Research Article

Numerical Study on the Characteristics and Effects of Gap Flow in Two Parallel Rotating Disks

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Received 14 February 2020; Revised 18 April 2020; Accepted 24 April 2020; Published 12 May 2020

Academic Editor: Mohamed Abd El Aziz

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Rotating plane flow distributing pairs have been widely used in engineering. It is very important to reduce the frictional resistance between the pairs to improve the sensitivity and prolong service life. In this paper, the traditional rotating plane flow pair is improved, and the boss is introduced to keep the rotor and the stator in incomplete contact, and then a gap is formed between the flow pairs. The relationship between the pressure distribution and the friction resistance of the flow pairs is established. The theoretical mathematical model of the fluid movement between the gaps of the structure is built. Given the complexity of model analysis, the numerical analysis method is used to analyze the fluid motion in the structure. The height of the boss is proved to be the most important parameter of the structure, the increase of the boss's height causes the pressure between the flow pairs to increase, and the pressure difference between regions gradually decreases, so the friction between the flow pairs will gradually decrease until reaching a stable value. The change of the boss's diameter has little influence on the pressure distribution of the clearance, but as the diameter increases, the friction between the flow pairs will increase linearly. Once the opening width of the waist hole on the upper rotor is smaller than the radius of the discharge hole on the lower stator, the smaller the width, the higher the pressure, but when the opening width is larger than the radius of the discharge holes, the change of the width has no obvious influence on the pressure distribution and the friction.

1. Introduction

Rotating plane flow distributing pairs have been widely used in engineering. It is very important to reduce the frictional resistance between the pairs to improve the sensitivity and prolong service life. There are many kinds of gap flow, just like parallel plane gap, annular gap, annular plane gap, and so on. Immense amounts of concrete research have been done on this subject [1–4]. In [5], the axial annular clearance flow with the same gap width was studied and calculation equation was obtained by carrying out the experiment. The effect of concentric slot flow on the performance of various piston pairs was studied by Yamaguchi [6, 7]. Oil film vortex was proved to exist in the plunger pair through experiments by Tanaka et al. [8]. The leakage problem of plunger pair and others axial annular gap flow was studied by Li et al. [9–13]. But most of the above studies have focused on annular gap

flow. Less research was focused on another kind of gap flow which exists between the parallel planes.

One kind of the structure which is composed of two relatively concentric rotating parallel disks as shown in Figure 1 was studied in this paper. This structure is widely used for flow distribution in directional drilling engineering (such as rotary steerable system) in oil and gas development [14]. The oriented actuator (as shown in Figure 1(a)) is used to push the pads to steer the drilling trajectory. When the disk valve is at a certain tool face angle, the drilling fluid moves the pads out by flowing through valves. Drilling fluid pushes one of the pads to the borehole and produces the steering force that can orient the drilling [15]. Then, it can be used to implement flow allocation when the connect area changes with the rotor rotation. One important problem existing in this structure is that the fluid flowing through it belongs to pressure fluid in most cases, so the force loaded

FIGURE 1: Schematic diagram of plane flow distribution pair structure used in rotary steerable system.

on the structure is often very big, which will result in greater friction between the rotor and the stator and affect the sensitivity of the relative rotation between them.

Many methods have been tried to improve the friction performance of the structure; one important improvement is to maintain a gap between the stator and rotor; this method is used in the plain porting mechanism in axial piston pump; a constant clearance is obtained by use the oil pressure to balance the preload of the spring. Cheng et al. studied the new Poiseuille–Couette flow of edge viscosity fluid in small gaps in parallel planes and analyzed the temperature field, velocity, flow rate, and friction force of this flow [16–19]. Danish et al. [18] obtained the exact solution of the velocity distribution and flow distribution of non-Newtonian fluid between parallel planes [20, 21]. Yang et al. [22] researched and obtained the pressure distribution of the axial piston pump’s plane port pair based on disk-shape gap. But it is usually difficult to ensure that the gap is small, especially when the equipment is large. Once the stator and rotor connect to each other, the friction between them will be much greater than the design value, resulting in the rotor not rotating normally or getting damaged soon by wear. One new structure was designed to solve this question as shown in Figure 2. One small boss is machined on the rotor. If the gap between the rotor and stator is big enough, the pressure applied on plane B is equal to the pressure P loaded on the plane A of the structure; then, the force F_p loaded on the structure can be obtained as

