



Article

Influence of Piston Elastic Deformation and Structure Design on the Lubrication Characteristics of Piston Pair: Simulation Analysis

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Abstract

Piston pair is a key friction pair of the axial piston pump, but the influence of elastic deformation and the structure design method is not clear. To reveal the real performance of piston pair, a new fluid–solid coupling calculation method is proposed. With the method, the oil film pressure and thickness field, elastic deformation, axial viscous friction and leakage of the piston pair are studied. The influences of the elastic deformation of the piston pair on oil film pressure, axial viscous friction, and leakage were revealed. To reduce the impact brought by deformation, a new piston with hollow piston structure (piston B) is designed. Compared with the traditional structure (piston A), piston B is featured with small elastic deformation, small leakage, large peak pressure, and large viscous friction force. The new fluid–solid coupling calculation method and hollow piston structure of this paper lay the foundation for the piston pair design of the axial piston pump.

Keywords: piston pair; axial piston pump; fluid-solid coupling; elastic deformation; structure design



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1. Introduction

The friction pair plays the role of lubrication, load bearing, and sealing in the working process of the axial piston pump [1,2]. Therefore, the lubrication state of the friction pair is a decisive factor which affects the overall performance of the axial piston pump. Among the major friction pairs, the piston pair involves a variety of physical field coupling effects, and the lubrication mechanism of which is relatively complex. Therefore, the research on the lubrication characteristics of the piston pair is crucial for the development of the axial piston pump.

At home and abroad, many experts and scholars have carried out relevant research on the lubrication of the piston pair. Monika et al. developed a simulation calculation tool CASPAR for the piston pair of oil film. Through this platform, the pressure field of oil film was analyzed [3]. Kumar et al. explored the effect of the number and position of grooves on the characteristics of the piston oil film through CFD calculation method [4]. Shang et al. developed a design approach for a temperature-adaptive piston [5]. Nie et al. studied the fluid–structure interaction of the piston pair in the deep sea, and discussed the deformation of the piston bushing and the energy loss characteristics of the water film under different working conditions [6]. Ma et al. studied the impact of the piston

structure on the lubrication performance of the piston pair and explored the influence of the length of the piston [7] and the texture of the piston surface [8] on lubrication performance. Wang et al. proposed a multi-physics model to study the influence of oil temperature, load pressure, and rotational speed on the lubrication performance of the piston pair [9], and built a test bench to measure the thickness and pressure of the oil film [10]. Jiang et al. established a simulation tool that can predict the lubrication flow in the piston pair gap, and studied the influence of the thickness of the piston bushing on the lubrication characteristics [11]. In addition, Hu et al. [12,13] conducted research and analysis on the solid–liquid coupling effect of the piston pair and the dynamic characteristics of the piston sliding shoe assembly; Lin et al. established a piston/cylinder pair dynamic model, and proposed a wear prediction method based on the dynamic and wear model [14]. Zhang et al. developed a fluid thermal structure coupling model for piston pair, and studied the influence of key system factors on the lubrication of piston pair [9]. Hu et al. [15], Li et al. [16], Qian et al. [17], and others studied the leakage of the piston pair; Lin et al. [18], Hang et al. [19], and others studied the friction and wear-related issues of the piston pair; Wang et al. [20], Yin et al. [21], Yu et al. [22], Zhao et al. [23], and others studied the temperature distribution of the piston auxiliary oil film.

Even many researchers have considered the elastic deformation of the piston pair when analyzing the lubrication characteristics of the piston pair; the current research is limited to the relevant parameters such as the surface texture and length of the piston. There are few related studies on the influence of the overall structure change in the piston on the lubrication characteristics. Therefore, this paper studies the effect of elastic deformation and overall structure change in the piston through simulation and establishes a theoretical framework for the design of the piston pair.

2. Mathematical Model

2.1. Oil Film Thickness Model of Piston Pair

The piston will be tilted in the piston hole due to external force, and its posture is shown in Figure 1.

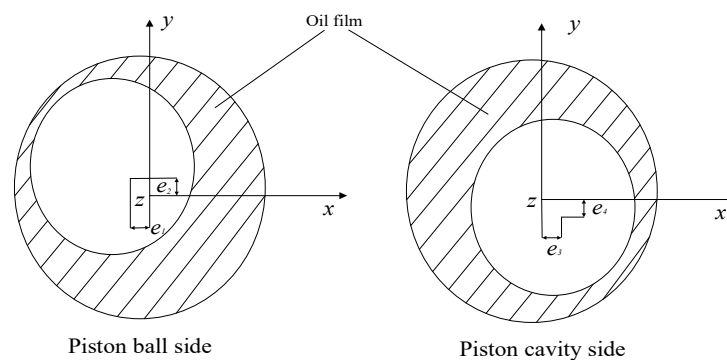


Figure 1. Schematic diagram of piston posture.

The offset of the piston in the cylinder bore is described by four parameters e_1, e_2, e_3, e_4 , where parameters e_1, e_2 are the offset distances in the horizontal and vertical directions of the center of the piston cross-section near the ball end, respectively; parameters e_3, e_4 are the offset distances of the piston cross-section center near the end of the valve plate, respectively [24]. By expanding the oil film into a plane, the thickness of the oil film at any point of the piston pair can be deduced based on the geometric relationship:

$$h = r_c - r_p - \left[e_2 - \frac{l_z(e_2 - e_4)}{l_{cz}} \right] \sin \varphi - \left[e_1 - \frac{l_z(e_1 - e_3)}{l_{cz}} \right] \cos \varphi \quad (1)$$

In Formula (1), h is the oil film thickness; e_1, e_2, e_3, e_4 are the piston offset; l_{cz} is the contact length of the piston pair; l_z is the distance from any position of the piston pair to the end surface of the piston pair close to the ball head; r_c is the inner diameter of the piston hole; r_p is the radius of the piston; φ is the rotation angle of the main shaft.

2.2. Oil Film Pressure Model of Piston Pair

The flow of oil film in the piston auxiliary gap is laminar flow [25]. Assuming that the piston auxiliary oil is an incompressible fluid, ignoring the mass force, flow velocity, and pressure change in the thickness direction of the oil film, the secondary oil film pressure equation is [26] as follows:

$$\frac{\partial}{\partial x} \left(\frac{h^3}{\mu} \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial y} \left(\frac{h^3}{\mu} \frac{\partial p}{\partial y} \right) = 6 \left(\omega r_p \frac{\partial h}{\partial x} + v_p \frac{\partial h}{\partial y} \right) + 12 \frac{\partial h}{\partial t} \quad (2)$$

In Formula (2), p is the piston auxiliary oil film pressure; ω is the rotational angular velocity of the piston; v_p is the movement speed of the piston along the axial direction of the piston; μ is the dynamic viscosity of the hydraulic oil.

