 0.1, 0.3, and 0.5, respectively. Therefore, five working conditions under different friction coefficients (0.1, 0.2, 0.3, 0.4, 0.5) were analyzed. When reciprocating velocity $v = 0.2$ m/s, the maximum contact stress-time curves of D-ring under different friction coefficients are shown in Fig. 9. At stable stage of outward stroke, the maximum contact stress of D-ring increases with the increasing of friction coefficient. Nevertheless, friction coefficient has a little influence on the maximum contact stress when reciprocating motion at stable stage of inward stroke. During the whole process of reciprocating motion, the amplitude of contact stress fluctuation increases with the increasing of the friction coefficient. The max amplitude of contact stress fluctuation appears at the beginning of outward stroke and inward stroke. When friction coefficient is 0.1, the contact stress variation of D-ring is the smoothest. When friction coefficient is 0.5, contact stress of D-ring fluctuates widly and the difference value between the maximum and the minimum is 11.84 MPa. Besides, creeping phenomenon appears at the beginning of outward stroke which could have a serious effect on sealing performance. Meanwhile, high friction coefficient may not only lead to sealing ring's serious cross section deformation but also greatly reduce working life of sealing ring.

Conclusions

(1) Under no-pressure working condition, high stress area appears in the semicircular convex of the D-ring but high stress area of O-ring is distributed into "dumbbell shape." With the action of medium pressure, the maximum von Mises stress appears in the main sealing surface of the D-ring, while the high stress area of the O-ring gradually transfers from the center to the underside and it is distributed in rectangle shape. O-ring could be well replaced by D-ring in static seal.

(2) In static seal, the maximum contact stress of both D-ring and O-ring increases with the increasing of pre-compression, medium pressure, and hardness of rubber material. And the maximum contact stress of the main surface of the D-ring is always higher than it of O-ring.

(3) In reciprocating dynamic seal, D-ring has a higher maximum contact stress and a smaller contact stress fluctuation than O-ring. D-ring can effectively avoid distortion and twisting, and thereby prolonging its working life. As friction coefficient grows, the amplitude of contact stress fluctuation as well as the fluctuation of the cross-section deformation increases which has a bad influence on the sealing performance, and thereby greatly reducing D-ring's fatigue life.

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References

1. A. Karaszkiewicz, Geometry and contact pressure of an O-ring mounted in a seal groove. Ind. Eng. Chem. Res. **29**, 2134–2137 (1990)
2. W. Wang, S.G. Zhao, Nonlinear finite element analysis of rubber O-sealing ring. Lubr. Eng. **30**(4), 106–110 (2005). **(in Chinese)**
3. S.X. Li, J.N. Cai, Q.X. Zhang, Performance analysis of O-ring used in compensatory configuration of mechanical seal. Tribology **30**(3), 308–314 (2010)
4. X. Cui, L.L. Dong, K.D. Zhao, Caculation of leakage and frication of combined dynamic seals based on ADINA. J. South China Univ. Technol. **38**(2), 95–100 (2010). **(in Chinese)**
5. C.J. Han, H. Zhang, J. Zhang, Structural design and sealing performance analysis of bio-mimetic sealing ring. Appl. Bion. Biomech. **5**, 1–11 (2015)
6. C.J. Han, J. Zhang, Finite element analysis and optimization of rectangle rubber seal. China Rubber Ind. **60**(2), 98–103 (2013). **(in Chinese)**
7. C.J. Han, J. Zhang, G. Huang, Sealing performance analysis of a star sealing ring in reciprocating seal. Lubr. Eng. **37**(9), 28–32 (2012). **(in Chinese)**
8. Q.B. Ren, T.M. Cai, R.Q. Wang, Investigation on structure parameters and failure criteria of "O"-type rubber sealing ring. J. Solid Rocket Technol. **29**(1), 9–14 (2006). **(in Chinese)**
9. J. Liu, X.Q. Chou, W.S. Bo, Numerical analysis on the maximum contact pressure of rubber O-ring. Lubr. Eng. **35**(1), 41–44 (2010). **(in Chinese)**