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Article

Study on Sealing Characteristics of Sliding Seal Assembly of Aircraft Hydraulic Actuator

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Abstract: The hydraulic actuator, known as the "muscle" of military aircraft, is responsible for flight attitude adjustment, trajectory control, braking turn, landing gear retracting and other actions, which directly affect its flight efficiency and safety. However, the sealing assembly often has the situation of over-aberrant aperture fit clearance or critical over-aberrant clearance, which increases the failure probability and degree of movable seal failure, and directly affects the flight efficiency and safety of military aircraft. In this paper, the simulation model of hydraulic actuator seal combination is established by ANSYS software, and the sealing principle is described. The change curve of contact width and contact pressure of combination seal under the action of high-pressure fluid is drawn. The effects of different oil pressure, fit clearance and other parameters on the sealing performance are analyzed. Finally, the accelerated life test of sliding seal components is carried out on the hydraulic actuator accelerated life test rig, and the surface morphology is compared and analyzed. The research shows that the O-ring is the main sealing element and the role of the check ring is to protect and support the O-ring to prevent damage caused by squeezing into the fit clearance, so the check ring bears a large load and is prone to shear failure. Excessive fit clearance is the main factor affecting the damage of the check ring, and the damage parts are mainly concentrated at the edge of the sealing surface. This paper provides a theoretical basis for the design of hydraulic actuator and the improvement of sealing performance.

Keywords: hydraulic actuator; contact stress; sealing characteristics; maximum shear stress

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0 Introduction

Hydraulic actuators, known as the "muscles" of military aircraft, are responsible for flight attitude adjustment, trajectory control, aircraft braking turn, landing gear retracting and other actions, which directly affect the flight efficiency and safety of military aircraft[1].

However, in the maintenance process of aircraft hydraulic actuator, it often has the situation of over-aberrant aperture fit clearance or critical over-aberrant clearance, which is difficult to meet the design clearance fit requirements of engineering drawings. The failure probability and degree of hydraulic actuator mobile seal failure are increased, and then the flight efficiency and safety of military aircraft can be directly affected[2].

At present, many experts and scholars at home and abroad had conducted in-depth research on the sealing principle, sealing characteristics and failure mechanism of the aircraft hydraulic actuator, and achieved the fruitful research results.

Professor Zhong[3] established the axis metrical 2-D finite element model of the seal ring and conducted finite element analysis to obtain the law of the influence of the section compression and circumferential tension of the seal ring on the seal performance, and finally calculated the fatigue wear life.

Professor Zhong[4] established a finite element model of O-ring based on ABAQUS finite element analysis software, and analyzed the effects of precompression amount, fluid pressure, friction coefficient and motion speed on the sealing performance of O-ring.

Professor Wang[5] established an axis metrical model of an O-shaped rubber seal ring. By means of nonlinear finite element method, the contact stress under different working pressures and different compression rates was analyzed, and the distribution law of the contact width with the initial compression rate was obtained, which provided relevant reference for the structural design of the O-shaped seal ring.

Professor Li[6] established the simulation model of O-ring static seal of aviation hydraulic actuator with Workbench software, studied the O-ring stress distribution under different pressure amount, oil pressure and temperature, and analyzed the influence of factors such as pressure amount, oil pressure and temperature on the O-ring static seal performance and service life.

Professor Yang[7] studied the premature failure of O-ring gasket of an electro-hydraulic system in a nuclear power plant. Through a series of macroscopic and microscopic analysis methods, the root cause of this unexpected failure is found out.

Professor Mo[8] conducted a series of material characteristics on the premature failure of O-ring of bearing cover of radial thrust bearing chamber, including macro/micro morphology observation, matrix material inspection, performance evaluation, etc., to find out the cause.

Professor Zhang[9] analyzed the reasons for the failure of O-ring seal of pump floating ring sleeve. Considering the influence of clearance size change and fluid pressure sudden change, the working state of O-ring was simulated by finite element analysis, and the distribution law of dynamic equivalent stress and contact stress of O-ring seal was obtained.

Professor Chang[10] proposed a method for the wear simulation calculation of O-ring based on the friction and wear model of Archard and the analysis of ANSYS software, in view of the current situation that many types of products using O-ring wear fail and have no relevant wear simulation means. The method of grid reconstruction is used to solve the problem of simulating large deformation and material wear in the simulation process.

Professor Tang[11] established a finite element analysis model of actuator seal structure based on ANSYS to solve the leakage problem of O-ring under high pressure conditions, and the effects of medium working pressure, radial clearance and sealing ring diameter on the structural performance of dynamic/ static seals are analyzed.

Professor Zeng[12] established a simulation model of plane O-seal based on ABAQUS to solve the failure problem of plane O-seal in pure water system, and analyzed the influence law of seal clearance and medium pressure on seal performance such as principal stress, contact stress and shear stress.

Previous studies have focused on the stress/strain law of O-ring under the influence of structural/ environmental parameters, and lack of research on the internal stress distribution and internal failure mechanism of sliding seals under different failure forms.Compared with the previous research results, the hydraulic actuator studied in this paper has a long maintenance cycle and higher reliability requirements. The key sealing parts are easy to produce over-aberrant or critical over-aberrant clearance phenomenon with the size, which virtually increases the risk of seal failure.

Therefore, this paper studies the sealing characteristics of the sliding seal assembly of a certain type of aircraft hydraulic actuator. Firstly, the finite element simulation model of the sliding seal assembly of the hydraulic actuator is established, and the load distribution and damage location of the seal assembly are analyzed. The damage failure mechanism of the seal assembly is revealed, and the influence law of different working parameters on the sealing characteristics of the hydraulic actuator is studied, and then the correctness of the finite element simulation is verified by experiments.

1 Hydraulic Actuator Model and Sealing Characteristics

The hydraulic actuator is mainly composed of end cap, piston, piston rod, outer cylinder, sealing sleeve, O-ring and check ring, etc. A sliding seal is formed between the piston rod and the inner ring of the sealing sleeve, and the other sliding seals is composed of the inner wall of the outer cylinder and the piston outer ring integrated with the piston rod and. The sliding seal assembly is mainly composed of an O-ring and a rectangular check ring. The appearance of the hydraulic actuator is shown in Fig.1 and the partial section view is shown in Fig.2.