2.3. Oil Film Pressure Equation Solution

For solving the pressure equation of the piston auxiliary oil film, the finite volume method is used in this paper, and the expanded piston auxiliary oil film is divided into staggered grids, as shown in Figure 2. Among them, the triangle nodes are the node distribution when solving the pressure field, and the triangle nodes and circular nodes are the node distribution for solving the thickness field of the oil film.

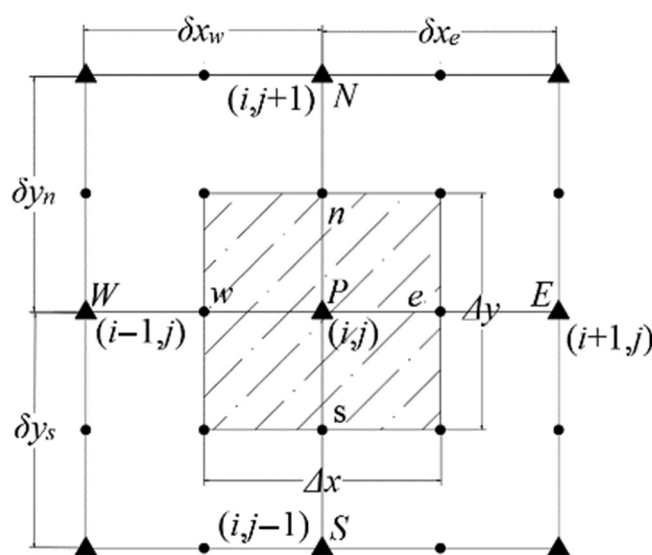


Figure 2. Mesh division of piston auxiliary oil film.

For the control body shown in the shaded part in Figure 2, Equation (2) is integrated, and the following formula can be obtained:

$$\alpha_P P_P = \alpha_E P_E + \alpha_W P_W + \alpha_N P_N + \alpha_S P_S + S \quad (3)$$

The expressions of each coefficient in Equation (3) are as follows:

$$\alpha_E = \frac{h_e^3}{\mu} \frac{\delta y}{\delta x} \quad (4)$$

$$\alpha_W = \frac{h_w^3}{\mu} \frac{\delta y}{\delta x} \quad (5)$$

$$\alpha_N = \frac{h_n^3}{\mu} \frac{\delta x}{\delta y} \quad (6)$$

$$h = r_c - r_p - \left[e_2 - \frac{l_z(e_2 - e_4)}{l_{cz}} \right] \sin \varphi - \left[e_1 - \frac{l_z(e_1 - e_3)}{l_{cz}} \right] \cos \varphi \quad (7)$$

$$\alpha_P = \alpha_E + \alpha_W + \alpha_N + \alpha_S \quad (8)$$

$$S = -6 \left[\omega r_p (h_e - h_w) \delta y - v_p (h_n - h_s) \delta x \right] - 12 \int_w^e \int_s^n \frac{\partial h}{\partial t} dx dy \quad (9)$$

This paper uses the SOR iterative method to iteratively solve Equation (3), and the iterative format is

$$P_P^{(k+1)} = P_P^{(k)} + \varepsilon \left(\frac{\alpha_E^{(k)} P_E^{(k)} + \alpha_W^{(k)} P_W^{(k)} + \alpha_N^{(k)} P_N^{(k)} + \alpha_S^{(k)} P_S^{(k)} + S}{\alpha_P^{(k)}} - P_P^{(k)} \right) \quad (10)$$

Calculation accuracy meets as follows:

$$\frac{\sum_{i=2}^{m-1} \sum_{j=1}^n |P_{ij}^{(k)} - P_{ij}^{(k-1)}|}{\sum_{i=2}^{m-1} \sum_{j=1}^n |P_{ij}^{(k)}|} \leq error = 0.001 \quad (11)$$

The relaxation factor ε in Equation (10) is set to 1.7 [27]. The boundary conditions for pressure calculation are as follows:

- (1) When $y = 0$, the oil film pressure is equal to the oil pressure in the piston chamber; when $y = l_z$, the oil film pressure is equal to the oil pressure in the casing.
- (2) When $\theta = 0^\circ$ and $\theta = 360^\circ$, the oil film pressure is the same, and θ is the angle in the circumferential direction.

2.4. Geometric Model Construction and Mesh Independence Verification

The influence of the elastic deformation of the piston pair on the oil film is considered. A geometric model of the piston pair is established, and the finite element analysis is performed in Abaqus 2020 software. The oil film thickness is calculated based on Matlab 2020b software, and the pressure load on the piston surface is derived based on the radial spring function in the Abaqus software. Data transfer between the Abaqus software and Matlab software is realized through Python 3.2. The established geometric model is shown in Figure 3. The piston is made of 42CrMo material, with a Young's modulus of 210 GPa and a Poisson's ratio of 0.3. The bushing is set to be made of CuZn30 material, with a Young's modulus of 100 GPa and a Poisson's ratio of 0.34. In order to simplify the calculation, one piston pair model was established, and the cylinder model includes two adjacent cylinder holes of the simulated piston pair. After simulation calculation verification, it is found that when the load is applied to the outer surface of the piston and the inner surface of the bushing body in the gap of the piston pair, the deformation is very small and can be ignored after passing through the left and right cylinder holes of the cylinder body. Therefore, the intercepted cylinder model is feasible. During the mesh division of the piston pair, the piston and bushing adopt the hexahedral mesh, while the cylinder part adopts the tetrahedral mesh. The mesh division of the piston and the bushing is shown in Figure 4.

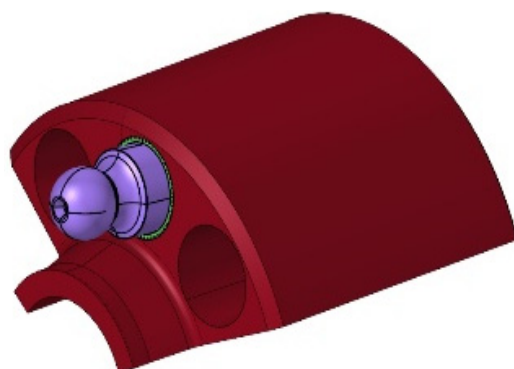


Figure 3. Geometric model of piston pair.

