Numerical simulation and abrasion life calculation of X-ring for hydraulic actuators of a certain aircraft

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Abstract—**In this paper, the X-ring for hydraulic actuators of a certain aircraft was analyzed in order to reinforce the sealing performance of hydraulic actuators, prolong the life of the sealing ring and prolong the service life of the sealing ring. The contact stress and equivalent stress under different compression are simulated by ABAQUS. The location where X-ring is prone to tear failure is found out. According to the contact stress, the wear life of X-ring is analyzed. Areas prone to wear have been found, and the wear life analysis method of X-ring is obtained. The work in this paper provides critical references for the engineering design and application of X-ring in the hydraulic actuator, and it can effectively reduce the failure rate and prolong the service life of the sealing ring. And they can be further applied in the safety analysis of airplane hydraulic actuator seal.**

Keywords-X-ring; contact Stress; Von Mises Stress; finite element analysis; wear longevity prediction

I. INTRODUCTION

In the field of aviation application, there are countless hydraulic actuators on both military aircraft and large passenger aircraft. The sealing performance of actuator under high pressure directly affects the pressure system of aircraft. Research shows that high pressure system can effectively increase the power-weight ratio of aircraft, reduce the weight and volume of aircraft. At present, the most commonly used sealing rings in aviation field are O-ring and square ring[1]. O-ring is easy to be machined and installed. However, O-ring is distorted easily, which leads to leakages in seal land. Square ring is easy to process and has little leakage. But the friction of the sealing ring is large and it is not easy to install. So both O-ring and square ring have limitations in engineering application[2].

X-ring is a circular rubber seal with four sealing lips, which can act as a two-way seal[3]. X-ring is similar to O-ring, and it has many advantages. X-ring will not twist and roll in reciprocating motion. X-ring has longer life than O-ring, because it requires less radial friction. The pressure distribution on the cross section of X-ring is uniform, so the sealing effect is better. A lubrication chamber is formed between the sealing lips, which improves the starting condition[4]. Because of these advantages, X-ring has wide applications in the field of aviation. A large number of X-type sealing seals are used in hydraulic driving system. The reliability of sealing ring has direct influence on the safety and reliability of hydraulic actuators.

Lee analyzed contact stress and deformed shape of a X-shaped ring shell under various compressive contact conditions, got contact stress and deformed shape by finite element analysis[5]. Zhou Chilou investigated the sealing characteristic of the rubber X-ring in high-pressure hydrogen service by ABAQUS. They find the X-ring seal may be superior to the O-ring seal[6]. Cao Wen-Han set up a numerical simulation method of O-ring wear-thermal-stress coupling, and analyzed the change rule of seal performance and life in the process of wear and the effect of medium pressure on sealing characteristics[7].

In this paper, the finite element model of an X-ring in the hydraulic actuators of a certain aircraft door actuators is established. The distribution of equivalent stress and maximum contact stress of X-ring is obtained by ABAQUS finite element analysis software. According to the results of finite element analysis, the wear rate and life of X-ring under different working conditions can be analyzed. The work in this paper provides critical references for the engineering design and application of X-ring in the hydraulic actuator, and it can effectively reduce the failure rate and prolong the service life of the sealing ring.

II. FINITE ELEMENT MODEL AND BASIC HYPOTHESIS OF X-RING

A. *Model of X-ring*

In order to facilitate the finite element analysis, the model is simplified to a planar two-dimensional graph, which is shown in Figure 1.

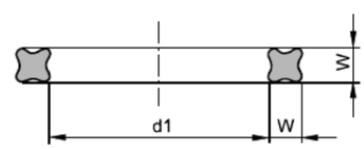

Figure 1. Cross Section of X-Ring

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The dimensions of sealing ring and groove are shown in Table 1.

TABLE I. THE DIMENSIONS OF X-RINGS AND GROOVES

X-ring		Groove		
diameter inside d_1(mm)	*diameter W(mm)*	*wide b(mm)*	*deep h(mm)*	*compression δ (mm)*
56.74	3.53	4	3.1	0.305~0.53

B. Material of X-ring

The material of sealing ring is rubber, a kind of non-linear material, which can be described by Mooney-Rivlin model[8]. There are lot of nonlinear hyper-elastic constitution equations, such as Ogden, Gent, Neo-Hookean, Mooney-Rivlin and so on. Mooney-Rivlin model can describe the mechanical properties of rubber accurately. Mooney-Rivlin equation was chosen, since it can describe mechanical properties of the rubber material within 35% deformation preferably. The strain energy density function can be described as the following[9]

$$W=C_1(I_1-3)+C_2(I_2-3)+a/2(I_3-1) \quad (1)$$

where, W is the strain energy density. C_1 and C_2 is the material quality of Mooney-Rivlin. I_1, I_2 and I_3 is the first, second and third strain invariant respectively. a is the bulk modulus of elasticity. For incompressible hyperelastic bodies,I_1 is equal to 1.

