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Flow analysis of asymmetric clearance and optimization of pressure equalization grooves to mitigate hydraulic spool valve sticking

Zhen-hao LIN[1,2], Yu-wei WANG[1], Zhe-hui MA[1], Tian-xiao ZHANG[1], Zhi-jiang JIN[1,3], Jin-yuan QIAN[1,4]✉

[1]*Institute of Advanced Equipment, College of Energy Engineering, Zhejiang University, Hangzhou 310027, China*
[2]*Hangzhou Valvever Intelligent Technology Co., Ltd, Hangzhou 310058, China*
[3]*Institute of Wenzhou, Zhejiang University, Wenzhou 325036, China*
[4]*State Key Laboratory of Fluid Power and Mechatronic Systems, Zhejiang University, Hangzhou 310058, China*

Abstract: The hydraulic spool valve is a critical control component in aerospace hydraulic systems. However, complex working environments can cause the valve core to become stuck, thus severely restricting the performance of such valves. This in turn can hinder precise control of hydraulic oil, reduce the stability of the hydraulic system, and lead to serious accidents in aerospace systems. The unbalanced radial force and solid particle intrusion into the fit clearance are the main factors behind this sticking. To better understand these issues, in this study we simulated the fluid dynamics and particle behavior within the clearance of the valve core, and analyzed the effects of inclination angle, clearance size, particle diameter, and pressure equalization groove (PEG) properties. The mechanism behind valve core sticking was revealed, and it was found that the PEG has an inhibitory effect on the unbalanced radial force and the particle intrusion. Furthermore, we propose an optimized structure for a triangular pressure equalization groove with an arc-shaped bottom (Tri-PEG). The structural parameters were determined through multi-objective optimization, with the objectives of minimizing the leakage at the clearance and maximizing the particle volume fraction at the bottom of the Tri-PEG. The optimal parameters were an arc-shaped radius of 0.2 mm, a groove depth of 0.392 mm, and a groove width of 0.215 mm. Comparing the Tri-PEG with a rectangular PEG, the leakage was reduced by 12%, and the particle concentration was increased by 6%. Overall, these findings serve as an important reference for alleviating spool valve sticking.

Key words: Spool valve; Flow characteristics; Multi-objective optimization; Solid-liquid two phase flow; Aerospace hydraulic systems; Valve sticking

1 Introduction

Hydraulic systems serve as fundamental subsystems in aerospace applications, supporting various safety-critical operations (Hou et al., 2025; Deng et al., 2023; Shanbhag et al., 2021). Hydraulic spool valves, being common control components in hydraulic systems, play an important role in controlling flow rate (Lu et al., 2022), pressure (Zhong et al., 2021), and flow direction (Zhao et al., 2021) in such systems (Li et al., 2023). Accordingly, faults in hydraulic spool valves can cause serious problems and accidents (Li et al., 2025). In particular, valve core sticking may lead to performance degradation and function maladjustment (Chu et al., 2021); this is one of the most common faults in hydraulic systems (Wang et al., 2023). Suppression or resolution of this problem is thus vital for improving the safety and stability of aerospace hydraulic systems (Qian et al., 2023).

Valve core sticking is the phenomenon of increased resistance and delayed response of the valve core during movement (Li et al., 2014). The increased resistance is due to the sticking force. For hydraulic spool valves, the main causes of valve core sticking are the unbalanced radial force (Zhang et al., 2024) and solid particle contamination (Yaobao et al., 2017; Liu et al., 2020; Chen et al., 2024).

✉ Jin-yuan QIAN, qianjy@zju.edu.cn
Zhen-hao LIN, https://orcid.org/0000-0003-3040-7656
Yu-wei WANG, https://orcid.org/0009-0000-7935-4765
Zhe-hui MA, https://orcid.org/0009-0008-9552-4873
Zhi-jiang JIN, https://orcid.org/0000-0002-8063-709X
Jin-yuan QIAN, https://orcid.org/0000-0002-5438-0833

