Analysis of fatigue characteristics of dual-drive axial piston motor cylinder block

W W Shi¹, H S Deng¹,², *, D B Hong¹ and X Y Shi¹

¹ School of Mechanical Engineering, Anhui University of Science and Technology, Huainan 232001, China;
² State Key Laboratory Mining Response and Disaster Prevention and Control in Deep Coal Mines, Anhui University of Science and Technology, Huainan 232001, China

* Corresponding author: H S Deng, dhs1998@163.com

Abstract. To improve the service life of axial piston motor with dual-driving, the cylinder model of axial piston motor with dual-driving is established, the cylinder force is derived and used as the boundary condition, based on the results of finite element analysis, the S-N curve of the cylinder material is modified and fatigue solution is carried out. The results show that: the main stress and strain of the cylinder block are mainly located at the cylinder block hole and the flow distribution window, the stress concentration is generated at the structural mutation, and the stress and strain are the largest; the cylinder block will be repeatedly subjected to high-pressure oil action will produce fatigue damage, fatigue damage mainly occurs at the location of higher stress.

1. Introduction
The cylinder block is one of the important rotating parts of axial piston motor with dual-driving [1], which is highly susceptible to fatigue loss damage under the repeated action of unbalanced torque and high-pressure oil [2]. Wang Yan et al. [3] analyzed the cylinder fatigue of an aviation piston pump and pointed out that the working pressure has a large influence on the fatigue loss and life of the cylinder. M. M. Rahman et al. [4] used the results of finite element analysis to predict the fatigue life of the cylinder using different frequency response methods. Lu et al. [5] analyzed the fatigue life of roller-slide bearings and pointed out that the damage of roller-slide bearings is smaller and more resistant to fatigue than cylindrical roller bearings. Xu Bin et al. [6] analyzed the structural strength and fatigue of light alloy wheels under bending conditions based on theoretical calculations and finite element method and pointed out that the finite element analysis was consistent with the results of theoretical calculations. The above studies provide theoretical guidance and a basis for the analysis of fatigue characteristics of the cylinder block of common axial piston motors.

The axial piston motor with dual-driving is a new type of hydraulic motor, and its mechanical characteristics are better than the common axial piston motor. To prolong its service life, this paper simulates the cylinder stress, strain, and fatigue life by establishing a mathematical model and structural model, and then compares the cylinder life results to obtain the factors influencing the cylinder life.
2. Finite element analysis

2.1. Cylinder block finite element modelling

The three-dimensional model of the dual-driving motor cylinder block was established and imported into the finite element analysis software. The structural material of the cylinder block is alloy steel with an elastic modulus of 210 GPa, a density of 7850 kg/m³, a tensile strength of 460 MPa, yield strength of 250 MPa, and a Poisson's ratio of 0.27.

Considering the porous structure of the cylinder, the cross-center angle is set to refine, the smooth and excessive are set to medium and fast respectively, and the minimum mesh size is 3 mm. The mesh generated a total of 47,489 cells and 197,617 nodes, and the average quality of the grid is 0.786.

The cylinder is axially and radially constrained by sett cylindrical constraints, with only the degrees of freedom to rotate about the z-axis.

The cylinder is mainly subjected to high-pressure oil pressure in the working process, where the oil pressure can be subdivided into the oil pressure between the valve-plate pair and the piston pair. According to the required swash plate support reaction force applied load, and the parameters into the formula to solve the relationship between the swash plate support reaction force and oil pressure, the results are shown in Figure 1.

Due to the assembly error between the plunger and the cylinder plunger hole will always exist a narrow gap, the plunger in the cylinder hole reciprocating movement, will bring the high pressure fluid into the gap, by concentric annular gap flow theory can be known, concentric circular gap flow pressure gradient service \( \frac{dp}{dx} \) obeys the law of linear decline, thus piston pair gap fluid pressure can be calculated, and finally get the cylinder hole pressure loading as shown in Figure 2.
The internal and external row seal zone of the cylinder distribution window is the variable pressure zone, and its flow conforms to the parallel disc radial flow model [7], and its variable pressure zone pressure varies with the radius, and the original coordinate system is used as a reference to establish the column coordinate system, and the radius is used as the independent variable to add the pressure function, and the pressure is applied to the part of the cylinder distribution window, and the variable pressure zone is loaded as shown in Figure 3.