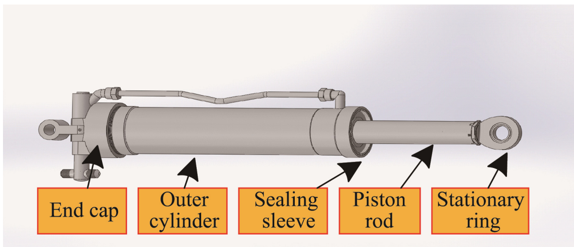

Fig.1 3D model of hydraulic actuator

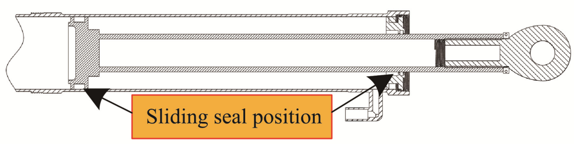

Fig.2 Partial profile of hydraulic actuator seal

Since the hydraulic actuator has a center-symmetric structure, it is necessary to establish a 1/2 symmetric model of the hydraulic actuator during Workbench simulation analysis, as shown in Fig.3.

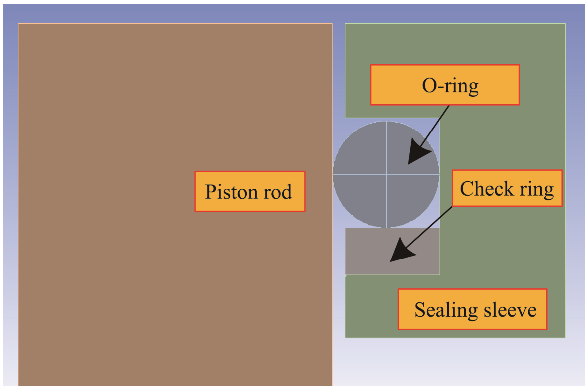

Fig.3 2D axisymmetric model

The stress-strain function of the O-ring in the hydraulic actuator seal assembly is established based on the superplastic Mooney-Rivlin model[13].

$$W=C_{10}(I_1-3)+C_{01}(I_2-3)+\frac{1}{d}(J-1)^2$$

Where, I_1 and I_2 are dimensionless stress tensor invariants, C_{10} and C_{01} are the material constants of rubber, C_{10}=1.328, C_{01}=0.066.

The O-ring is made of nitrile rubber, and the check ring is made of polytetrafluoroethylene. The material of the piston rod, outer cylinder and sealing sleeve are structural steel.

Fig.4 shows 2D sealing contact surface of the hydraulic actuator piston rod and the sliding sealing part of the sealing sleeve.

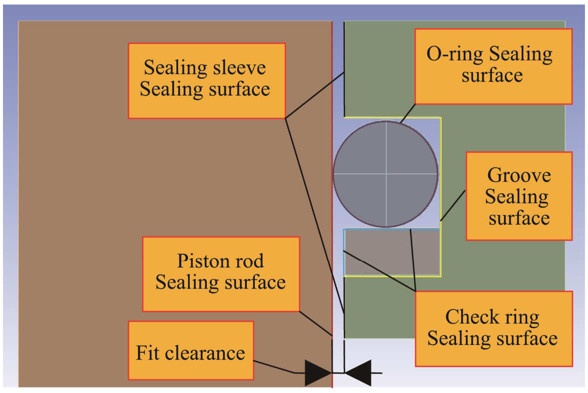

Fig.4 Schematic diagram of sealing two-dimensional contact surface

As shown in Fig.4, there is a fit relationship between the piston rod and the sealing sleeve for the hole shaft, and there is a certain initial mating clearance, so the sealing surface of the piston rod and the sealing cover of the sealing sleeve are not initially in contact.

The sealing contact characteristics of the hydraulic actuator seal assembly mainly include material nonlinearity, geometric nonlinearity and contact nonlinearity[14-15].

1.1 Material Nonlinear Characteristics

Because the sealing ring studied in this paper has more complex load deflection characteristics: when the external load is small, the rubber material follows Hooke's law, the elastic modulus remains unchanged, the stresses and the strains should become directly proportional; When the external load increases to the critical value, the elastic modulus decreases rapidly with the increase of the external load, and the rubber becomes soft. When the external load continues to increase to another critical value, the elastic modulus increases and the rubber hardens. Therefore, it is necessary to adopt the Mooney-Rivlin model to describe the strain energy function of O-ring and check ring materials:

$$\varepsilon=\frac{D-d}{d}\times 100\%$$

1.2 Geometric Nonlinear Features

When the external load is large, the O-ring and the check ring will have large deformation, which makes the geometric structure have strong nonlinear characteristics. Therefore, the complex loading and deformation process of O-ring and check ring needs to be discretized as a whole, so as to obtain the finite element equation of complete Lagrange nonlinear problem:

$$([{}_0^t k]_L+[{}_0^t K]_{NL})\{{}^0 u\}=\{{}_0^{t+\Delta t} R\}-\{{}_0^t F\}$$

Where,$[{}_0^t k]_L$ and$[{}_0^t k]_{NL}$ are the linear and nonlinear parts of the total rigid matrix; The global coordinate at time $\{{}^0 u\}$–t=0 is the incremental node displacement vector relative to the datum; $\{{}_0^{t+\Delta t} R\}$–$t+\Delta t$ is the external load vector of the moment; $\{{}_0^t F\}$ is the original stress at time t.

1.3 Contact Nonlinear Characteristics

Due to the highly nonlinear contact behavior characteristics between O-ring/check ring and cylinder wall/piston rod, its stiffness matrix is related to the displacement of the desired node, and the boundary conditions of the displacement node change all the time under the combined action of contact analysis, contact, contact extrusion, friction and other factors, so the contact state is highly uncertain. It is also the complexity and the key of this study.

In this paper, the penalty function element method is adopted, that is, a pseudo-element is established between the nodes of two contact surfaces to simulate the contact between surfaces to solve the problem. The basic idea is: in each step of the calculation, if there is a penetration of the main surface from the node, a larger contact force is introduced between the penetration point and the main surface, which is equivalent to adding a normal spring between the node and the main surface to be penetrated, and its size is proportional to the penetration depth and the stiffness of the main plate, which is called the penalty function value to limit the penetration of the main surface from the node.

2 Seal Failure Mechanism and Failure Criteria of Hydraulic Actuator Seal Assembly

2.1 Failure Mechanism of Hydraulic Actuator

Based on the field maintenance experience of a certain type of aircraft, it can be seen that the failure forms of the hydraulic actuator seal structure can be mainly divided into clearance bite, wear failure, aging failure and damage failure[16-18].

1. Clearance Bite

Because there is an initial clearance between the sealing surface of the piston rod of the hydraulic actuator and the sealing cover of the sealing sleeve, and the O-ring is made of rubber and has a soft texture, when the pressure oil is applied under the simulated working conditions, the O-ring will deform and move to the side without pressure or low pressure under the action of large medium pressure. It has direct physical contact with the piston rod sealing surface, groove sealing surface and check ring sealing surface, and even will be squeezed into the clearance between the piston rod sealing surface and the check ring sealing surface, resulting in a sudden increase in local stress of the O-ring, and bite on the O-ring sealing surface.