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The unbalanced radial force on the surface of the valve core is caused by chaotic turbulent flow of the fluid (Zhang et al., 2025a). The fluid boundary in the valve is formed by the valve body and internal components; this boundary is asymmetric and complex. Thus, analytical calculation of the unbalanced radial force is difficult. A multitude of numerical simulation studies have been conducted to analyze the factors influencing the unbalanced force. For instance, Gao et al. (Gao et al., 2020) analyzed the torque of the butterfly plates of flip check valves and swing check valves. Under fully-open conditions, the former was 53.86% of the latter, which is also the reason why flip check valves are prone to vibration. Also, Qian et al. (Qian et al., 2016; Qian et al., 2021) analyzed the formation mechanism of unbalanced force on the bottom surface of piston-type valve cores. Their results indicated that due to the curvature of the inlet flow channel, the inlet fluid formed an impact on the bottom surface of the valve core. This caused the pressure on the bottom surface of the valve core near the inlet to be lower than that near the outlet, resulting in an unbalanced torque and tilting of the valve core. Moreover, Lisowski et al. (Lisowski et al., 2018) studied the fluid force on the core of a proportional control valve during the initial stage of spool opening. They found that a single incision can enable the valve to operate at very low flow rates, and that there is significant radial hydraulic asymmetry.

During processing and assembly of hydraulic spool valves, inevitable problems arise such as eccentricity, reverse coning, inclination, and end protrusion. These problems can lead to asymmetric fit clearance between the valve core and the valve body. In asymmetric fit clearance, the pressure drop is large at positions with small clearance, and is small at positions with large clearance. This results in an uneven pressure difference on the surface of the valve core (see Fig. S1).

When solid particles from outside the valve, or those generated by corrosion and wear of valve components, are mixed into the fluid, these solid particles enter the clearance between the valve core and other components with the fluid. When solid particles, the valve core, and other components are in contact with each other, the resistance of the valve core increases. This can cause the valve core to become stuck or jammed (Zhang et al., 2025b). It may even threaten the stability and safety of the entire system (Sun et al., 2021). Therefore, it is crucial to analyze the flow characteristics of solid particles and reveal their patterns of motion. With this in mind, Terrell et al. (Terrell and Higgs III, 2007) proposed a kinematic trajectory model for predicting material wear particles; their predicted results were compared with experimental data and shown to be accurate. Also, Mittal et al. (Mittal and Iaccarino, 2005) simulated the flow of pollutant particles in the fitting clearance based on the submerged boundary method, solving the problem of submerged solid boundary flow. Furthermore, Domagala et al. (Domagala et al., 2018) constructed collision angle and velocity models for particles of different materials inside the spool valve. This model was used to predict the motion of particles inside the valve body.

In hydraulic spool valves, when unbalanced radial forces and solid particles both exert influence, they can cause coupling sticking issues (see Fig. S2). The unbalanced radial force causes the valve core to tilt, resulting in an asymmetric fit clearance. The reduction of clearance and the invasion of solid particles together led to the sticking of the valve core.

Numerous studies have focused on preventing control valve core sticking and optimizing valve core designs. For instance, Amirante et al. (Amirante et al., 2016) reduced the hydraulic force on the valve core by designing a central conical surface and side structure of the valve core. Also, Zhang et al. (Zhang et al., 2020) analyzed the flow characteristics of coupled throttling channels. A neural network and genetic algorithm were used to optimize the design of the throttling groove structure, with the goal of evaluating the throttling stiffness for flow stability. Moreover, Gui et al. (Gui et al., 2022) conducted optimization analysis on the valve core structure using the optimal non-dominated sorting genetic algorithm (NSGA-II). Their results showed that the optimized hydraulic force was significantly reduced, and the piezoelectric servo valve had good response performance.

Many scholars have proposed asymmetric pressure equalization groove structures to reduce the hydraulic force of the valve core in the fitting clearance. For example, Hong et al. (Hong and Kim, 2016) proposed a spiral pressure equalization groove structure and compared its performance with typical pressure equalization grooves. It was found that valve cores with spiral grooves are more likely to alleviate the situation of uneven pressure distribution around the valve core. Plus, Zheng et al. (Zheng et al., 2020) proposed a composite groove structure consisting of an annular pressure equalization groove and a spiral pressure equalization groove. The flow characteristics of the composite groove piston were analyzed from the aspects of pressure distribution, tilting torque, clearance leakage, pressure balance, and clearance flow rate. The above two types of groove structures can effectively reduce the upper and lower pressure difference of the valve core. However, due to the connected state between adjacent pressure equalization grooves, when the end of the valve core does not coincide with the valve body, it may cause significant leakage, therefore this is unsuitable in some situations.