$$F_p = PS, \quad (1)$$

where S refers to the acting area of fluid pressure, which is equal to the area of the boss when the gap between the stator and rotor is big enough. The smaller the size of the boss, the better in theory at this moment. This is because the area S of the contact area C between the stator and the rotor decreases as the diameter of the boss decreases. Under the same pressure distribution, the axial positive force caused by fluid pressure loaded on the stator and the rotor becomes smaller too. However, the diameter of the boss cannot be reduced without limit. When the size of the boss is reduced to zero, the gap between the stator and the rotor needs to be maintained in other ways, and the structural form changes. Given that the boss diameter is not zero, the strength of the boss needs to be considered.

Once the force loaded on the structure F_p is obtained, the friction force F_μ can be obtained:

$$F_\mu = F_p \mu, \quad (2)$$

where μ represents the friction coefficient of the boss of the rotor and the stator; this is usually an experience value or obtained by experiment. Then, the friction torque M_f can be calculated:

$$M_f = F_\mu R, \quad (3)$$

where R represents the equivalent radius of the connect area. The smaller the R , the smaller the M_f , and the boss can be put on the center of the rotor as much as possible as shown in Figure 2.

But it is impossible to take a large gap in the actual use. This is because once the gap is so big, the pressure of the gap will be equal to the pressure of other fluid region, and then the leakage of the gap may render the plane flow pairs useless. The role of the distribution pairs is to open only one or some special flow channels at a time, rather than opening all channels without a significant difference in flow.

So, how to determine the height of boss is very important. The leakage through the gap will increase with the height increase. However, it is important that we want the height of the boss to be as small as possible; the smaller height of the boss is more difficult to be obtained and easier to be damaged by wear, and the pressure in other areas will be balanced by the fluid pressure present in the gap. If there is enough pressure in the gap to balance the pressure loaded on another plane of rotor once the gap is small enough is a question.

In the structure involved in this paper, the stator and the rotor are in incomplete contact, which creates a gap between them, and the fluid distribution law in the gap is affected by factors such as the diameter and height of the contact boss, and it affects the frictional resistance between the stator and the rotor together with the contact boss. In this paper, CFD (computational fluid dynamics) was used to analyze the influence of each parameter of the structure on the slit flow. All factors influencing the pressure distribution were discussed besides the height of the boss, just like the boss diameter, the size of the waist hole in the rotor, and so on. The pressure distribution and the leakage of the fluid in the gap were obtained in different conditions. At the same time, the frictional resistance between each other is also obtained. In-depth analysis of the interaction between the contact boss and the fluid is beneficial for understanding the friction

FIGURE 2: Diagram of the structure of incomplete contact distribution disks and comparison with traditional disks. (a) Structure diagram of the incomplete contact distribution disks. (b) Structure diagram of the traditional distribution disks. (c) Description of the structure of incomplete contact distribution disks.

reduction mechanism of the structure and enriching the gap flow theory.

2. Numerical Simulation Methodology

2.1. Mathematical Model. To build the mathematical model, the structure can be simplified to concentric disk plane pairs, as shown in Figure 3. There are two parts in this model: rotor and stator; the gap between them is h , and their relative speed is W . The fluid around the plane pair is high-pressure liquid. Then, based on the continuity assumption and no-slip boundary conditions, model the fluid between the

microscale of the concentric disk plane pairs. The liquid used around the pair is Newton fluid, and it is assumed to be steady, incompressible, and laminar with no cavitation. The density of the liquid is assumed to be constant. But the viscosity will change with the temperature. The walls are smooth and adiabatic. Then, the fluid between the concentric disk plane pair can be described by governing equations (4)–(9) [21, 23].