![Figure 3: Schematic diagram of pressure in variable pressure zone of cylinder block distribution window](image)

### 2.2. Finite element simulation results and analysis

From Figure 4, it can be seen that the equivalent stress distribution of the internal and external row is basically the same, both appear in the high-pressure area, where the stress at the distribution window is more obvious, and the stress closer to the cylinder orifice the smaller the stress. The maximum stress appears at the intersection of the external distribution window and the plunger hole, where the structural shape of the sudden change, generating stress concentration, the maximum equivalent force is about 213.48MPa.

![Figure 4: Equivalent stress nephogram of the cylinder block at the inlet pressure of 31.5MPa](image)

(a)Flow distribution window  (b) Outer wall of the cylinder block

From Figure 5, it can be seen that the cylinder deformation mainly occurs to the high-pressure area against the external wall and the distribution window area, the maximum deformation is about 0.011mm.

![Figure 5: Cylinder deformation](image)
Figure 5. Total deformation nephogram of the cylinder block at the inlet pressure of 31.5MPa

Figure 6(a)–(d) is the maximum deformation of the cylinder block hole, the maximum deformation at the bottom of the cylinder block hole, and then gradually decreasing until the cylinder block hole slightly radial shrinkage, but because the wall thickness of the internal row is greater than the external row wall thickness, so the external row hole deformation is greater than the internal row. The external row radial deformation maximum elongation 0.0056mm, minimum shortening 0.0018mm, the internal row radial deformation maximum elongation 0.0026mm, minimum shortening 0.0005mm; external row axial deformation maximum elongation 0.0047mm, minimum elongation 0.0024mm, internal row axial deformation maximum elongation 0.0025mm, minimum elongation 0.00005mm.

Figure 6. The Maximum total deformation nephogram of cylinder block hole at the inlet pressure of 31.5Mpa
2.3. Effect of oil inlet pressure on cylinder stress-strain

As can be seen from Figure 7, with the increase of the inlet pressure, the maximum equivalent stress and strain of the cylinder block are increased approximately linearly, and the growth rate fluctuation is small, but the maximum stress and strain position are little changed.

As shown in Figures 8 and 9, the maximum radial displacement of the internal and external cylinder block hole increases with the increase of inlet pressure, and the maximum radial displacement position is at the bottom of the cylinder block hole. The maximum deformation of the internal and external row cylinder block hole in the orifice position is a slight radial contraction, until the maximum deformation of the cylinder block hole near the bottom of the plunger stay cylinder is a radial increase.

From Table 1, it is obtained that the maximum radial strain of the external cylinder block hole is always larger than that of the internal cylinder block hole, with the increase of the inlet pressure, the internal and external cylinder block hole radial strain increases approximately linearly, but the growth rate of the external cylinder block hole is higher than that of the internal cylinder, because the external wall is thinner and easier to strain than that of the internal cylinder block hole.

| Oil inlet pressures (MPa) | 22.5 | 27.5 | 31.5 | 35.5 | 39.5 | 43.5 |
|---------------------------|------|------|------|------|------|------|
| The maximum radial strain of internal row (mm) | 0.0041 | 0.0049 | 0.0056 | 0.0064 | 0.0072 | 0.0079 |
| The maximum radial strain of external row (mm) | 0.0018 | 0.0022 | 0.0026 | 0.0029 | 0.0032 | 0.0035 |
Figure 10 shows that as the inlet pressure increases, the maximum axial strain of the internal and external cylinder block hole increases linearly, and the value and growth rate of the axial strain of the external cylinder block hole is always larger than that of the internal row.

![Figure 10. Relation between oil inlet pressure and the maximum axial strain of cylinder block hole](image)

3. Fatigue characterization

3.1. Fatigue analysis modelling

The main factors considered in the correction of the material S-N curve[8-9] are the effective stress concentration factor $K_\sigma$, the size factor $C_\sigma$ and the surface quality factor of the component $B_\sigma$. The final correction factor is:

$$X_e = K_\sigma C_\sigma B_\sigma$$  \hspace{1cm} (1)

The effective stress concentration factor is 1.1 and the size factor is 0.64 as shown in Table 2. The surface of the cylinder block is shot-peened and the surface quality factor is 1.3. The S-N correction factor is 0.887 calculated from equation (1).

| Diameter/(mm) | size factor |
|---------------|-------------|
|               | Carbon Steel | Alloy Steel |
| 60–70         | 0.78         | 0.68        |
| 70–80         | 0.75         | 0.66        |
| 80–100        | 0.73         | 0.64        |

By using the modified cylinder S-N curve, the time-load spectrum was generated in the finite element analysis software.

