The Design and Analysis of Roller Cone Bit Bearing Ring

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Abstract. Increasing the sealing performance of bearing rings of roller-cone bits plays an important role in reducing the bearing seal failure and improving the service life of drill bits. This paper conducts (1) the structure design of the sealing rings. (2) a simulation on the new designed structure of roller cone bit. The contact pressure together with the equivalent stress of the new tooth bars was analysed by building up the finite element model of the bionic non-smooth surface structure, beyond that the comparisons between the new structure and O type sealing ring were carried out. The results indicate that the ribbed sealing structure is capable of improving the wear resistance, postponing the wear abrasion and ageing, and increasing the service life of rubber ring.

1. Introduction
A roller cone bit plays an important role in a drilling project, it not only affects the drilling quality, but also has impact on the drilling cost and efficiency[1,2]. After analysing the field collected failed bi-metal seal structure, the researchers discover that the major factor of the failure is the abrasion of rubber ring[3]. The abrasion of rubber ring will result in the failure of bearing and further cause the breakdown of drilling bit, which not only influences the drilling efficiency but also raises the cost. To counter the problems above, abundant of researches have been carried out. In 2010s, some scholars and institution focused on the utilization and improvement of rubber sealing ring. Hughes company improved the original O-shaped rubber sealing ring and provided a new HAR rubber sealing ring which increasing the durability and reliability[4], Y Zhou et al. also proposed the improving methods of metal ring and rubber ring in bi-metal seal, and analysed the rubber ring of bi-metal seal structure by using ANSYS[5,6]. X Xiao et al. carried out an optimization analysis on bi-metal seal structure of high-speed roller bit, and the results shows that the support angle and chamfer size of bi-metal seal structure have an intensive impact on the stress of O-shaped rubber sealing ring[7]. L A Otto et al. put forward a floating metal seal structure which includes metal rotating seal ring, metal stationary seal ring, O-shaped rubber ring and unbalance loading spring[8]. At the aspect of bionic non-smooth surfaces, Gao K designed drilling bits based on dung beetles’ non-smooth heads and developed bionic diamond drilling bits, increasing the life span of drilling bit[9]. Y H Fu et al. developed a mechanical seal which has the laser surface cross-scale textured mould[10]. T Sugihara and T Enomoto machined the rake faces of tools to the concave textured surface and grooved textured surface, and then the concave textured surface is proved to have a better friction and wear properties[11]. All the researches mentioned above have provided a lot of valuable and directive achievements. Whereas, there is hardly any research integrated the bionic non-smooth surface into the sealing structure design of drilling bit. Thus, in order to reduce the wear of
rubber ring and extend the life of a roller cone bit, this paper proposed a novel sealing structure of roller cone bit based on the bionic non-smooth surface theory.

2. The Design of Sealing Structure

2.1. Ontology Design
(1) Inside Diameter
In this paper, 52mm of the inside diameter is the final selection.
(2) Compression Ratio design
The compression ratio $\varepsilon$ is the measurement index of resilience force, which can be expressed by the formula:

$$\varepsilon = \frac{d_0 - h_0}{d_0} \times 100\%$$

Where, $d_0$ is the diameter of cross section when the O type sealing ring is under the free condition (mm); $h_0$ is the height of cross section when the O type sealing ring is under the compression (mm).
In this paper, the compression ratio of the bionic non-smooth surface sealing structure is determined as 10% to ensure the effectiveness and prevent the excessive deformation of the sealing ring.
(3) Cross Section and Outside Diameter
According to the cone dimensions, the hole diameter and shaft diameter of which are 60 mm and 50 mm respectively, the parameter $h_0$ can be calculated as 5mm. Furthermore, the compression ratio of bionic non-smooth surface sealing structure is determined to be 10% as previously mentioned, and then the cross-section diameter $d_0 \approx 5.55$mm can be obtained by reverse derivation of the formula (1). The outside diameter $D$ is gained from the following formula:

$$D = d + 2 \times d_0$$

For the convenience of calculation, the value of $d_0$ is 6mm and the outside diameter $D$ is 64mm then accordingly.

2.2. Tooth Profile Design
The surface of chlamys farreri, a kind of aquatic mollusk is ribbed which is wear-resistant. The millimeter scale ribbed structure of its surface suffers from the wearing of sea sands, so it boasts excellent wear resistance against abrasives. To ensure that the distance between ribs can produce optimal wear resistance, the bottom width and distance of ribs are divided according to angle (°) rather than length (mm), which are the bottom width angle and the angle of distance, as shown in Figure 1.

![Figure 1. Bottom Width Angle and Angle of Distance](image)

The fillet $R = 1$ mm is adopted to simplify the model. On the ring, the bottom width angle and the angle of distance are evenly distributed at the ratio of 1.5.
We choose the bottom width angle as $12^\circ$ with corresponding number of ribs being 12. The 3-dimensional model of ribbed toothed bars built in Pro/Engineer in Figure 2.