2.1.1. Navier–Stokes Equation of Motion

$$u_r \frac{\partial u_r}{\partial r} + \frac{u_\theta}{r} \frac{\partial u_r}{\partial \theta} + u_z \frac{\partial u_r}{\partial z} - \frac{u_\theta^2}{r} = -\frac{1}{r} \frac{\partial p}{\partial r} + 2 \frac{\partial}{\partial r} \left(\eta \frac{\partial u_r}{\partial r} \right) + \frac{\partial}{\partial \theta} \left[\eta \left(\frac{\partial u_r}{\partial \theta} + \frac{\partial u_\theta}{\partial r} - \frac{u_\theta}{r} \right) \right] + \frac{\partial}{\partial z} \left[\eta \left(\frac{\partial u_r}{\partial z} + \frac{\partial u_z}{\partial r} \right) \right] + \frac{2\eta}{r} \left(\frac{\partial u_r}{\partial r} - \frac{\partial u_\theta}{r \partial \theta} - \frac{u_r}{r} \right), \quad (4)$$

$$u_r \frac{\partial u_\theta}{\partial r} + \frac{u_\theta}{r} \frac{\partial u_\theta}{\partial \theta} + u_z \frac{\partial u_\theta}{\partial z} + \frac{u_r u_\theta}{r} = -\frac{1}{\rho} \frac{\partial p}{\partial \theta} + 2 \frac{\partial}{\partial r} \left(\eta \frac{\partial u_\theta}{\partial r} \right) + 2 \frac{\partial}{\partial \theta} \left(\frac{u_r}{r} \right) + \frac{\partial}{\partial z} \left[\eta \left(\frac{\partial u_\theta}{\partial z} + \frac{\partial u_z}{\partial \theta} \right) \right] + \frac{\partial}{\partial r} \left[\eta \left(\frac{\partial u_r}{\partial \theta} + \frac{\partial u_\theta}{\partial r} - \frac{u_\theta}{r} \right) \right] + \frac{2\eta}{r} \left(\frac{\partial u_r}{\partial r} - \frac{\partial u_\theta}{\partial \theta} - \frac{u_\theta}{r} \right), \quad (5)$$

$$u_r \frac{\partial u_z}{\partial r} + u_\theta \frac{\partial u_z}{\partial \theta} + u_z \frac{\partial u_z}{\partial z} = -\frac{1}{\rho} \frac{\partial p}{\partial z} + 2 \frac{\partial}{\partial z} \left(\eta \frac{\partial u_z}{\partial z} \right) + \frac{\partial}{\partial r} \left[\eta r \left(\frac{\partial u_r}{\partial z} + \frac{\partial u_z}{\partial r} \right) \right] + \frac{\partial}{\partial \theta} \left[\eta \left(\frac{\partial u_\theta}{\partial z} + \frac{\partial u_z}{\partial \theta} \right) \right]. \quad (6)$$

2.1.2. Continuity Equation

$$\frac{u_r}{r} + \frac{\partial u_r}{\partial r} + \frac{1}{r} \frac{\partial u_\theta}{\partial \theta} + \frac{\partial u_z}{\partial z} = 0. \quad (7)$$

2.1.3.