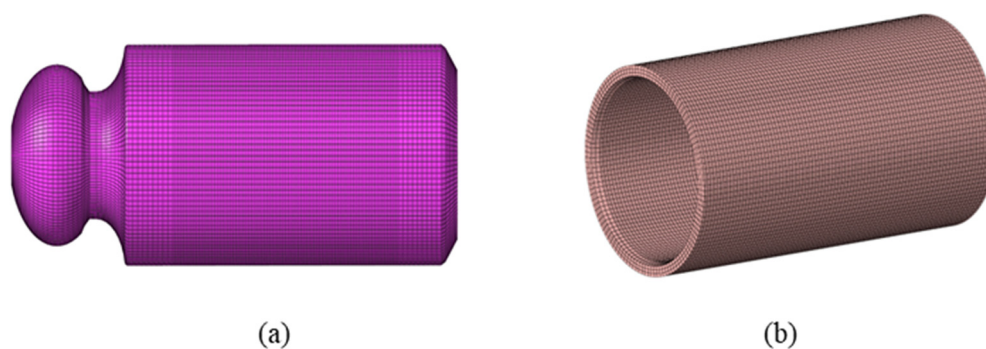


Figure 4. Meshing of (a) piston body and (b) bushing body.

To select the suitable number of grids, the piston pair was divided into grids of different sizes, and the designed grid numbers were 212,625, 249,645, 288,145, 375,515, and 419,285. For the above three-dimensional models with different mesh numbers, a pressure load of 40 MPa was applied to the surface of the piston pair, and the other constraints and loads remained the same. The elastic deformation is shown in Figure 5.

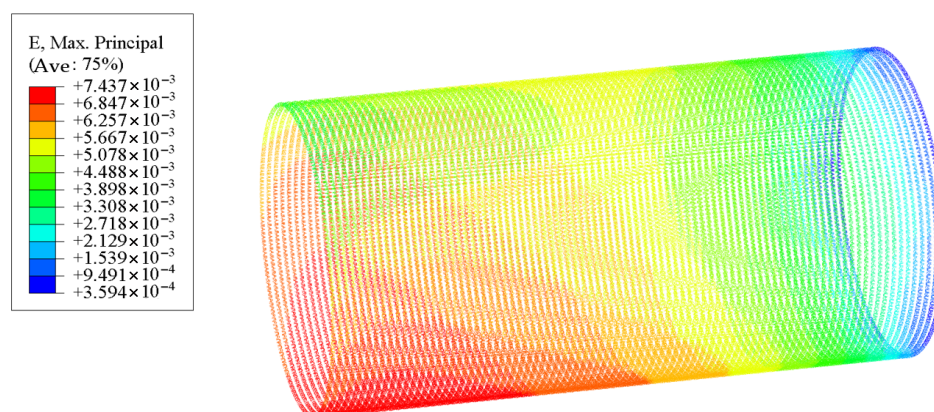


Figure 5. Elastic deformation of the piston pair.

The peak elastic deformation of the piston pair under different grid numbers is shown in Table 1. According to the results, the calculation accuracy is satisfactory within the selected range of grid numbers. The result of 212,625 meshes is a reference for other calculations; hence, there is no calculation error for 212,625 meshes. When the grid number increases, the calculation error of the peak elastic deformation of the piston pair gradually decreases, but the errors are all lower than 1.7%. Therefore, it can be considered that within the range of the selected grid number, the calculation accuracy meets the requirements.

Considering the calculation accuracy and time cost, the number of nodes in the oil film thickness grid is selected as 288,145 in this paper.

Table 1. Mesh quantity and calculation error.

Number of Volume Meshes	Maximum Deformation/ μm	Calculation Error
212,625	7.2011	-
249,645	7.3148	1.58%
288,145	7.4367	1.67%
375,515	7.5183	1.1%
419,285	7.5869	0.91%

Figure 6 shows the simulation calculation process. Initially, the thickness and pressure of the piston pair oil film are solved by inputting the relevant initial parameters. With the Abaqus simulation software, the deformation of the piston pair is calculated based on the piston deputy oil film pressure. Then, finite element analysis corrects the thickness and pressure of the oil film according to the elastic deformation of the piston pair. When the relative error of the oil film pressure before and after the update meets the convergence standard, the oil film pressure and thickness distribution calculated by the simulation is the output. According to the oil film pressure and thickness distribution, the oil film force is further calculated and the force balance equation is analyzed. If the force balance equation is satisfied, enter the calculation at the next moment; otherwise, return to continue the iteration. Finally, the calculated angle is judged. If the starting and ending positions are the same, the calculation of one cycle is completed; otherwise, the calculation of the next angle is continued. This paper focuses on the elastic deformation caused by oil pressure and does not consider the multi-body dynamics modeling of the piston pump, thus not involving inertial forces. Table 2 shows the relevant parameters for the fluid–solid coupling calculation of the piston auxiliary oil film. In this paper, the angle between the piston of the pump and the shaft is designed to be 5 degrees. Hence, the piston angle was set as 5° .

Table 2. Relevant parameters for oil film simulation calculation.

Parameter Setting	Parameter Values
Piston diameter d_p (mm)	$45 - 0.087$
Piston hole diameter d_c (mm)	$45 + 0.005$
Piston quantity n_p	9
Speed n (rev/min)	1600
Cradle angle β ($^\circ$)	15
Piston angle γ ($^\circ$)	5
High pressure of piston cavity PPH (MPa)	40
Low pressure of piston cavity PPL (MPa)	2
Case pressure PC (bar)	0.5
Weight of piston group (kg)	1.74