2. Wear Failure

Due to the relative movement between the piston rod and the sealing sleeve of the hydraulic actuator, the friction value is large, which makes the friction heat increase rapidly, resulting in the wear of the O-ring sealing surface, resulting in the seal failure.

3. Aging Failure

During long-term storage, due to the influence of oxygen, ultraviolet and thermal energy, the elastic modulus of the O-ring of the hydraulic actuator decreases, cracks appear on the outer surface, and the equivalent strength decreases gradually. And under the action of high temperature/high speed/high frequency/high pressure impact load for a long time, it further aggravates the aging trend of O-ring elasticity reduction, surface hardening, and even fatigue cracking, which reduces the sealing characteristics of the hydraulic actuator.

4. Damage Failure

Because the clearance between the piston rod and the sealing sleeve is very small, the O-ring and the check ring scratch phenomenon are easy to occur during assembly, which leads to a rapid increase in stress concentration in the actual work process, and greatly reduces the sealing characteristics of the O-ring and the check ring.

2.2 Failure Criteria of Hydraulic Actuator

According to the actual operating conditions and design criteria, the maximum contact compressive stress and shear stress are set as failure criteria in order to ensure the reliability of the seal structure of the O-ring and the check ring[19-20].

1. Maximum Contact Compressive Stress Criterion

When the maximum contact compressive stress between the O-ring and the piston rod sealing surface and the groove sealing surface is less than the working pressure of the hydraulic oil, the hydraulic oil will leak and the seal will fail. Therefore, the maximum contact compressive stress is the primary condition for the failure criterion and failure criterion of the hydraulic actuator seal assembly, ensure that the minimum contact stress of the seal contact surface is always greater than the working oil pressure.

2. Shear Stress Criterion

When the maximum shear stress of the check ring between the sealing sleeve sealing surface and the groove sealing surface exceeds the shear strength of the polytetrafluoroethylene material, the check ring will be destroyed, resulting in seal failure.

2.3 Sealing Mechanism of Hydraulic Actuator

The installation process in the sliding seal part of the simulated hydraulic actuator is shown in Fig.5, Because the section diameter of the O-ring is larger than the groove depth of the sealing part, a certain precompression will be generated, The O-ring surface has a certain length of contact with the piston rod sealing surface, the groove sealing surface and the check ring sealing surface. It can be inferred that the sealing surface after applying oil pressure is based on the extension of the length of the three sealing contact surfaces.

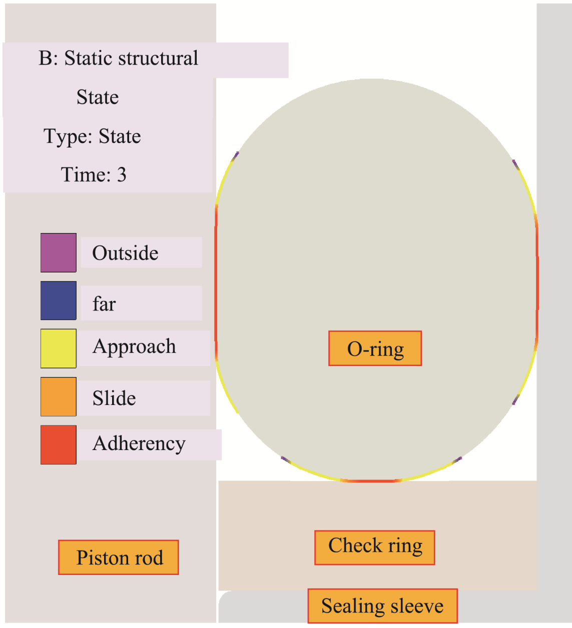

Fig.5 Sealed contact state under precompression

Under the action of 20 MPa hydraulic oil pressure, the O-ring is in close contact with the check ring, the piston rod and the groove sealing, and the hydraulic oil will exert a certain pressure on the O-ring. When the pressure is large, O-rings produce a large amount of deformation, the mutual contact process is shown in Fig.6.

When the hydraulic oil pressure exceeds a certain rating value, the check ring will have a large deformation, and part of it will be squeezed into the fit clearance between the hole shaft, as shown in Fig.7, the black wire frame in the Figure is the state of the piston rod, O-ring, check ring and sealing sleeve before deformation. After the application condition, the O-ring and the check ring are deformed. When the fit clearance is larger than the design clearance, it is easy to cause the check ring part to be squeezed into the initial clearance, and the maximum shear stress exceeds the shear strength, resulting in shear failure. Fig.8 is the three-dimensional distribution diagram of the shear failure point of the check ring.

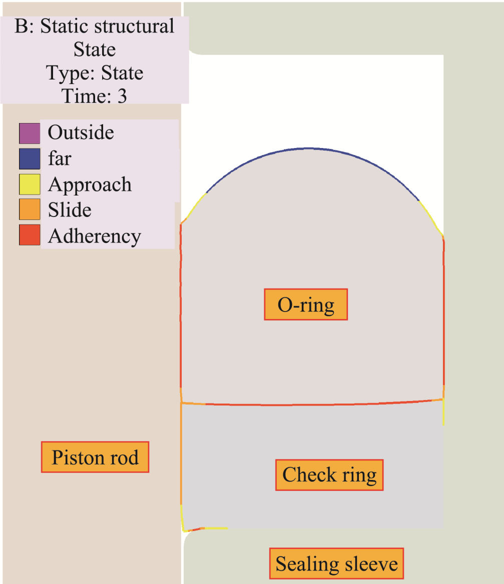

Fig.6 Contact state of each sealing surface

When the initial fit clearance of the hole shaft exceeds a certain rating, the deformation of the check ring under the action of hydraulic oil is far less than the fit clearance between the hole shaft, resulting in the O-ring being forced into the initial clearance and a large stress concentration phenomenon and failure as shown in Fig.9 and 10.