Energy

Equation

$$\rho c \left(u_r \frac{\partial T}{\partial r} + u_\theta \frac{\partial T}{\partial \theta} + u_z \frac{\partial T}{\partial z} \right) - \alpha T \left(u_r \frac{\partial p}{\partial r} \right) = \frac{\partial}{\partial r} \left(\lambda \frac{\partial T}{\partial r} \right) + \frac{\lambda}{r} \frac{\partial T}{\partial r} + \lambda \frac{\partial^2 T}{\partial z^2} + \phi, \quad (8)$$

$$\phi = \eta \left\{ 2 \left[\left(\frac{\partial u_r}{\partial r} \right)^2 + \left(\frac{u_r}{r} \right)^2 + \left(\frac{\partial u_z}{\partial z} \right)^2 \right] + \left(\frac{\partial u_\theta}{\partial z} \right)^2 + \left(\frac{\partial u_r}{\partial z} \right)^2 + \left(\frac{1}{r} \frac{\partial u_r}{\partial \theta} + \frac{\partial u_\theta}{\partial r} + \frac{u_\theta}{r} \right)^2 \right\}, \quad (9)$$

FIGURE 3: Schematic of the Poiseuille–Couette flow in concentric disk plane pair in the cylindrical coordinate system.

FIGURE 4: Friction calculation diagram of plane flow distribution pair structure.

where u_r , u_θ , and u_z represent the velocity in r , θ , and z directions respectively; p is the pressure; T is the temperature; ρ is the density; η is the dynamic viscosity; λ is the thermal conductivity; c is the specific heat; and ϕ is the term of viscous dissipation. The continuity equation and Navier–Stokes equation of motion in r and θ directions do not include the velocity W .

2.2. Friction Model. As mentioned above, the pressure distribution of the gap flow between the plane flow pairs has an important influence on the friction torque of the plane flow pairs, so the calculated model of friction should be modified. As shown in Figure 4, the fluid zone can be divided into three parts. Zone Z_f refers to the waist hole of the rotor of the plane flow pairs, where the high-pressure fluid can flow to the distribution channel of stator of the plane flow pairs. So, the pressure of this part is almost the same; then, the area of this zone should be subtracted when calculating the force loaded on the plane flow pairs of the pressure. Zone Z_c refers to the

connect area C of the rotor and the stator; as mentioned above, there is no fluid here, so the fluid pressure of this zone is zero, and the differential pressure is just the high pressure P_i . Zone Z_g refers to the nonconnect area of the rotor and the stator, and the height of the gap is h_c . This zone is filled with fluid in which pressure can be defined as P_g . Then, the differential pressure ΔP_g of this zone and plane A (upper plane of rotor as shown in Figure 2) is $P_i - P_g$. But the pressure P_g is not a constant value, it not only changes with the height h_c of the gap but also differs with places of zone Z_g . As shown in Figure 3, zone Z_g can be divided into infinite number of infinitesimal regions $Z_g(r, r + \Delta r; \theta, \theta + \Delta \theta)$, where the pressure can be defined as $P_{g(r,\theta)}$. r refers to the radius of the infinitesimal region, and θ refers to the angle between the region and the basis line. As discussed above, the axial force F_P loaded on the plane flow pairs caused by the high-pressure fluid can be obtained:

$$F_P = P_i \pi \frac{d_c^2}{4} + \int_{d_c/2}^{r_i} \int_0^{2\pi} (P_i - P_{g(r,\theta)}) r d\theta dr + \int_{r_i}^{r_o} \int_0^{(\pi/2) - (\alpha_r/2)} (P_i - P_{g(r,\theta)}) r d\theta dr + \int_{r_i}^{r_o} \int_{(\pi/2) + (\alpha_r/2)}^{2\pi} (P_i - P_{g(r,\theta)}) r d\theta dr + \int_{r_o}^R \int_0^{2\pi} (P_i - P_{g(r,\theta)}) r d\theta dr, \quad (10)$$

$$\alpha_r = \alpha_c + 2 \arccos \left(\frac{r_i r_o + r^2}{r(r_i + r_o)} \right), \quad (11)$$

where d_c is the diameter of the boss in the connect zone Z_c , r_i is the inner radius of the rotor's waist hole, r_o is the outer radius of the rotor's waist hole, α_r refers to the waist hole's angle at the circle with radius r , α_c refers to the waist hole's opening angle, and R refers to the radius of the plane flow pairs. Then, the friction force F_μ caused by the fluid can be obtained by equation (2). As shown in equation (7), the pressure distribution $P_{g(r,\theta)}$ of the gap between the plane flow pairs should be obtained if the loaded force F_P is to be calculated.