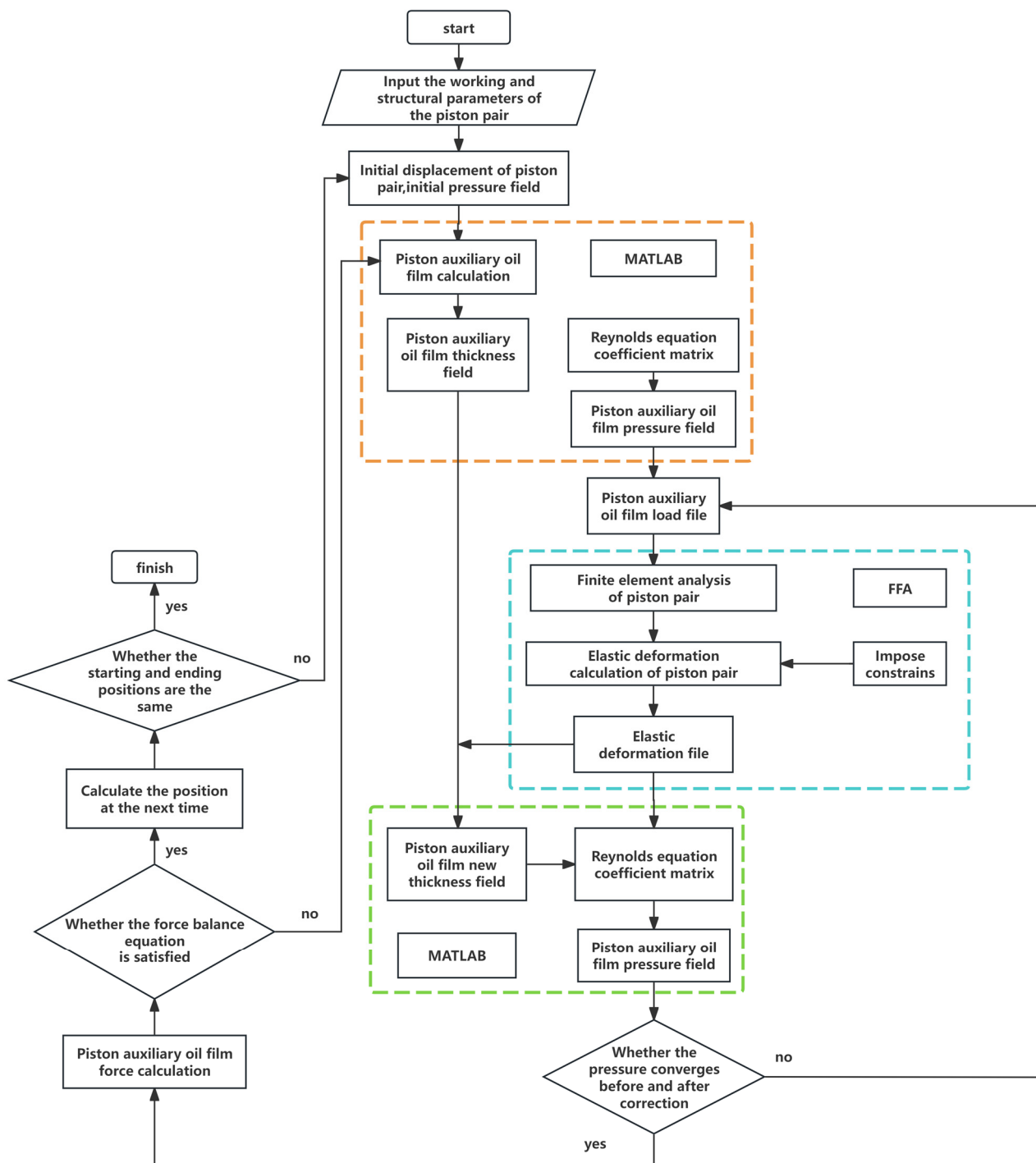


Figure 6. Simulation calculation process.

3. Simulation

3.1. Influence of Elastic Deformation on the Working Characteristics of Piston Pair

In order to explore the influence of the elastic deformation of the piston pair on its lubrication characteristics, two sets of calculations were carried out on the oil film. In the first group of simulation experiments, the piston pair was assumed to be a rigid to study the influence of the oil film on the piston pair; in the second set of simulation experiments, the influence of the elastic deformation of the piston pair was considered while calculating of the oil film.

3.1.1. Field Distribution of Oil Film Thickness in Piston Pair

When the main shaft rotation angle is 90° , Figure 7 reflects the distribution of the piston auxiliary oil film thickness obtained by two sets of simulation calculations. The axial length L_a is the contact length between the piston and the piston bushing. The starting point of the axial length is the end of the piston cavity, and the end point is the end of the piston housing. L_c is the circumferential length of the auxiliary oil film of the piston after it expands in the circumferential direction. It can be seen that the elastic deformation of the piston pair does not change the oil film thickness distribution. Because of the eccentric load of the piston, the distribution of the oil film thickness of the piston pair is not uniform, and small-thickness oil film area appears. By comparing the thickness of the oil film in Figure 7a,b, it can be seen that the elastic deformation of the piston pair increases the oil film thickness. Moreover, when the elastic deformation of the piston pair is taken into account, the thickness of small-thickness oil film in the purple area in Figure 7b increases, which improves the lubrication of the piston pair.

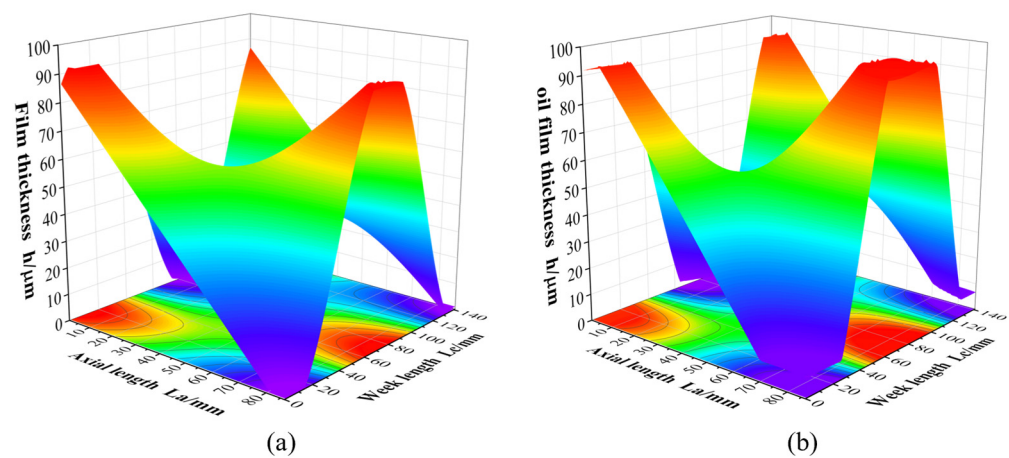


Figure 7. Oil film thickness distribution of the piston pair (a) without considering elastic deformation and (b) considering elastic deformation.

3.1.2. Pressure Field Distribution of Piston Auxiliary Oil Film

When the main shaft rotation angle is 90° , Figure 8 reflects the pressure distribution of the piston auxiliary oil film obtained by the two sets of simulations. It can be seen that the elastic deformation of the piston pair will affect the pressure peak of the piston pair oil film.