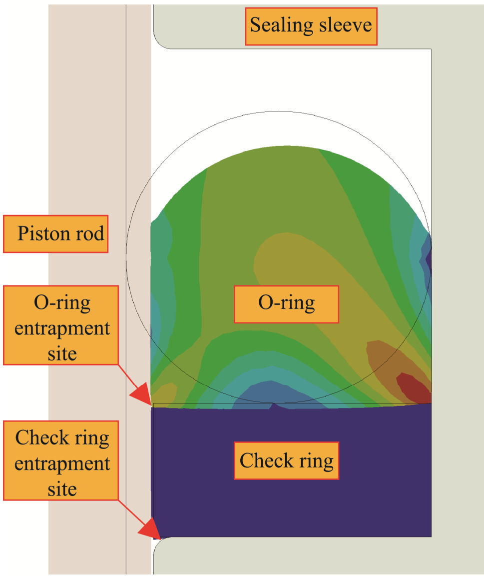

Fig.7 Deformation diagram of sealing assembly under working conditions

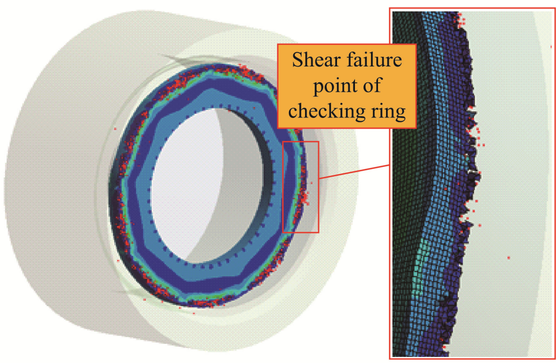

Fig.8 3D diagram of edge damage of check ring

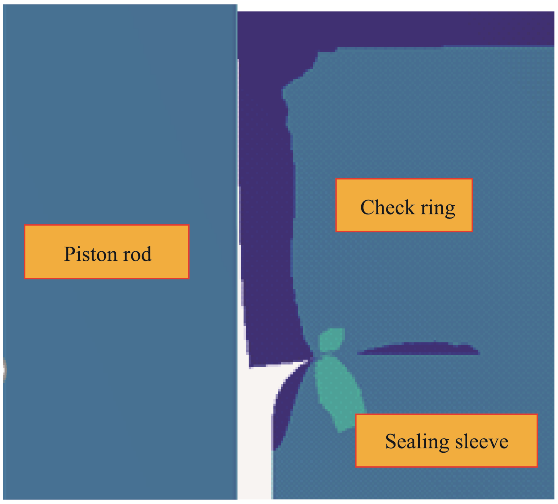

Fig.9 Schematic diagram of check ring diapir

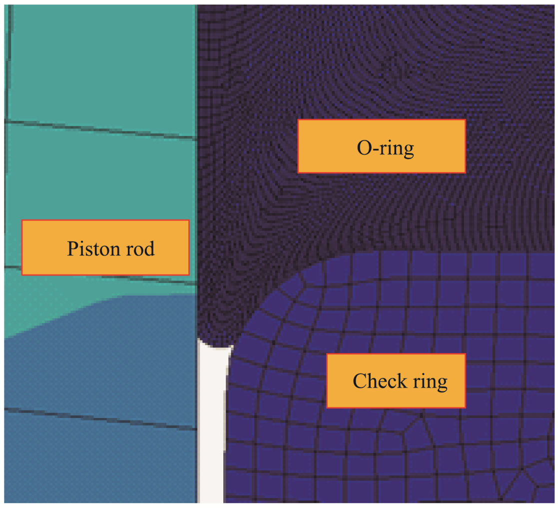

Fig.10 Schematic diagram of O-ring diapir

3 Sealing Contact State and Stress Analysis of Hydraulic Actuator Seal Assembly

Set the hardness of O-ring to 78 Hr, shear strength to 7.8 MPa. The shear strength of the check ring is 25 MPa and the friction coefficient is 0.16.

The simulation process of compression and extrusion deformation of hydraulic actuator seal assembly mainly includes two parts.

The first step is to move the piston rod upward to simulate the radial precompression process during the installation of the O-ring /check ring.

The second step is to apply uniform load to the non-contact area on one side of the O-ring to simulate the axial compression process of the O-ring/check ring under the action of fluid pressure.

Finally, the contact state and pressure distribution of the contact surface of the hydraulic actuator seal part after radial precompression and 20 MPa fluid axial compression are obtained, as shown in Fig.11 and Fig.12.

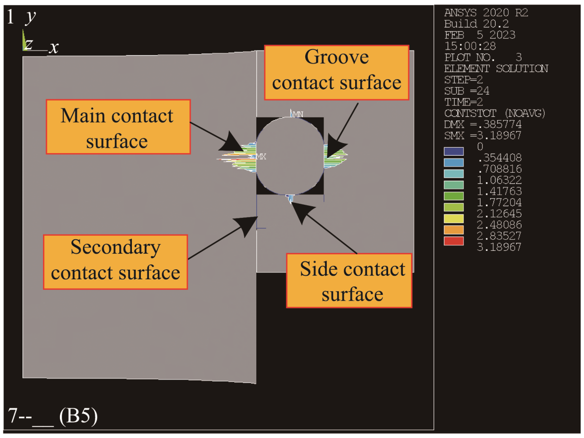

Fig.11 Contact pressure distribution of sealing part after radial precompression

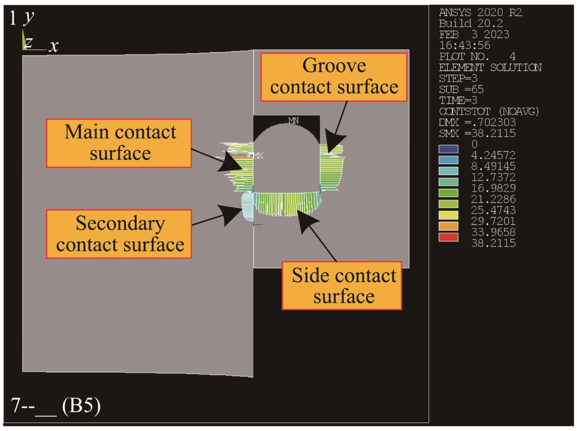

Fig.12 Contact pressure distribution of sealing part after axial compression

According to Fig.4, Fig.6, Fig.11 and Fig.12, under the action of the initial fit clearance, the main contact surface formed by the close contact between the O-ring and the piston rod, the groove sealing surface formed by the close contact between the O-ring and the groove, and the side contact surface formed by the close contact between the O-ring and the check ring. The check ring is in close contact with the piston rod to form a secondary sealing surface.

As can be seen from Fig.11, when the O-ring/check ring is installed, due to the influence of the initial clearance, the O-ring is slightly deformed under radial precompression, and the secondary contact surface has no contact pressure, while the other contact surfaces all have certain precompression contact pressure, so as to ensure that the hydraulic actuator seal assembly has strong oil sealing capacity under certain oil pressure.

The simulated contact stress distribution of each contact surface of the O-ring is shown in Fig.13, and the relationship curve between the contact width and contact pressure is shown in Fig.14.