In fact, the pressure distribution $P_{g(r,\theta)}$ can be obtained by setting the boundary conditions and solving equations (4)–(9). But the amount of calculation for this process is quite large. Hence, in this paper, CFD (computational fluid dynamics) numerical analysis method based on finite volume method is used to solve this problem.

2.3. Numerical Analysis Method. The software based on finite volume method is used to simulate the hydrodynamic state of this condition. The velocity and pressure distribution can be obtained from the simulation based on governing equations (4)–(9). The result can be used to solve the engineering problem like the force of friction.

For simulation, the 3D simulation model is needed to be built as shown in Figure 5. Then, the full runner model can be obtained. The runner model is divided into control

FIGURE 5: Simulation model and schematic diagram of parameters.

volumes with even structure quadrilateral grids as shown in Figure 6.

Due to the small clearance of the plane flow pairs, the whole fluid domain is not completely turbulent, and laminar flow may occur locally with a large pressure gradient. Therefore, the k- ϵ model of Fluent's two-equation turbulence model is selected. For simulating near-wall flows better, a boundary layer needs to be generated in grid division. The grid height of the first layer y_{wall} refers to the boundary layer which is determined by the following equation [24]:

$$y_{wall} = \frac{y^+ \mu}{\mu_\tau \rho} \quad (12)$$

In equation (12),

$$\begin{aligned} \mu_\tau &= \sqrt{\frac{\tau_{wall}}{\rho}}, \\ \tau_{wall} &= \frac{1}{2} C_f \rho U^2, \\ C_f &= 0.027 \text{Re}^{-(1/7)}, \\ \text{Re} &= \frac{\rho U L}{\mu}, \end{aligned} \quad (13)$$

where ρ refers to the density of the fluid, kg/m^3 ; μ refers to dynamic viscosity, $\text{Pa} \cdot \text{s}$; U refers to velocity of the fluid, m/s ; L refers to the length of runner, m ; and y^+ is selected based on the wall function [25].

In the simulation process, the flow boundary condition is used as the input and set to 21 L/S according to the actual use environment of the structure. The pressure boundary condition is used as the output, which is atmospheric pressure. Preliminary analysis suggests that the height h_c and the diameter d_c of the boss, the opening angle α_c of the waist hole, and the width R_c of the waist hole are the main factors affecting the discharge and pressure distribution of the gap flow, so the simulation scheme shown in Table 1 is designed.

Generally, the precision of numerical calculations is improved with the increase of mesh number, but more grids mean more computations and longer computation time, so it is useful to determine the number of grids reasonably. For

this paper, the mesh independence study was performed to assess the accuracy of converged solutions. A group of parameters ($h_c = 0.25 \text{ mm}$; $d_c = 10 \text{ mm}$; $\alpha_c = 90^\circ$; and $R_c = 8 \text{ mm}$) was selected to verify the object. The model was discretized into mesh cells of 1.03 million, 2.59 million, 3.26 million, and 4.47 million approximately by changing the base size of the mesh element. For each mesh size model, the inlet pressure P_i depends on the flow Q , and the result is plotted in Figure 7. Sufficiently accurate solutions were found, while the number of the mesh cells was fixed at approximately 3.26 million because the results for 3.26 million and 4.47 million elements were almost the same for all cases of the flow.

3. Results and Discussion

3.1. Pressure Distribution. According to the simulation scheme shown in Table 1, the pressure distribution of the rotating plane flow pairs under different conditions can be obtained, as shown in Figures 8–11.

Corresponding pressure cloud patterns have similarities when the height of the boss h_c , the diameter of the boss d_c , the angle of the opening of the waist hole α_c , and the width of the waist hole R_c are changed, respectively.