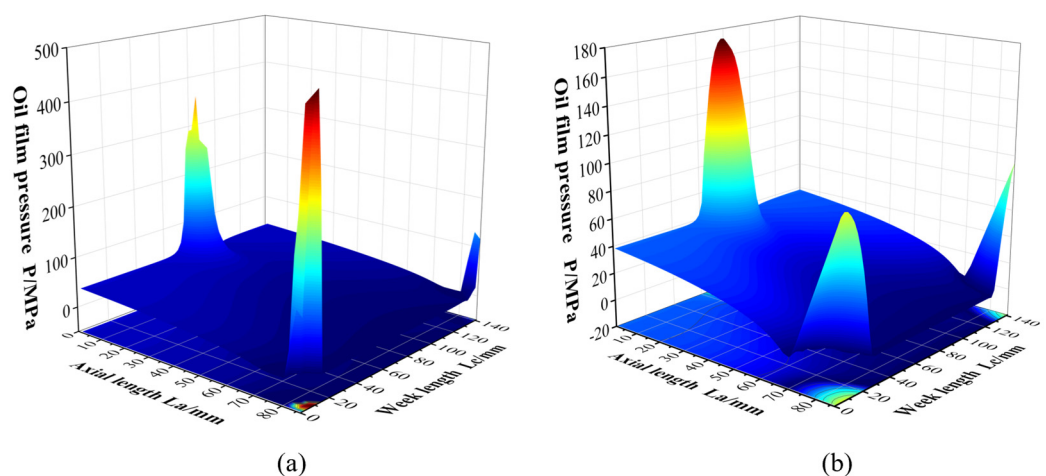


Figure 8. Distribution of oil film pressure in piston pair (a) without considering elastic deformation and (b) considering elastic deformation.

Due to the eccentric load of the piston, the pressure peak reaches 490 MPa in the auxiliary oil film of the piston at both the end of the piston chamber and the end of the sliding shoe. The peak oil film pressure drops to 172 MPa if the elastic deformation is considered. When the elastic deformation is not considered, the peak pressure of the oil film appears at the end of the sliding shoe. In contrast, the peak pressure of the oil film appears at the end of the piston cavity when the elastic deformation is considered. Therefore, the elastic deformation has a greater effect on reducing the peak pressure of the oil film.

Figure 9 reflects the effect of the elastic deformation of the piston pair on the peak oil film pressure at different spindle angles. As seen from the figure, after considering the elastic deformation, the peak oil film pressure of the piston pair decreases significantly in both the oil-discharging and oil-absorbing zones. The maximum decrease in peak oil film pressure is 950.1 MPa, and the average decrease is 390.9 MPa in the oil discharge area. The maximum decrease in peak oil film pressure is 153.3 MPa, and the average decrease is 68.6 MPa in the oil suction area. It can be seen that the elastic deformation of the piston pair has a significant effect on the oil film pressure of the piston pair, and the peak oil film pressure decreases significantly when considering the elastic deformation of the piston pair.

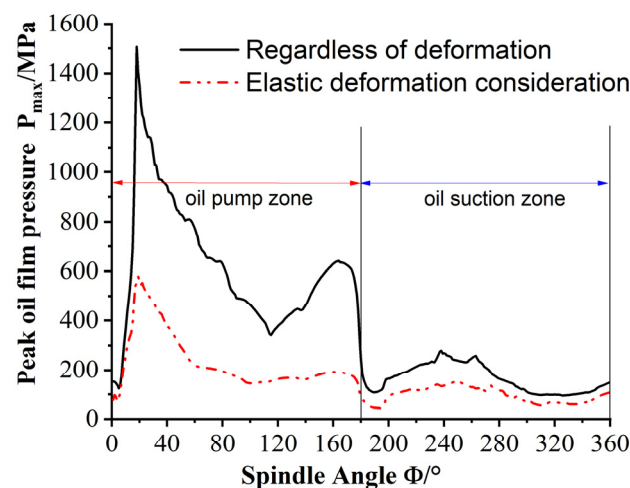


Figure 9. Effect of elastic deformation on peak pressure of oil film.

3.1.3. Distribution of Elastic Deformation of the Piston Pair

The distribution of elastic deformation of the piston sub at a spindle rotation angle of 90° is shown in Figure 10. It can be seen that the deformation of the piston pair is not uniformly distributed, and the piston pair deforms high at both ends and low in the middle in the circumferential direction, which is due to the pressure spikes of the oil film at both ends of the circumferential direction. It is easier to produce larger deformation in the area near the end of the sliding shoe, and it is analyzed that this situation occurs because the thickness of the cylinder body at the end of the sliding shoe is smaller, and the overall stiffness is lower than that at the end of the piston cavity.

The peak elastic deformation of the piston pair in an operation cycle of the axial piston pump is shown in Figure 11. As seen from the figure, the change in the peak value of the piston pair is similar to that of the oil pressure at the end of the piston cavity, showing the change rule of fluctuating up and down in the oil discharge area, while remaining relatively stable in the oil suction area. The peak deformation of the piston sub in the oil discharge area reaches $5.94 \mu\text{m}$ on average, while that in the oil suction area is $0.71 \mu\text{m}$. The average piston sub gap of the axial piston pump studied in this paper is $46 \mu\text{m}$, and the average value of the maximum deformation of the piston sub in the oil discharge area occupies 12.9% of the piston sub gap, while the average value of the maximum deformation of the piston sub in the oil suction area occupies 1.5% of the piston sub gap. It can be seen that

the elastic deformation of the piston pair significantly affects the oil film thickness and pressure of the piston pair.

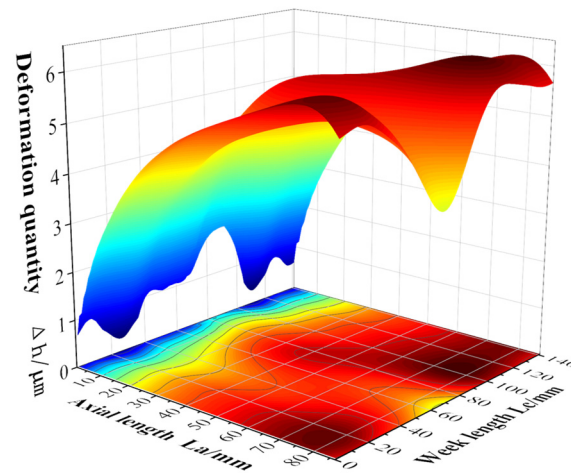


Figure 10. Distribution of elastic deformation on the surface of the piston pair.

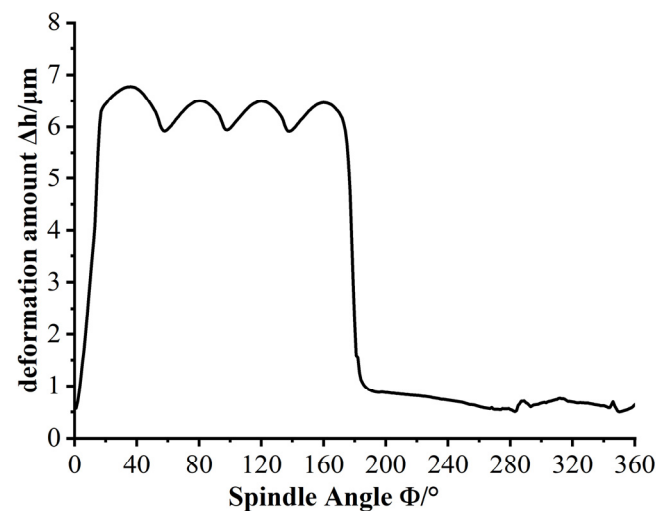


Figure 11. Maximum elastic deformation curve of piston pair.