For incompressible hyperelastic materials, strain energy function can be expressed as scalar functions of strain or deformation tensor. The stress can be expressed as the partial derivatives of the strain energy functions. The constitutive equation is

$$S_{ij}=\frac{9w}{9E_{if}} \quad (2)$$

where S_{ij} is Piola-Kirchhoff stress, W is the strain energy density, E_{if} is components of Green strain tensors.

In this paper, Mooney-Rivlin binomial strain energy is used to describe the rubber hyperelastic materials. The parameters are shown in the table 2.

TABLE II. THE PARAMETERS OF MOONEY-RIVLIN MODEL

Paramter	*C_1*	*C_2*
Value	1.87	0.47

C. Boundary Conditions Setting

Because of the contact between X-ring and groove is highly non-linear, penalty function algorithm of contact element is used in the finite element analysis. There are two steps in the exertion of loads. The first step, in order to simulate the pressure of X-ring, we give the upper pressure plate a displacement along the radial direction of the X-ring. The second step, give the X-ring a right-facing pressure, which can simulate the hydraulic pressure of the X-Ring. The boundary conditions are shown in Figure 2.

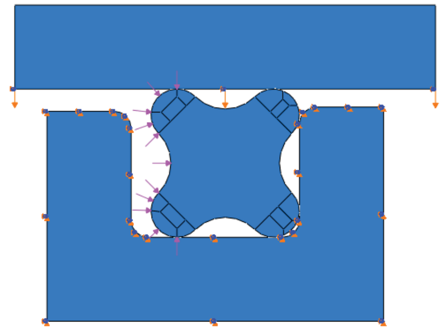

Figure 2. The boundary conditions

D. Mesh generation

The mesh model is shown in Fig 3.

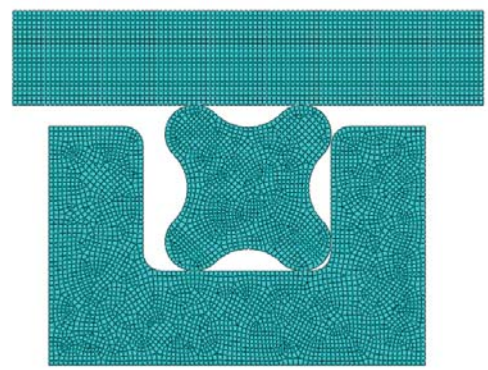

Figure 3. The mesh model

III. FEM RESULTS AND ANALYSIS

ABAQUS software was used to simulate the von mises stress and contact stress of X-ring. Get the trend of the change of the X-ring in different compressibility and fluid pressure. Figure 4 shows the typical Von Mises stress distribution of X-ring. Figure 5-7 shows the typical contact stress distribution of X-ring.