Since the flow holes of the stator correspond to the outlet low-pressure region, the pressure gradient is large in the vicinity of the drain hole. The waist hole of the rotor is connected to a certain hole of the stator to form a high-pressure flow channel, and part of the fluid flows into the remaining holes through the gap between the plane distribution pairs to form the low-pressure channels. In the actual simulation analysis, two more holes are symmetrically arranged, so the pressure distribution is symmetrical basically.

To analyze the influence of different parameters on the pressure distribution more clearly, three paths on the bottom surface of the upper rotor of the rotating plane distribution pair are selected for analysis, as shown in Figure 10. P1 passes vertically through the center of the high-pressure runner hole H1, and P2 does not pass through any drain hole, and the angle with P1 is 45° . P3 passes through the low-pressure runner hole H2 or H3. Since the purpose of analyzing the pressure distribution is to calculate the frictional resistance between the rotor and the stator, the pressure

FIGURE 6: Diagram of the model mesh grid.

TABLE 1: Simulation parameters.

Simulation number	Height of boss (h_c , mmmm)	Diameter of boss (d_c , mmmm)	Angle of waist hole (α_c , °)	Width of waist hole (R_c , mmmm)
01–05	0.25/0.5/1/2/4	10	90	8
11–15	0.5	4/6/8/10/12	90	8
21–25	0.5	10	30/60/90/120/150	8
31–35	0.5	10	90	4/6/8/10/12

FIGURE 7: Model mesh independent analysis.

FIGURE 8: Variation of pressure p_g distribution caused by the change in height of the boss h_c .

FIGURE 9: Variation of pressure p_g distribution caused by the change in diameter of the boss d_c . FIGURE 10: Variation of pressure p_g distribution caused by the change in angle of the waist hole opening α_c . FIGURE 11: Variation of pressure p_g distribution caused by the change in width of the waist hole opening R_c .

selected here is the value on the lower surface of the rotor (plane B, as shown in Figure 12).

As shown in Figure 13(a), in P1-2 part, the pressure is distributed as the valley type. As the distance from the center of the plane flow pairs increases, the pressure decreases first and then increases. In P1-1 part, the pressure is U-shaped and the pressure difference changes more obviously. The smallest area of P1 is the section which passes through H1. H1 is a high-pressure discharge port where the flow rate is large and the pressure decreases faster. The height of the boss h_c has a significant influence on the pressure. As h_c increases, the pressure on the bottom surface of the upper rotor decreases, but when h_c increases to a certain value (4 mm in the simulation example), the pressure distribution of the P1 path essentially remains a horizontal line, which means that the pressure is the same everywhere, and there is no distinction between high-pressure and low-pressure channels in the entire basin. It is also confirmed in Figures 14 and 15. As shown in Figures 13(b) and 13(c), the diameter of the boss d_c and the angle of the waist hole α_c have a small effect on the magnitude of the pressure at P1. As shown in Figure 13(d),

when the waist hole width R_c is smaller than the diameter of H1, the pressure on the path is large and the change is obvious, but when the waist hole is larger than the diameter of the drain hole, the size changes on P1 have no obvious influence on the pressure distribution essentially.

As shown in Figure 14, there is suddenly change in pressure at the position of the waist hole, and the pressure change in the rest is not obvious on P2 path. When the waist hole angle α_c is less than 45° , P2 does not pass through the liquid flow hole, and the pressure change of P2 is the same as the pressure change of P1-2 section. As α_c increases, the pressure on P2 path also increases; this is because the high-pressure domain increases as α_c increases. The boss diameter d_c has little influence on the pressure distribution on P2 path. Similar to P1 path, there will be a pressure increase once the waist hole width R_c is smaller than the radius of the discharge port H1.

As shown in Figure 13, the pressure distribution on P3 path is similar to that on P1-1. Generally, it is U-shaped, but the pressure change of the H2 region fluctuates more obviously, and its value is lower than the pressure value of the

FIGURE 12: Diagram of pressure distribution feature path selection.