3.1.4. Axial Viscous Friction Analysis

The changes in the axial viscous friction force on the piston are obtained by two groups of calculations, as shown in Figure 12.

It can be seen that the elastic deformation does not change the variation trend of axial viscous friction with the spindle angle. After considering the elastic deformation, the axial viscous friction decreases significantly in the oil discharge area, with the peak value decreasing from 183.6 N to 163.2 N (an 11.1% decrease), while no significant change occurs in the oil suction area. The reason for this change is that the elastic deformation of the piston pair increases the thickness of the oil film, which leads to the reduction in the oil velocity gradient, and further leads to the reduction in axial viscous friction. The change mainly occurs in the oil discharge area because the elastic deformation mainly occurs in the oil discharge area. In contrast, the deformation in the oil suction area is small, which has little impact on the thickness of the oil film.

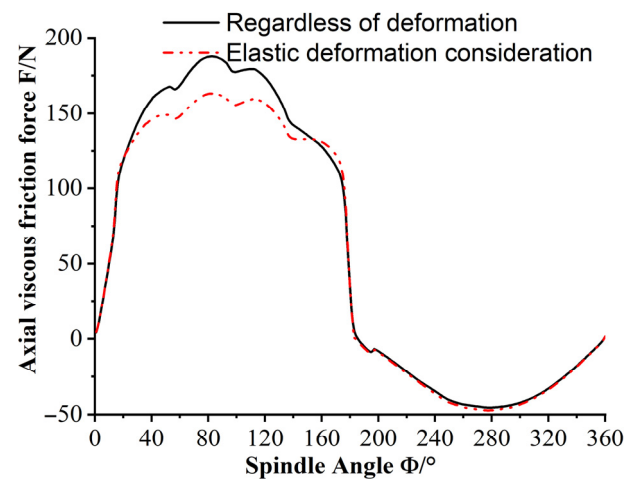


Figure 12. Effect of elastic deformation on axial viscous friction force.

3.1.5. Piston Pair Leakage Analysis

Figure 13 reflects the changes in the leakage of the piston pair. The elastic deformation of the piston pair does not affect the change law of the leakage rate with the main shaft angle, but affects the leakage rate. While the elastic deformation of the piston pair is considered, the leakage of the piston pair in the oil discharge area increases, and the peak leakage increases from 1.47 L/min to 1.73 L/min, with a change rate of 17.7%, while the leakage does not change significantly in the oil suction area. This difference can be explained by the fact that, in the oil discharge area, the piston pair undergoes large elastic deformation, and the oil film thickness increases, resulting in a substantial increase in leakage; while in the oil suction area, the elastic deformation is small, so the change in the oil film thickness is not obvious. Therefore, the leakage change is not significant.

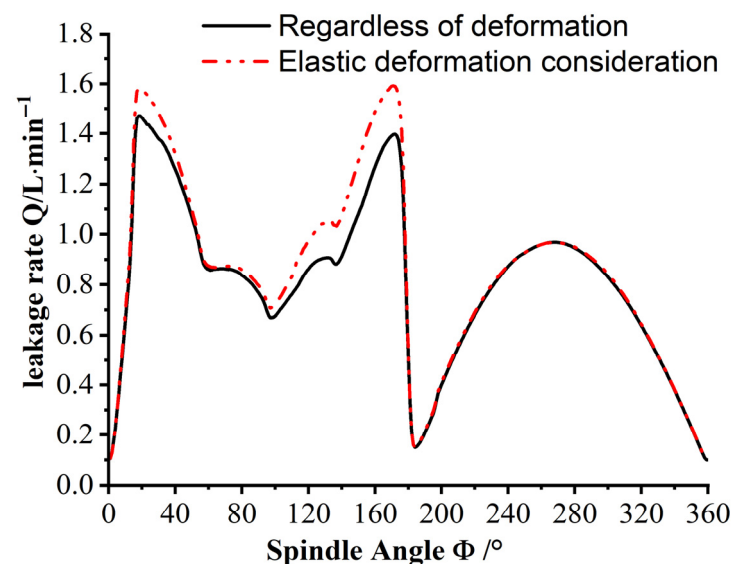


Figure 13. Effect of elastic deformation on the leakage of the piston pair.

3.2. Influence of Piston Structure on the Working Characteristics of Piston Pair

To reduce the elastic deformation caused by the oil pressure and lower the gap leakage, this paper proposes a new hollow piston structure (piston B). The manufacturing process of the new hollow piston structure includes friction welding, rough turning, fine turning, rough grinding, nitriding, and fine grinding. In order to study the influence of the piston structure on the lubrication characteristics of the piston pair, this paper conducts simulation

and analysis for the piston with two structures. The two piston structures are shown in Figure 14. Figure 14a shows the solid piston structure used in the calculation in Section 3.1 of this paper, while Figure 14b shows the new hollow piston structure. Hereinafter, the piston shown in Figure 14a is referred to as piston A; the piston shown in Figure 14b is referred to as piston B. The relevant calculation parameters are shown in Table 3.

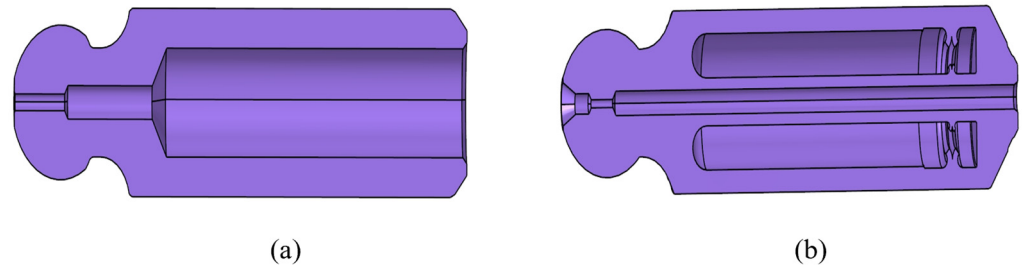


Figure 14. Schematic diagram of (a) solid piston structure (piston A) and (b) hollow piston structure (piston B).

Table 3. Calculation parameters for different pistons.