As can be seen from Fig.13, the contact stress value in most areas in the middle of the contact surface is the largest, and the contact stress value decreases continuously from the middle to both sides along the contact surface, and the contact stress at the two edges of each contact surface decreases to close to 0 MPa.

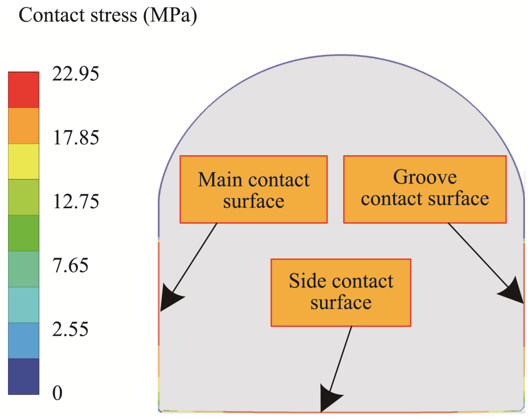

Fig.13 Contact stress distribution diagram of O-ring

The relationship curve between contact width and contact pressure was obtained by simulation, as shown in Fig.14.

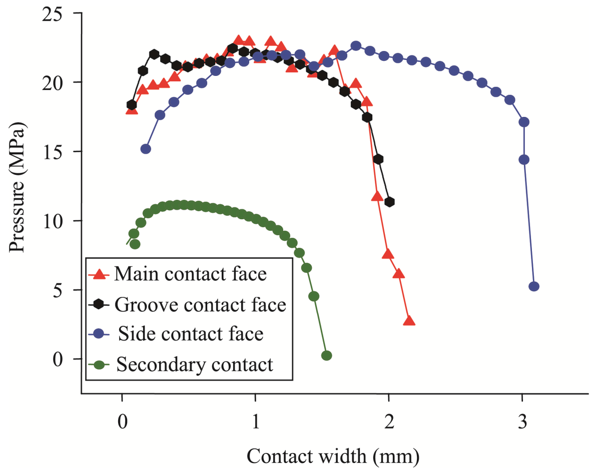

Fig.14 Contact pressure distribution curve of each contact surface of combined seal

According to Fig.12, 13 and 14, the contact pressure of the four contact surfaces firstly increases and then decreases with the increasing of the contact width. Among them, the maximum contact pressure of the main contact surface, groove contact surface and side contact surface are greater than 20 MPa of the working oil pressure, while the maximum contact pressure of the secondary contact surface is less than 20 MPa.

According to Fig.16, 12 and 13, the contact pressure of the main contact surface, groove contact surface and side contact surface is greater than the working oil pressure, and the sealing contact state is in the adhesive state at a longer contact width. The three sealing surfaces are well sealed with no risk of leakage.

The contact pressure of the secondary sealing surface is less than the working oil pressure, and the contact state of the sealing surface is sliding. The secondary contact surface does not have the main sealing effect, and the O-ring is the main sealing part.

The elastic modulus and shear strength of the check ring are much higher than that of the O-ring, which can effectively reduce the extrusion damage and stress concentration of the rubber material under the action of high-pressure hydraulic oil, and improve the service life of the O-ring.

4 Maximum Shear Stress of Hydraulic Actuator Seal Assembly

4.1 Influence of Fit Clearance on Sealing Performance

Fig.15 shows the stress change curve of the check ring under different fit clearance.

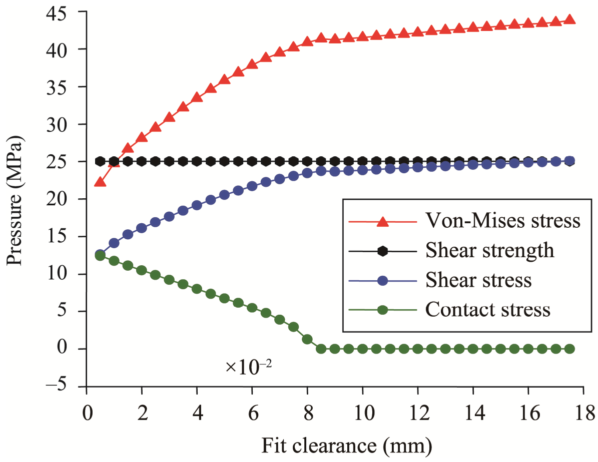

Fig.15 Change curve of check ring stress with fit clearance

According to Fig.15, at the operating oil pressure of 20 MPa, the equivalent Von-Mises stress and maximum shear stress increased with the increasing of the fit clearance, while the increasing rate decreased continuously, and the increasing rate remained basically unchanged when the clearance was greater than 0.08mm. The maximum shear stress value is less than and constantly close to the shear strength of the check ring 25 MPa, and the contact stress value of the check ring sealing surface decreases with the increase of the fit clearance. When the fit clearance is greater than 0.08mm, the contact stress decreases to zero and remains unchanged.

It can be inferred that when the fit clearance is less than 0.08mm, the maximum shear stress of the retainer is less than the shear strength of the check ring material, and the contact stress is greater than 0, that is, the sealing surface of the check ring is in close contact with the sealing surface of the piston rod, and the check ring can have a certain protection effect on the O-ring. Therefore, it can meet the normal service requirements of military aircraft.

When the fit clearance is greater than 0.08mm, the contact stress of the check ring is 0, and the maximum shear stress is infinitely close to the shear strength of the check ring material. In combination with Fig.7 and 9, a dangerous section appears in the part that is easy to be squeezed into the fit clearance. With the further increase of the fit clearance, it is cut off at the dangerous section. the result is seal failure of actuator.

According to Fig.16, under the operating oil pressure of 20 MPa, the contact stress of the O-ring basically kept fluctuating within the range of (22,25) MPa and was always greater than the operating oil pressure with the increasing of the fit clearance, and the equivalent Von-Mises stress and maximum shear stress increased slowly with the increasing of the fit clearance. The maximum shear stress is always less than the shear strength of the O-ring material.