![Figure 2. The Design of Ribbed Toothed Bars](image)
3. Establishing of the Simulation Model

3.1. Geometric Model Construction
The Pro/Engineer software is employed to construct the geometric model for bionic non-smooth surface sealing ring. The simplified Pro/Engineer models for O type sealing ring and ribbed sealing ring are demonstrated in Figure 3.

![Simulation Model of O Type Sealing Ring](image1)
![Simulation Model of Ribbed Sealing Ring](image2)

Figure 3. Simplified Simulation Model

3.2. Definition of Material Attribute
The material attribute should be defined here includes two types, one is the metal material used in the roller bearing and shaft diameter, and another is the bionic non-smooth surface sealing ring material. In terms of metals, 9CrW18Mn is selected, the Young’s modulus and the Poisson’s ratio of which is set as $2 \times 10^{11}$ Pa and 0.3 separately. The nitrile butadiene rubber (NBR) is defined as the attribute of bionic non-smooth sealing ring, for which the operating temperature is -30$^\circ$C~120$^\circ$C, and the elasticity modulus is 2~5.5 MPa. Moreover, the isotropic elastic material is chosen for the linear elastic material of sealing ring, the Young’s modulus of which is set to be $6 \times 10^9$ Pa and the Poisson’s ratio is 0.499. Finally, selecting the Mooney-Rivlin to be the hyperelastic material of sealing ring, and the related parameters d is 0.001, C01 is -4.587 MPa, C02 is 0.6986 MPa, C10 is 7.2668 MPa, C11 is -2.187 MPa, C20 is 3.353 MPa.

3.3. Definition of Unit Attribute
The Solid185 is chosen for the unit type, which is primarily used for the 3-dimensional solid structure construction. The unit is defined by 8 nodes, and each node has three degrees of freedom that can shift towards X, Y, Z directions. The cell enjoys great resilience, stress stiffness, creep deformation, distortion and adaptability. At the same time, almost incompressible elastoplastic material together with the incompressible hyperelastic material is simulated from the hybrid mode

3.4. Contact Pair Setting
For the simulation model established in this paper, two contact pairs are existed, one of which is between the bionic sealing ring and roller cone, and another is at the surface of bionic sealing ring and bearing.

3.5. Mesh Generation
With regard to the two different kinds of materials, the mesh generation should be conducted according to their corresponding models. Specifically, for the mental material model which is developed for the roller bearing and shaft diameter, the unit size is controlled to 2mm when generating the mesh. While with respect to the NBR material used for the bionic non-smooth sealing ring, the unit size of the mesh is restrict to 1.5mm. As to the two contact pairs, the unit size of mesh is set to be 1mm.

3.6. Displacement and Load Application
The displacement load application in this simulation experiment includes two different procedures, one is assembly another is working. In terms of the assembly procedure, the load application is implemented by the relative displacement between the bearing and shaft diameter. Due to the $d_0$ is equal to 5mm, and $h_0$ is 6mm, the displacement during the assembly procedure is 1mm. Beyond, the shaft diameter is fully fixed which means fully restricted when applying the displacement, and then the relative motion is produced by the displacement of bearing so as to achieve the displacement load application.
The load application for the working procedure is simulated by applying the pressure on the model. In order to prevent the incoming of the outer mud to the seal cavity, and the dropping of the lubricating grease in the seal cavity, the maximum value of sealing surface contact pressure has to be higher than the inner and outer pressure difference in the seal cavity. Based on the actual situation, the maximum pressure difference is 0.7 MPa in the pit, therefore, the value of pressure difference during the working procedure in the simulation experiment is set to be 1 MPa. Besides, the bearing is fully restricted when applying the displacement as mentioned before, hence the pressure is only able to be imposed to the roller cone. The last required explanation that because the ribbed sealing ring has the same ontology structure and assembly method with the O type sealing ring, the load application is treated in the same way.

4. Experiment Results and Analysis

4.1. Contact Pressure Analysis

(1) The ribbed tooth sealing ring

The contact pressure of the ribbed teeth is shown in Figure 4. The figure shows that the contact pressure is mainly concentrated on the ribbed patterns for the ribbed teeth, with the largest contact pressure 2.5805 MPa occurs mainly on the end face of bottom width angles. There is no pressure on the contactless surface, larger pressure in the centre, smaller pressure on both sides, and comparatively large pressure on the junction of ribbed rack angles and pitch angles with partial contact.

![Figure 4. The Contact Pressure of Ribbed Teeth Sealing Ring](image)

(2) The O type sealing ring

The distribution of contact pressure for O type sealing ring is shown in Figure 5. It can be seen from the figure that the largest value of contact pressure is 3.2432 MPa which is occurred on the contact surface at both ends.