Parameter	Value
Cradle angle/ β ($^{\circ}$)	15
Piston angle/ γ ($^{\circ}$)	5
Speed/ n (r/min)	1000
Load pressure/ P_p (MPa)	35

3.2.1. Effect of Piston Structure on Oil Film Pressure

Figure 15 shows the pressure distribution of the auxiliary oil film of the two kinds of pistons when the main shaft rotation angle is 90° .

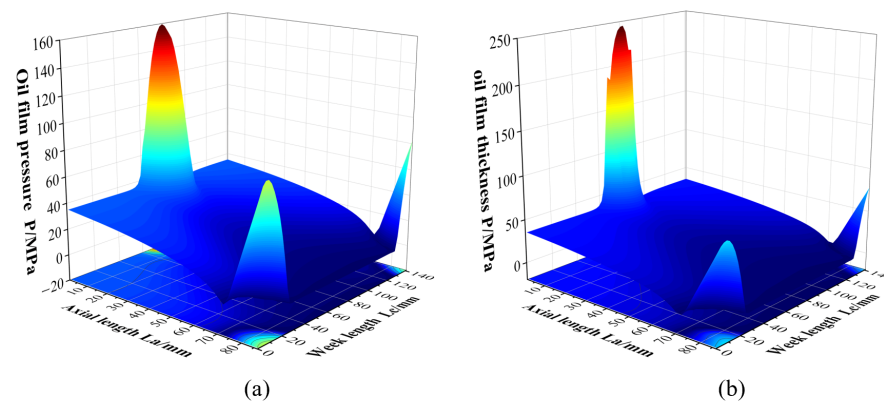


Figure 15. Oil film pressure distribution of (a) piston A and (b) piston B.

It can be seen that changes in the piston structure will not change the pattern of the pressure distribution, but cause changes in the pressure. The oil film has pressure peaks at both the piston chamber end and the sliding shoe end. Compared with piston A, the peak pressure of the piston auxiliary oil film of piston B increases significantly. The peak oil film pressure of piston A increases from 159 MPa to 244 MPa, which is an increase of 53.5%. It can be seen that the change in the piston structure will have a significant impact on the pressure of the piston auxiliary oil film.

During one operation cycle, the change in the peak value of the oil film pressure of the piston structures is presented in Figure 16. It can be seen that the change in the piston structure does not affect the variation trend of the peak oil film pressure with the spindle

angle, but it significantly changes the maximum pressure value. In the oil discharge area, compared with piston B, the peak pressure of the auxiliary oil film of piston A decreases by 514.3 MPa at the maximum, with an average decrease of 161.6 MPa; while in the oil suction area, there is no obvious difference in the peak value of the oil film between the two. Therefore, the change in the piston structure has a significant impact on the pressure of the oil film, which is mainly reflected in the oil discharge area.

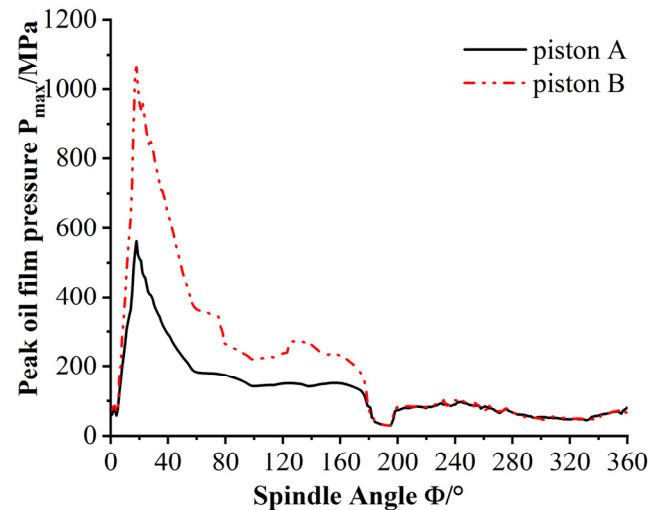


Figure 16. Effect of piston structure on peak oil film pressure.

3.2.2. Effect of Piston Structure on Elastic Deformation

When the main shaft rotation angle is 90° , the distribution of the elastic deformation of the two piston structures are shown in Figure 17. It can be seen that the change in the piston structure has an influence on the distribution of the elastic deformation. Compared with piston A, the elastic deformation of piston B shows a small area of large deformation. However, the distribution of elastic deformation of piston A is more uniform and shows a broad area of large deformation. Therefore, compared with piston A, the structure of piston B increases the structural stiffness of the piston, making the elastic deformation of the piston pair smaller under the same load pressure, but this also leads to higher oil film peak pressure.

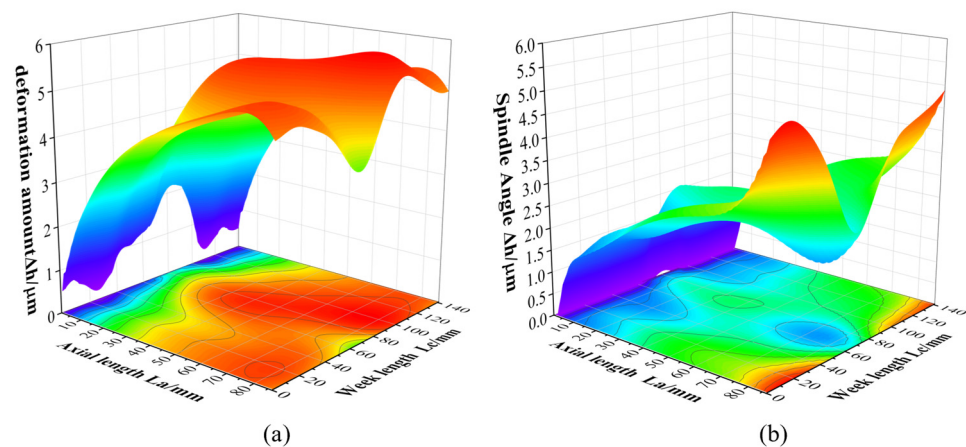


Figure 17. Distribution of elastic deformation of (a) piston A and (b) piston B.

Figure 18 shows the peak value variation in the elastic deformation of the two piston structures in one operation cycle of the axial piston pump. It can be seen that the change in the piston structure leads to a significant change in the maximum deformation of the

piston pair, which mainly happens in the oil discharge area. In the oil discharge area, the average maximum deformation of the piston pair of piston A is $5.47\text{ }\mu\text{m}$, and the average maximum deformation of the piston pair of piston B is $4.86\text{ }\mu\text{m}$. Compared with piston A, the maximum deformation of piston B decreased by 11.2%. In the oil suction area, the average maximum deformation of piston A is $0.6\text{ }\mu\text{m}$, and the average maximum deformation of piston B is $0.54\text{ }\mu\text{m}$, which is 10% less than the former. It can be seen that the change in the piston structure will have an influence on the elastic deformation of the piston pair.