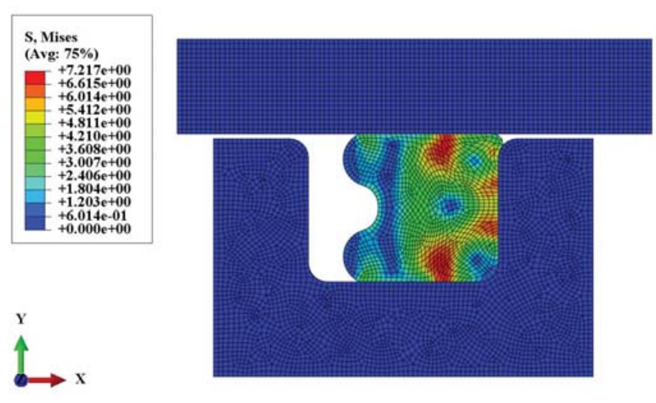

Figure 4. The result of Von Mises stress

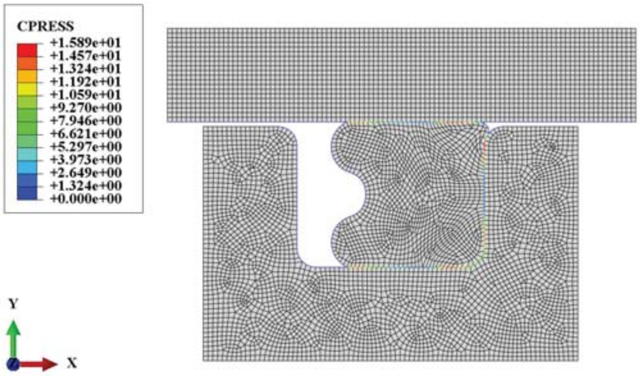

Figure 5. The result of contact stress

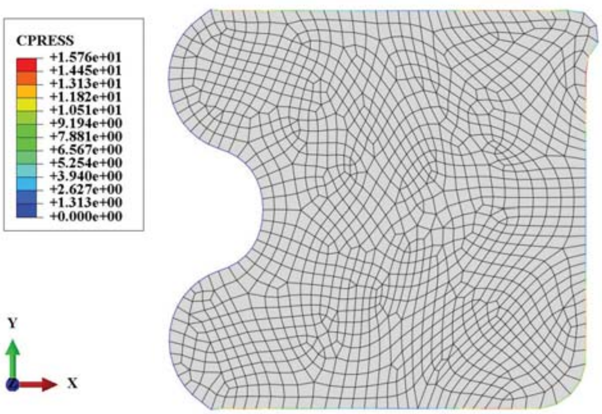

Figure 6. The contact stress of X-ring

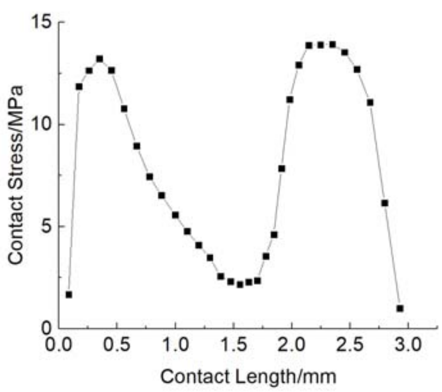

Figure 7. Distribution of contact stress on the upper surface

From figure 4, the equivalent stress distribution of each part of the X-ring can be obtained. The maximum equivalent stress occurs in the concave circle of the X-ring, which is prone to tear. From figure 5-7, contact stress distribution between the X-Ring and Groove can be obtained. The maximum contact stress occurs in the lip of the X-ring, which is prone to wear failure.

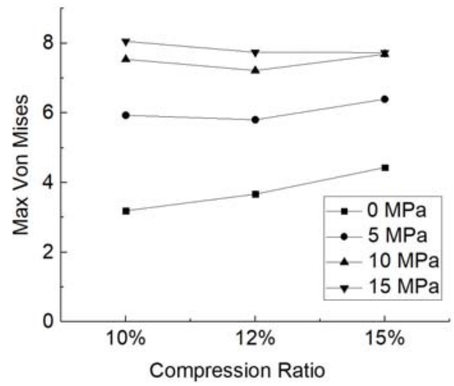

Figure 8. Relationship between compression and maximum

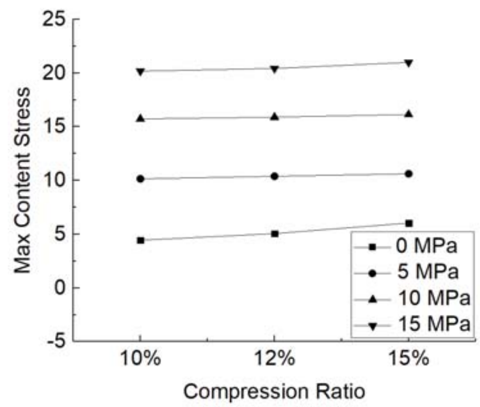

Figure 9. Relationship between compression and maximum contact stress

From figure 8 and Figure 9, we can see that when the fluid pressure is 5~15 MPa, the maximum equivalent stress decreases first and then increases with compression. When the fluid pressure is 0~15 MPa, the maximum contact stress increases with compression. Overall, the maximum contact stress is always greater than the fluid pressure, which can ensure good sealing performance and high reliability of the X-ring.

IV. LIFE ANALYSIS OF X-RING

According to the finite element analysis we can get the friction force of X-ring. Then we can calculate the wear life of X-ring by the mathematical model of reliability life on the wear of the X-ring. The formula is shown as the following

$$\Delta V = K \frac{F_N}{H} L \tag{3}$$

where F_N is the frictional force between the X-ring and the cylinder. H is hardness of the X-ring. According to the Sealing Ring Manual, the shaw hardness of rubber material is 85. L is the wear distance. ΔV is the wear volume of one round trip. K is coefficient of wear. There is a relationship between Coefficient K and coefficient of friction μ

$$\log K = 5\log\mu - 2.27 \tag{4}$$

M is equal to 0.2. So the Coefficient K if equal to 1.718×10^{-6}. F_N can be calculated by the following formula

$$F_N = \frac{F_{N_{MAX}} + F_{N1}}{2} \tag{5}$$

where F_{N1} is the contact force between X-ring and piston when X-ring has failed. F_{NMAX} is the contact force between X-ring and piston in initial state, it can be obtained by ABAQUS post-processing. The contact force curve is shown in Figure 10 below.