FIGURE 13: Continued.

FIGURE 13: Pressure distribution on path P1 (left side shows the pressure distribution of path P1-1 and right side shows the pressure distribution of path P1-2). (a) Change in height of the boss h_c . (b) Change in diameter of the boss d_c . (c) Change in angle of the waist hole opening α_c . (d) Change in width of the waist hole opening R_c .

H1 region; this means that the discharge hole H2 which the path P3 passes through is a low-pressure discharge port.

In summary, it can be seen that the influence of each influencing factor, including the height of the boss h_c , the diameter of the boss d_c , the opening angle of the waist hole α_c , and the opening width of the waist hole R_c , on the changes of the pressure distribution of paths P1, P2, and P3 is similar, respectively. As the height of the boss h_c increases, the pressure of the fluid in the gap gradually decreases, and the pressure difference of each part also gradually decreases. When the height increases to 4 mm, the pressure of each part tends to be consistent, indicating that the flow distribution plane pairs are no longer functioning. The diameter of the boss d_c and the opening angle of the waist hole α_c have less influence on the pressure distribution of the fluid in the gap. When the waist hole opening width R_c is smaller than the

diameter of H1, the decrease in width R_c causes an obvious increase in pressure, and when the width is larger than the diameter of H1, the influence on the pressure change is inconspicuous.

3.2. Discharge Change. Through simulation analysis, the flow curve of each orifice can be obtained. Since the flow rate curves of the discharge holes H2 and H3 coincide, only two flow leakage change curves can be obtained, as shown in Figure 16. According to the fluid dynamics theory, in parallel pipelines, the flow and pressure of each branch are related, and the pressure distribution has been analyzed in the previous section. As shown in Figure 16(a), as the height of the boss decreases, the movement resistance of the fluid between the clearance of the plane flow pairs becomes

FIGURE 14: Pressure distribution on path P2. (a) Change in height of the boss h_c . (b) Change in diameter of the boss d_c . (c) Change in angle of the waist hole opening α_c . (d) Change in width of the waist hole opening R_c .

FIGURE 15: Continued.

FIGURE 15: Pressure distribution change on path P3. (a) Change in height of the boss h_c . (b) Change in diameter of the boss d_c . (c) Change in angle of the waist hole opening α_c . (d) Change in width of the waist hole opening R_c .

FIGURE 16: Continued.

FIGURE 16: Diagram of discharge change. (a) Change in height of the boss h_c . (b) Change in diameter of the boss d_c . (c) Change in angle of the waist hole opening α_c . (d) Change in width of the waist hole opening R_c .

FIGURE 17: Diagram of friction change. (a) Change in height of the boss h_c . (b) Change in diameter of the boss d_c . (c) Change in angle of the waist hole opening α_c . (d) Change in width of the waist hole opening R_c .

stronger, and it is more and more difficult for the high-pressure fluid to enter H2 and H3 through the clearance, resulting in a decrease of the flow rate and the pressure of H2 and H3. When the height of the boss h_c is larger than 4 mm, the movement resistance of the fluid between the clearance of the plane flow pairs is small, and the flow rates of each discharge holes are substantially the same. As shown in Figures 16(b) and 16(c), the change of the diameter of the boss d_c and the opening angle of the waist hole α_c cannot cause a drastic change in the pressure distribution, so the flow distribution of the three holes remains unchanged. As shown in Figure 16(d), when the waist hole width R_c is smaller than the radius of H1, the inlet for the fluid entering the channel of H1 is narrow, and this will block the flow of the fluid, so the pressure change will increase, the flow rate will decrease, and then the flow rate of H2 and H3 will increase; when R_c is larger than the radius of H1, the width will become larger and there will be almost no influence on the pressure distribution. So, the flow distribution of the flow holes is basically unchanged with the change in R_c .