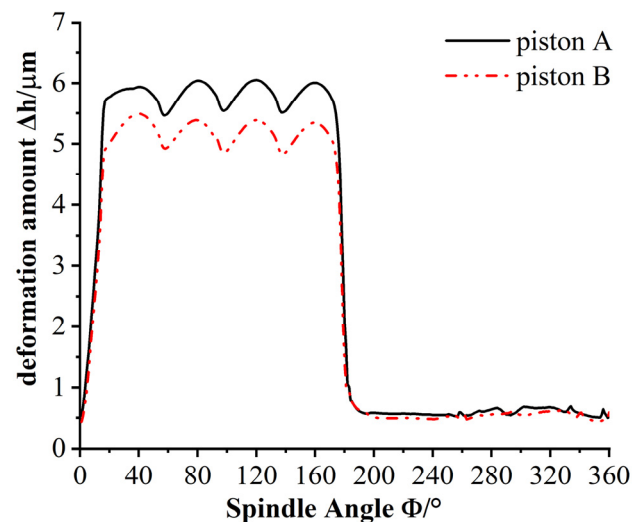


Figure 18. Effect of piston structure on elastic deformation.

3.2.3. Effect of Piston Structure on Axial Viscous Friction Force

Figure 19 reflects the variation in the axial viscous friction force of the two piston structures in one operation cycle of the piston pump.

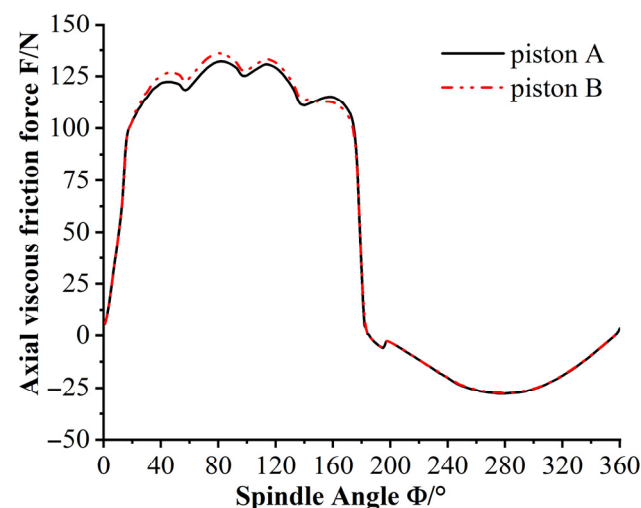


Figure 19. Effect of piston structure on axial viscous friction force.

It can be seen that the change in the piston structure leads to a slight variation in the axial viscous friction force which is mainly concentrated in the oil discharge area. When using piston A, the peak value of the axial viscous friction force is 132.2 N , and the average value is 111.8 N . While using piston B, the peak value of the axial viscous friction force is 136 N and the average value is 113.6 N , which are 2.9% and 1.6% higher than the former. In the oil suction area, when piston A is used, the peak value of the axial viscous friction force

is 27.5 N, and the average value is 15.93 N; when piston B is used, the two are 27.8 N and 15.7 N, which are, respectively, increased by 1.1% and decreased by 1.4%.

3.2.4. Effect of Piston Structure on Leakage

Figure 20 shows the changes in the leakage of the piston pair during one operating cycle of the piston pump when using two piston structures. It can be seen that changes in the piston structure will lead to changes in the leakage of the piston pair, and this change is mainly reflected in the oil discharge area. In the oil discharge area, compared with piston A, the use of piston B will significantly reduce the leakage of the piston pair. When piston A is used, the peak leakage of the piston pair is 1.74 L/min, and the average value is 1.19 L/min; when piston B is used, the two are 1.49 L/min and 1.04 L/min, respectively. Compared with piston A, the peak and average leakage volume of piston B are reduced by 14.4% and 12.6%, respectively. In the oil suction area, the peak and average leakage of piston A are 0.612 L/min and 0.423 L/min, respectively; the peak and average leakage of piston B are 0.608 L/min and 0.417 L/min, respectively, which decreased by 0.65% and 1.4%, respectively. Therefore, the impact of the piston structure on the leakage of the piston pair is mainly reflected in the oil discharge area, and the change is not obvious in the oil suction area.

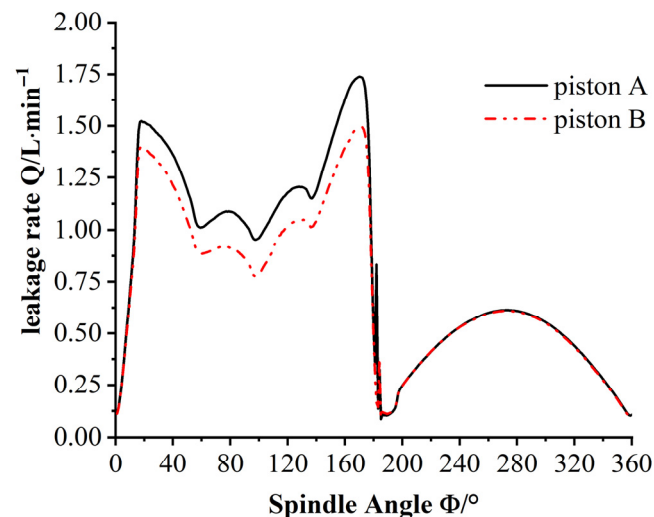


Figure 20. Effect of piston structure on leakage.

4. Conclusions

In this paper, a fluid–solid coupling calculation method is proposed for the oil film of the piston pair of the axial piston pump, and the effects of the elastic deformation and structure design on the working characteristics of the piston pair are analyzed. The conclusions are as follows:

- (1) The piston pair's elastic deformation does not affect the pressure and thickness distribution law of the oil film, nor the variation law of axial viscous friction force and leakage along with the spindle rotation angle, but it will affect the oil film pressure peak.
- (2) The piston pair's elastic deformation mainly occurs in the oil discharge area. While considering the elastic deformation, the peak oil film pressure and the axial viscous friction force decrease, but the leakage of the piston pair increases.
- (3) The change in the piston structure will have a significant impact on the lubrication characteristics of the piston pair, which is mainly reflected in the oil discharge area. Compared with piston A, the higher structural rigidity of piston B leads to a decrease

in the elastic deformation of the piston pair, an increase in the peak pressure of the oil film, a decrease in the leakage of the piston pair, and a small increase in the axial viscous friction force.

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