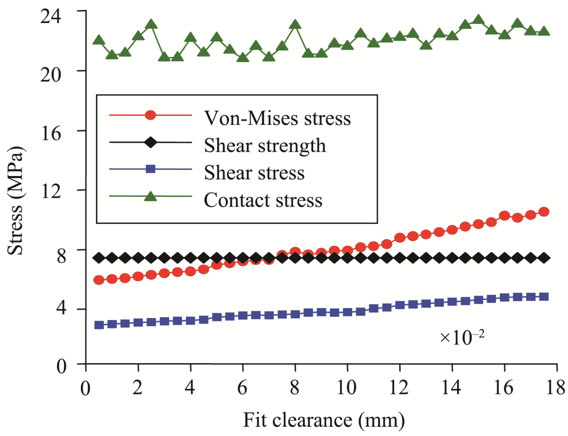

Fig.16 O-ring stress variation curve with fit clearance

It can be inferred that the contact stress of the O-ring is always greater than the working oil pressure, which can meet the normal service requirements of actuator. Even in the case of a large fit clearance, the maximum shear stress is still less than the shear strength of the O-ring material, it is not easy to be damaged by shear. Compared with the O-ring compression deformation/stress shear damage, the check ring is more prone to shear failure, resulting in the failure of the seal assembly structure, so the check ring shear failure is the key part and weak link of the hydraulic actuator.

4.2 Influence of Working Oil Pressure on Sealing Performance

Fig.17 shows the curve of the gradual change of different stresses on the check ring with the operating oil pressure when the fit clearance is 0.015mm.

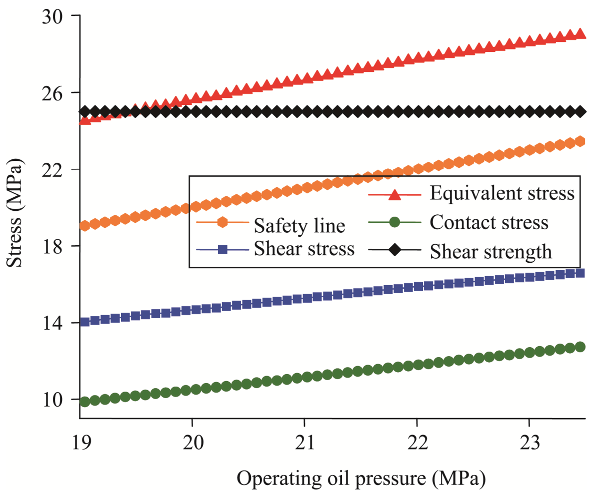

Fig.17 Variation curve of operating oil pressure and check ring stress

As can be seen from Fig.17, with the continuous increase of operating oil pressure, the stress borne by the check ring also increases. The equivalent stress value is the largest, the contact stress is always less than the stress value of the pressure safety line, and the maximum shear stress is always less than the shear strength of the check ring material.

It can be inferred that the contact stress of the sealing surface of the check ring is less than the working oil pressure, and the main role is to protect the O-ring from shear damage, and itself does not play a sealing role. The maximum shear stress is always within the safe range with the increase of the operating oil pressure. it can ensure the normal service of actuator.

Fig.18 shows the curve of different types of O-ring stress changing gradually with operating oil pressure when the clearance is 0.015 mm.

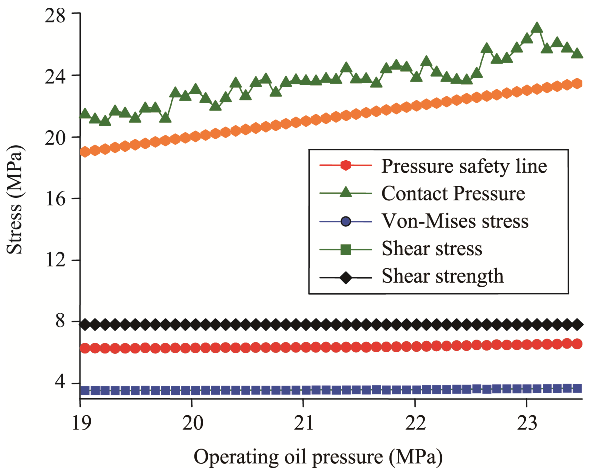

Fig.18 Curve of O-ring stress variation with operating oil pressure

As can be seen from Fig.18, as the operating oil pressure continues to increase, the maximum contact pressure borne by the O-ring presents a trend of slow increase with fluctuations, but it is always greater than the operating oil pressure. The maximum shear stress and equivalent stress remain basically unchanged, the contact stress is always greater than the stress value of the pressure safety line, the sealing condition is good, and the hydraulic actuator can be normally served.

5 Experimental Study on Sealing Characteristics of Hydraulic Actuator

Set accelerated life test bench is built and the failure characteristics of sliding seal assembly under typical working conditions is studied in order to verify the correctness of the theoretical analysis of the performance of aircraft hydraulic actuator seal assembly.

5.1 Introduction of Actuator Acceleration Life Test Bench

Fig.19 is the hydraulic actuator acceleration life test bench, and Fig.20 is the hydraulic principal acceleration life test bench.

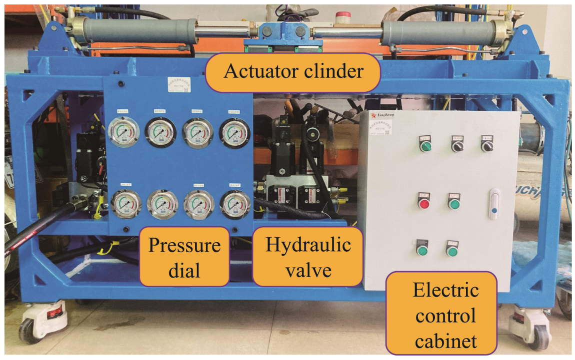

Fig.19 Hydraulic actuator acceleration life test bench

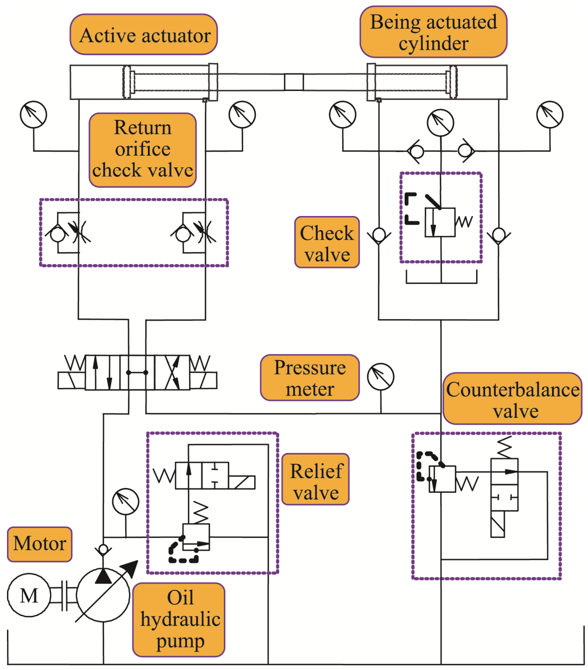

Fig.20 Hydraulic principle of hydraulic actuator acceleration life test bench

The key parameters of the self-designed hydraulic actuator acceleration life test bed are shown in Tab 1.