![Figure 5. The Contact Pressure of the O-shaped Sealing Ring](image)

From the above figure we can see that the value of the maximum contact pressure for Type C ribbed tooth sealing ring is 2.805 MPa, and for O type sealing ring is 3.2432 MPa. Furthermore, it also can be seen that the contact surface of the new designed ribbed tooth sealing ring is significant less than that of O type sealing ring. Thus, the ribbed tooth sealing ring has improved the excessive contact area problem of the traditional sealing structure. In conclusion, the ribbed tooth can effectively reduce contact pressure and the size of contact area.

4.2. Equivalent Stress Analysis

(1) The ribbed tooth sealing ring
Figure 6 shows the equivalent stress of the ribbed tooth sealing rings. It can be seen that the maximum value of equivalent stress for the ribbed teeth is 1.9847 MPa. The cross section appears to be dark blue, which represents that there is little stress and relevant equivalent stress distribution. No equivalent stress occurs on surfaces without contacts. The equivalent pressure is larger in the centre and smaller on both sides of the contact surfaces. Likewise, the pressure is larger on the junctions of the bottom width angles and pitch angles of the ribbed patterns.

![Figure 6. Equivalent Stress of the Ribbed Teeth](image)

(2) The O type sealing ring
The maximum equivalent stress of O type sealing ring is 2.7168 MPa which is occurred on the cross section of both ends, as shown in Figure 7. It is obviously that the stress distribution of the O type sealing ring has the higher local stress and lower local stress. The distribution trend of the equivalent stress with respect to the O type sealing ring is that: it is larger in the centre towards the direction of the rack angle and bearing contact surface, and weaker near the contact surface of roller cone and the ribbed rack angles.

![Figure 7. The Distribution of the Equivalent Stress for O Type Sealing Ring](image)

Both the ribbed tooth and the O type sealing ring have the maximum equivalent stress on the contact ends. The value of the maximum equivalent stress for ribbed tooth is 1.9847 MPa which appears on the connection of the angle of distance and bottom width angle. Compared with the O type sealing ring, the maximum equivalent stress of O type sealing ring is higher than the value of ribbed tooth. Therefore, a conclusion can be reached that ribbed tooth enjoy smaller contact area, lower contact pressure and less possibilities of deformation.

5. Conclusion
The existing roller cone bit bearing sealing ring has the bad disadvantages of abrasion-resistance and heat-resistance, as well as the problem of short life span. Thus, a novel sealing structure is designed on the basis of the bionic non-smooth surface theory so as to improve the structure style, slow down the wear and ageing, and prolong the life span of the O type sealing ring through the reduction of contact area. Moreover, the contact pressure together with the equivalent stress is analysed from the finite element simulation. Consequently, the conclusions can be acquired as bellow:

The ribbed tooth sealing structure presented in this paper is capable of reducing the contact surface, and the gaps exist between the racks can achieve the heat dissipation and reserve some lube to lower the drag. And eventually can realize the operating temperature reduction, the wear delay, and the working life extension of rubber rings through the above design.
References

[1] C Han, C Yu, Li Y, et al. Mechanical performance analysis of hollow cylindrical roller bearing of cone bit by FEM[J]. Petroleum, 2015, 1(4):388-396.

[2] B Rashidi, G Hareland, Z Wu. Performance, simulation and field application modeling of roller cone bits[J]. Journal of Petroleum Science and Engineering, 2015, 133:507-517.

[3] Y Zhou, Z Huang, L Tan, et al. Cone bit bearing seal failure analysis based on the finite element analysis[J]. Engineering Failure Analysis, 2014, 45(1):292-299.

[4] J Day, J Q Yu, R Baker. Innovation bearing and seal package improves roller cone bit performance and reliability[C]. SPE Asia Pacific Oil and Gas Conference and Exhibition, Brisbane, Queensland, Australia, 2010, 134239.

[5] Y Zhou, Z Q Huang, X B An, et al. Failure Analysis and Improvement of High-speed Roller Bit Bearing Bi-metal Seal[J]. Oil field Machinery, 2011, 40(8):50-53.

[6] Y Zhou, Z Q Huang, Q Li, et al. Finite element analysis of high-speed roller bit bearing bi-metal seal[J]. Science and Technology Information, 2011, (3): 423-424,433.

[7] X Xiao, B Chen, C M Sun, et al. Optimization of bi-metal seal structure of high speed roller bit bearing[J]. China Petroleum Machinery, 2014, 42(10):30-33.

[8] L A Otto. Rock bit having a flexible metal faced seal[P]. United States Patent: US9163458, Oct 2015.

[9] K Gao, Y H Sun, R F Gao, et al. Application and Prospect of Bionic Non-smooth Theory in Drilling Engineering[J]. Petrol Explor Develop, 2009, 36(4):519-522,540.

[10] J H Ji, Y H Fu, L Wei, et al. Experiment research on lubrication properties of laser surface texturing mechanical seal[J]. Journal of Drainage and Irrigation Mechanical Engineering, 2011, (5):427-431.

[11] T Sugihara, T Enomoto. Performance of cutting tools with dimple textured surfaces: A comparative study of different texture patterns[J]. Precision Engineering, 2017, 49:52-60.