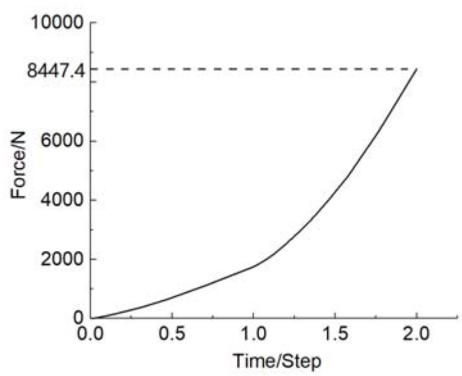

Figure 10. Contact force curve

Because of wear, the contact force between the seal ring and the shell surface decreases gradually until it fails. It is assumed that the wear volume of the X-ring is V when it fails. The wear volume can be described as the following

$$V_{wear} = \pi D_1 \times L_1 \times \Delta h = 322.65 \tag{6}$$

where D_1 and L_1 is the size of X-ring which can be obtained by measuring.

Finally, the wear longevity of X-ring are shown in the table 3. We can see that when the compression ratio is kept constant, the wear longevity decreased with the increase of the fluid pressure. This is principally because the contact force is increased with the increase of the fluid pressure. When the fluid pressure is kept constant, the wear longevity increased with the increase of the compression ratio. This is principally because the wear volume of the X-ring is increased with the increase of the compression ratio.

TABLE III. THE WEAR LONGEVITY OF X-RING

Compression Ratio	*Fluid pressure*	*Wear volume(mm^3)*	*Wear volume of one stroke (mm^3)*	*Life(Usage counter)*
10%	5MPa	243.53	1.59×10^{-3}	1.53×10^{5}
	10MPa	243.53	3.24×10^{-3}	7.51×10^{4}
	15MPa	243.53	5.11×10^{-3}	4.76×10^{4}
12%	5MPa	322.65	1.75×10^{-3}	1.85×10^{5}
	10MPa	322.65	3.46×10^{-3}	9.33×10^{4}
	15MPa	322.65	5.33×10^{-3}	6.05×10^{4}
15%	5MPa	444.05	2.04×10^{-3}	2.18×10^{5}
	10MPa	444.05	3.83×10^{-3}	1.16×10^{5}
	15MPa	444.05	5.71×10^{-3}	7.78×10^{4}

V. CONCLUSION

In this paper, the contact stress and equivalent stress under different compression is simulated by ABAQUS. The location where X-ring is prone to tear failure is found out. And according to the contact stress, the wear life of X-ring is analyzed. Areas prone to wear have been found, and the wear life analysis method of X-ring is obtained, which can provide some reference for engineering application. Seal life under the coupling action of wear and aging of sealing ring shoud be studied in the future work.

REFERENCES

[1] S. Y. Woo, Y. J. Lee, and I. M. Choi, “How to reduce no-linearity in effective area caused by the O-ring seal in a high-pressure balance,” Metrologica, vol 39, pp.537-541, 2002.

[2] H.S. Lee, B.S. chun , and S.Y. Kim, “Contact stress analysis on the X - shape ring, ” Materialwissenschaft und Werkstofftechnik, vol. 39, pp.192–193, 2008.

[3] Andrzej Kazmierczak, “Computer simulation of the new ring seal coaction in combustion engine,” Industrial Lubrication and Tribology, Vol 56, pp.210-216, 2004.

[4] R.M .Hoffman, “Nonline Elastic -plastic Analysis Using Modified Finite Quasi -prism Element,” Finite Elements in Analysis and Design, vol.40, pp. 449–460, 2004.

[5] Lee Hyun-Seung, Lee Young-Shin, and Lee Jung-Hyun, “A study on the contact stress analysis for X-ring,” Transactions of the Korean Society of Mechanical Engineers, vol. 32, pp. 733-739, September 2008.

[6] Z Chilou, H Mohan, and C Guohua, “Numerical study on sealing characteristic of rubber X-ring exposed to high-pressure hydrogen by considering swelling effect,” Industrial Lubrication and Tribology, vol. 71, pp. 133-138, January 2019.

[7] C Wenhan, J Gong ,W Honggang, “Numerical simulation on wear-thermal-stress coupling behavior of cap-seal seal and optimization design,” Journal of Zhejiang University, vol. 53, pp. 258-267, February 2019.

[8] Z. Zuo, and R. Liao, “Finite Element Analysis of Rubber Sealing Ring for Model 12150 Diesel Engine,” Chinese Internal Combustion Engine Engineering, vol. 17, no. 2, pp. 46-49, 1996.

[9] Zhou Entao, Li Jianxun and Lin Junzhe, “Finite Element Analysis of Piston Seals Property of Hydraulic Cylinders,” Lubrication Engineering, vol. 4, pp. 84-85, 2006.