3.3. Influence on Friction. As shown in the friction calculation equation (10), the pressure distribution is one of the main factors affecting the friction. The pressure distribution is solved to more accurately solve the frictional resistance in the relative motion between the plane distribution pairs. Since the pressure distribution between the gaps is not uniform, the integral method is used to obtain the friction between the distribution pairs. The method calculates the friction force on the microelement based on the simulated pressure distribution and then sums all the microelements to obtain the friction between the distribution pairs.

As shown in Figure 17(a), as the height of the boss h_c increases, the gap pressure gradually decreases, so the frictional force gradually decreases. When the height of the boss h_c is larger than 4 mm, the pressure of the gap remains substantially unchanged, so the frictional force remains unchanged. As shown in Figure 17(b), although the boss diameter d_c has little effect on the pressure of the gap, as the boss diameter increases, the area of fluid pressure increases, resulting in the frictional resistance to increase as well. As shown in Figure 17(c), the influence on the frictional resistance of the waist hole angle α_c is small, and the frictional force is only slightly decreased when α_c changes from 30° to 150° . As shown in Figure 17(d), when the waist hole width R_c is increased in the interval smaller than the H1 radius, since the pressure at the clearance is reduced, the frictional resistance is reduced too, but when the waist hole width is larger than the diameter of H1, the pressure does not change any more, so the friction also remains stable.

4. Conclusion and Discussion

Rotating plane flow pairs have been widely used in engineering, and how to improve their sensitivity and service life is a problem. In this paper, the traditional rotating plane flow pair is improved, and the boss which is easy to realize in engineering is introduced, so as to form a gap between the

flow pairs; the theoretical analysis of the fluid movement between the gaps is carried out, and the relationship between the pressure distribution and the friction between the flow pairs is established. From simulation, the effect of the height of the boss, the diameter of the boss, the waist hole opening angle, and waist hole opening width on pressure distribution and leakage amount was found as follows:

- (1) As the height of the boss increases, the pressure of the fluid in the gap gradually decreases, and the pressure difference of each part also gradually decreases. Once the height of the boss increases to a certain value (4 mm in this study), the discharge of the three holes is the same, and the flow pairs lose their effectiveness. The increase of the height of the boss will significantly reduce the friction between the distribution pairs, but the rate of change will gradually decrease with the increase of height, and eventually it will have no effect on the friction.
- (2) The change of the diameter of the boss has no obvious influence on the pressure distribution between the flow pairs and the flow distribution between the discharge holes, but as the boss diameter increases, the area of fluid pressure increases, resulting in the frictional resistance to increase as well.
- (3) The change of the opening angle of the waist hole has no obvious influence on the pressure distribution between the distribution pairs and the flow distribution between the discharge holes.
- (4) When the opening width of the waist hole is smaller than the radius of the drain hole, the pressure changes obviously, and the flow difference between the high-pressure drain hole H1 and the low-pressure drain holes H2 and H3 becomes small, and the frictional resistance gradually decreases too. But when the width is larger than the radius of the drain hole, the pressure distribution, the discharge flow change, and the friction resistance change are not obvious as the width increases.

The above conclusions are beneficial for the determination of the parameters in the design process, and it is helpful to understand the antifriction mechanism of the structure and enrich the research of gap flow. For most distribution pairs, the structure is generally used in a relatively low rotation speed condition. Therefore, this paper does not focus on the rotational pressure versus pressure distribution and friction. But in fact, as the rotational speed increases, the disturbance between the fluids becomes more obvious, and the effect of viscosity on the friction is more obvious, and thus the temperature will become an important factor, which will become the important direction of the work in the future.

Data Availability

All data, models, and codes generated or used during the study are available from the corresponding author upon request.

Conflicts of Interest

The authors declare that they have no conflicts of interest.

Acknowledgments

This study was supported by the National Natural Science Foundation of China (51704264) and the Fundamental Research Funds for the Central Universities (2-9-2018-096).

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