Table 1 Key equipment and parameters

Equipment	Equipment type	Argument	Value
Hydraulic oil pump	Rexroth	Rated pressure/MPa	0~35
Actuator cylinder	/	Working pressure/MPa Retraction frequency/s	20 6-10
Hydraulic valve	VTOZ	Maximum pressure/MPa	0~35
Oil temperature	/	℃	30-50
Working flow	/	L/min	0~20

The hydraulic actuator acceleration life test bench uses two reciprocating actuators against each other to act as a load on the top actuator, adjusts the oil pressure of the relief valve opening under simulated working conditions, adjusts the throttle valve opening speed of the actuator under simulated working conditions, and calculates the times by reciprocating action time.

The hydraulic actuator simulates the work of actual service state under the motion of the accelerated test bed. When the motion time reaches the actual maintenance time, the accuracy of the theoretical simulation is verified by disassembling and cleaning, an electron microscope is used to observe the change of the surface microstructure of the sliding seal component and measure the change of the matching clearance of the metal seal part, so as to verify the correctness of the theoretical simulation.

5.2 Sealing Part metal Fit Clearance Measurement

Through simulating the actual service action and maintenance and disassembly of the hydraulic actuator cylinder for many times, the inner diameter dial indicator and outer diameter micrometer were used to measure the metal inner diameter and outer diameter of the sealing part and calculate the fitting clearance of the sealing part, and the changing curve of the fitting clearance of the sealing part was obtained with the increase of the service time.

The measuring method of the sliding seal part of the hydraulic actuator is shown in Fig.21. The variation curve of fit clearance with service time is shown in Fig.22.

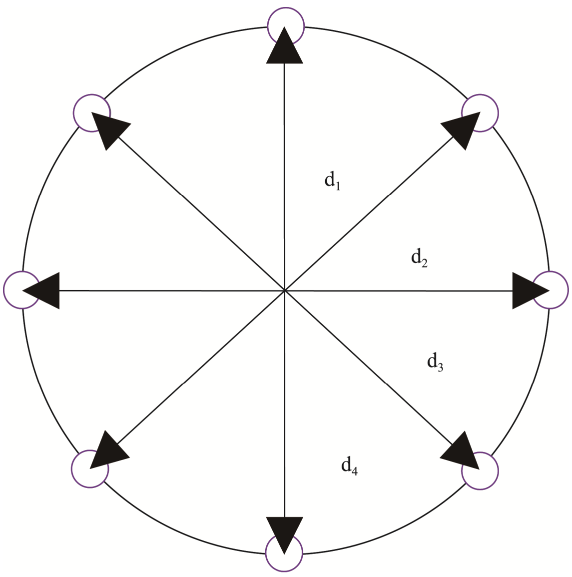

Fig.21 Measurement method diagram

As can be seen from Fig.22, the metal fit clearance of the sliding seal part of the hydraulic actuator here increases slowly with the increase of service time. Under normal service and maintenance time, the fit clearance of the sliding seal keeps expanding and increasing, but the change range is within the design clearance range. In the test, the pressure and temperature of the hydraulic actuator are not abnormal, which can ensure the normal operation requirements of service.

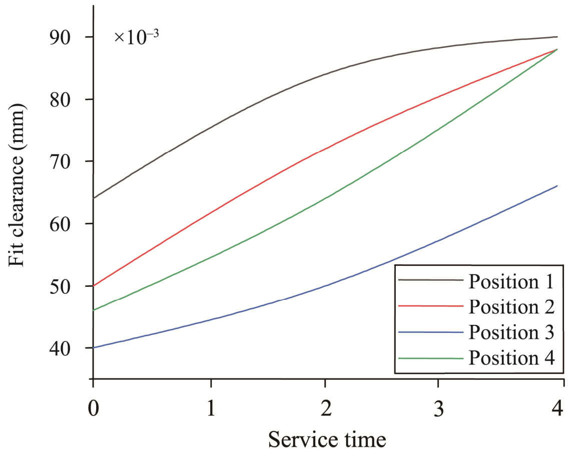

Fig.22 Variation curve of fit clearance with service time

5.3 Observation of Wear Surface Morphology of Sealing Components

Combined with the simulation law of the hydraulic actuator seal assembly, the service status under different working conditions was simulated. After a long maintenance cycle, it was disassembled and the structural changes of the sealing part and the changes of the surface morphology of the sealing component were observed to verify the accuracy of the finite element simulation.

Fig.23~Fig.26 respectively show the changes in the macroscopic and microscopic morphology of the hydraulic actuator seal assembly before and after the test, given different working conditions of the oil pressure under the normal design coordination clearance.

Fig.23 Photo of O-ring

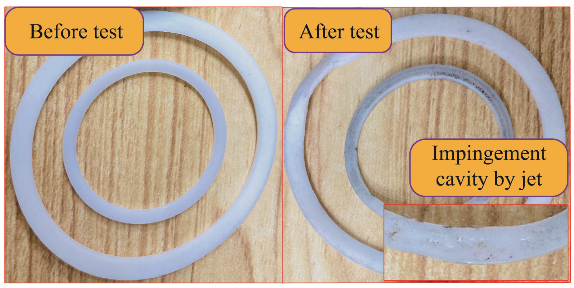

Fig.24 Photo of check ring

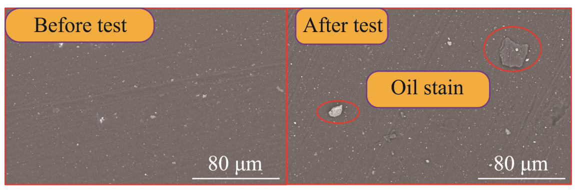

Fig.25 Micro-surface morphology of O-ring

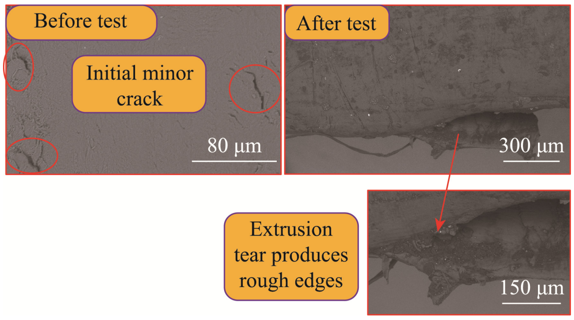

Fig.26 Microscopic surface morphology of check ring

Compared with Fig.23 and Fig.24, it can be seen that when the clearance of the seal part of the hydraulic actuator meets the service requirements, changing different working oil pressure can simulate a service condition with a long maintenance cycle, the sliding seal component is affected to different degrees. the surface of the O-ring is smooth without any obvious changes such as wear marks or tears. Polytetrafluoroethylene check ring produces a certain degree of extrusion deformation under high pressure. With the continuous expansion of the hydraulic actuator, the check ring is affected by different degrees of impact and produces pits.