3.2. Fatigue simulation results and analysis

As shown in Figures 11 and 12, the fatigue damage mainly occurred at the stress concentration with a maximum damage value of $1.6 \times 10^{-9}$ at node N130255. The minimum life at the distribution window damaged severely is $6.248 \times 10^8$ cycles and the average is $7.904 \times 10^{11}$ cycles. There is a structural mutation at the intersection of the distribution window and the hole of the cylinder, which generates stress concentration, and the stress and strain are maxima here, so the fatigue damage at the structural mutation is larger and the corresponding fatigue life is smaller.
As shown in Figures 13 and 14, cylinder block hole fatigue damage mainly occurs in the hole filled with high-pressure oil, hole fatigue damage range of about $2.121 \times 10^{-13} - 1.265 \times 10^{-12}$, fatigue life range of about $4.714 \times 10^{12}$ cycles. Each cycle of the hydraulic motor plunger is inhaled high-pressure oil into the cylinder block hole, in the hole to produce large stress, and therefore more vulnerable to fatigue damage.

As Figure 15 shows, with the increase of oil pressure, the minimum fatigue life of the cylinder is nonlinearly decreasing trend, at low pressure the minimum fatigue life decreasing rate is larger, to high pressure the fatigue life decreasing rate is slowed down. As the inlet pressure increases, the maximum fatigue damage of the cylinder increases nonlinearly, and the minimum fatigue damage increases at a smaller rate at low pressure, and the fatigue life increases at a higher rate at high pressure.
4. Conclusion

Based on the finite element mechanical analysis and fatigue characteristics analysis of the cylinder block of the axial piston motor with dual-driving, the stress-strain and fatigue damage and life of the cylinder block were studied, and the influence of the inlet pressure on the above characteristics of the cylinder block was discussed, and the following conclusions were made:

(1) The main location of stress-strain in the cylinder block of the axial piston motor with dual-driving is at the cylinder block hole and the distribution window, and the stress concentration is generated at the structural abrupt change with the largest stress-strain. In the cylinder block hole, the deeper the cylinder block hole the greater the radial strain. The strain value of the external cylinder block hole is always larger than the strain of the internal cylinder block hole. With the increase of the inlet pressure, the stress-strain of the cylinder block is increased, but the major deformation position did not change much.

(2) The fatigue damage of the cylinder block of the axial piston motor with dual-driving mainly occurs at the location of higher stress. With the increase of the inlet pressure, the maximum fatigue damage of the cylinder block nonlinearly increases, while the minimum fatigue life nonlinearly decreases. The reasonable set of the cylinder block hole distribution circle size, change the cylinder wall thickness, reduce the structure of the abrupt change will effectively reduce the cylinder stress-strain and fatigue damage, to increase the life of the cylinder.

Acknowledgments

Project (GXXT-2020-061) supported by the University Synergy Innovation Program of Anhui Province; Project (GXXT-2019-048) supported by the University Synergy Innovation Program of Anhui Province; Project (gxbjZD11) supported by the Top-Notch Talent Program of University (Profession) in Anhui Province.

References

[1] Deng H, Wang L, Guo Y, Zhang Y, Wang C and Shadloo M 2020 *Math. Probl. Eng.* **2020** 1-14
[2] Bachschmid N, Pennacchi P, Tanzi E, Verrier P, Hasnaoui F and Aabadi K 2004 *Int. J. Rotating. mach.* **10** 121-33
[3] Wang Y, Wang X, Guo S, Lu Y and Liu S 2019 *J. Beijing Univ. Aeronaut. Astronaut.* **45** 1314-21
[4] Rahman M and Ariffin A 2006 *J. Zhejiang Univ., Sci.* **2006** 352-60
[5] Lu L, Yu Y and Zeng G 2017 *Journal of Mechanical Transmission.* **41** 114-9
[6] Xu B and Guo S 2008 *J. Chongqing Univ., Technl. Nat. Sci.* **22** 14-6
[7] Yang H, Wang B and Zhou H 2010 *J. Zhejiang Univ., Eng. Sci.* **44** 976-81
[8] Li H, Song G and Liu Y 2002 *J. Zhengzhou Univ., Eng. Sci.* **23** 26-9
[9] Fatemi A and Yang Y 1998 *Int. J. Fatigue.* **20** 9-34