According to Fig.23~Fig.26, the O-ring is basically intact after different oil pressure service conditions, while the outer edge of the sealing contact surface of the polytetrafluoroethylene check ring is shear torn due to high pressure impact extrusion and accompanied by different degrees of rough edges. The area with large compression deformation has a certain degree of crack, and the whole has a significant damage.

Fig.27~Fig.30 respectively show that after the hydraulic actuator is under normal working oil pressure and is simulated to be in service for a long time, there is an out-of-clearance fault in a sliding seal part. After measuring, the clearance exceeds the designed fit clearance, and the seal assembly here is obviously damaged.

In Fig.27~Fig.30, the O-ring and the check ring have different degrees of damage.

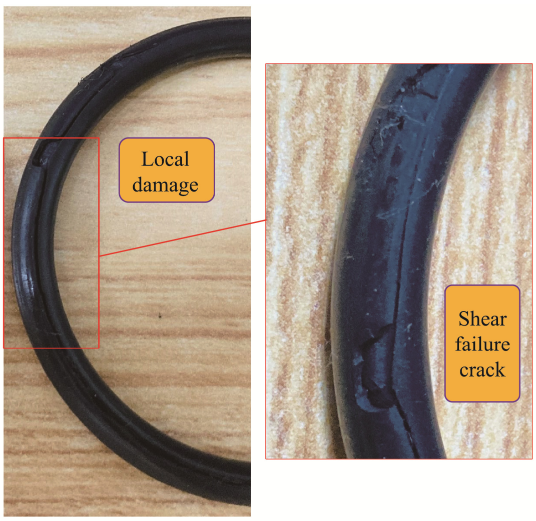

Fig.27 Damaged O-ring

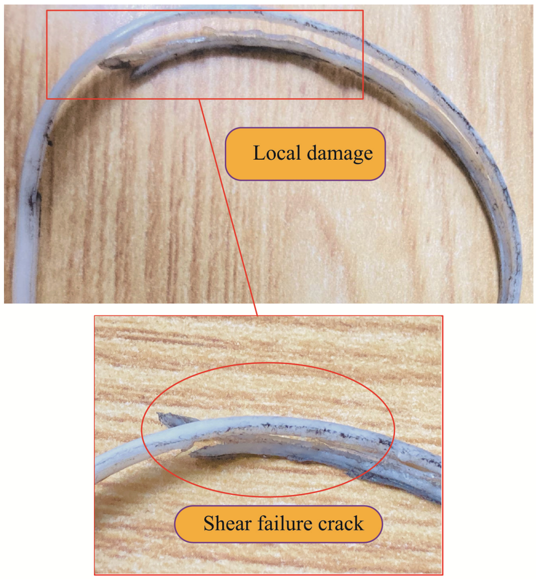

Fig.28 Damaged check ring

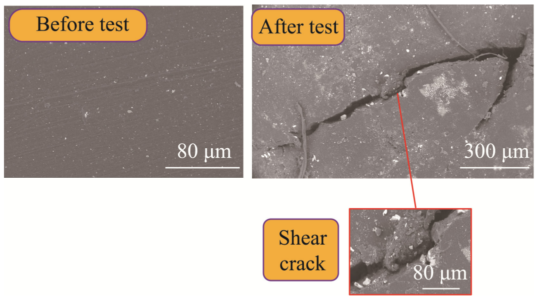

Fig.29 Microscopic morphology of damaged O-ring

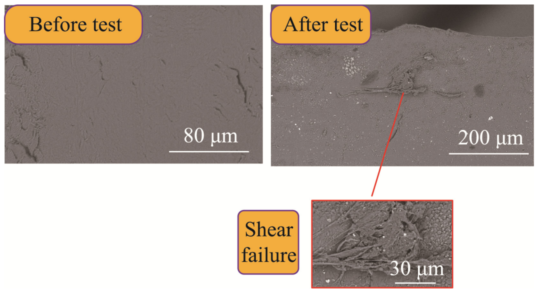

Fig.30 Microscopic morphology of damaged check ring

Compared with the finite element simulation analysis, the large clearance of hydraulic actuator sliding seal is easy to cause significant damage to the seal assembly, resulting in seal failure. The influence of oil pressure change on sealing components is obviously less than that of fit clearance. Excessive oil pressure is not enough to cause the failure of seal assembly, and when the fit clearance is larger than the design standard, it is enough to cause significant damage to the seal assembly and seal failure.

Combined with the finite element simulation results, it can be seen that the main damage part of the sliding seal assembly of the hydraulic actuator appears at the edge of the sealing surface of the polytetrafluoroethylene check ring. It can be inferred that the sliding seal fit is too large due to eccentricity or metal wear during service, which leads to the shear failure of the check ring when it is squeezed into the piston rod and the sealing sleeve seal fit clearance. The O-ring has no obvious damage under the protection of the check ring and can continue to serve normally.

6 Conclusion

1. The contact pressure of the check ring/ O-ring increase with the increase of hydraulic oil pressure, and the order of main contact pressure is hydraulic oil pressure > O-ring > check ring, so the O-ring is the main sealing element of the hydraulic actuator.

2. The impact of the fit clearance on the sealing performance is greater than that of the other parameters. When the fit clearance is large, the O-ring is not easy to be damaged while the check ring is easy to be shear damaged, the ability of protecting O-ring is reduced, The main failure mode of sliding seal of hydraulic actuator is the seal failure caused by the shear failure of sealing check ring.

3. The sliding seal assembly of the hydraulic actuator is damaged due to excessive clearance, and the damaged parts are mainly concentrated at the edge of the sealing surface of the polytetrafluoroethylene check ring. The clearance over difference will cause extensive damage to the sealing assembly and the seal failure.

Author Contributions:

Li Weinan: Provide project research object parameters and provide project financial support; Shi Saixin: Paper writing, simulation analysis of actuator sliding seal, stress cloud map drawing, actuator test bench design and construction; Zhang Jiawei: Paper writing, curve drawing, actuator sliding seal test; Tang Hao: Paper writing, micro-morphology observation of sliding seal components; Tang Hongxia; Chen Liang: The content of the paper involves secret review; Zhao Jianhua: Guide the paper writing, improve the structure of the paper.

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Data Availability:

The authors declare that the main data supporting the findings of this study are available within the paper and its Supplementary Information files.

Conflict of Interest:

The authors declare no competing interests.

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