Although numerous studies have been conducted to investigate valve core sticking in control valves and associated sticking suppression methods, there are still limitations in current mechanisms of valve core sticking as well as optimized designs of valve cores. Furthermore, studies on valve core sticking under the coupled effect of unbalanced radial force and solid particles are still relatively limited. Meanwhile, there is sparse research on valve core optimization considering multiple target quantities.

To address these limitations, we analyze the fluid dynamics and particle behavior within the fit clearance of a hydraulic spool valve under different inclination angles, clearance sizes, and particle diameters. Moreover, we introduce a triangular pressure equalization groove with an arc-shaped bottom (Tri-PEG), and optimize its structural parameters to achieve a balance between reducing leakage volume and lessening solid particle aggregation.

2 Methods

2.1 Physical model

The hydraulic spool valve is mainly composed of the driving component, valve body, valve sleeve, and valve core, as shown in Fig. 1. At the position marked by the red frame, the valve core and the sleeve can easily come into contact. We assume that there is no eccentricity resulting from the assembly of the valve core, valve body, and valve sleeve, and that the valve core has no reverse coning or burrs from processing. We establish a model for the fit clearance after the valve core shoulder is tilted, including the clearance flow channel models with and without pressure equalization grooves, as depicted in Fig. 1b-1c, respectively. The inclination angles are γ= 0.01 °, 0.03 °, and 0.05 °, and the initial values of the clearance between components are δ= 15 μm, 20 μm, and 25 μm. When the clearance is 10 μm or less, due to the inclination angle, the minimum gap may be smaller than the particle diameter. The valve core will then experience severe mechanical jamming, and so the flow characteristics and force analysis will have no research significance.

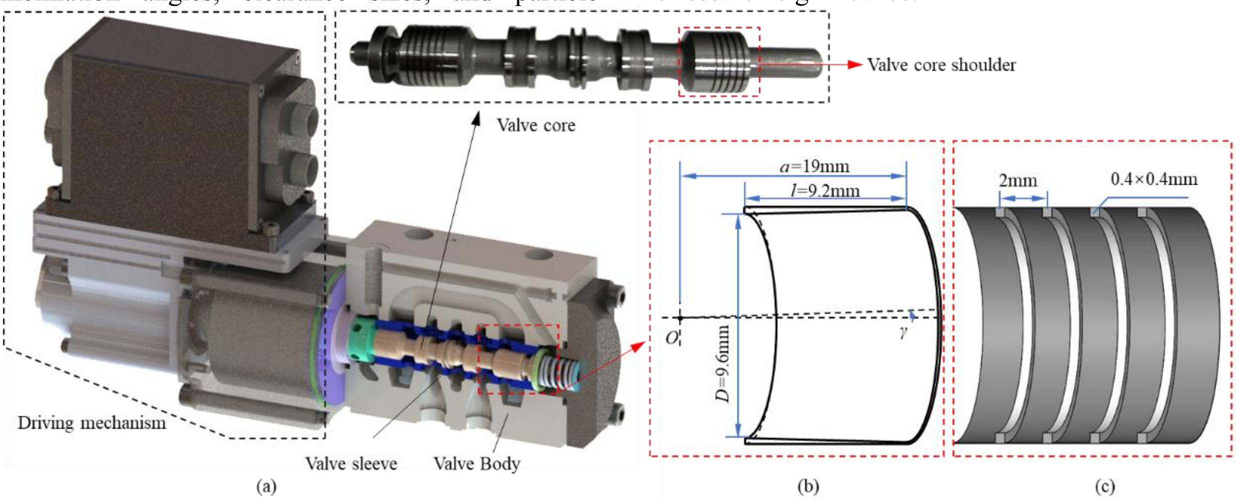


Fig. 1 Hydraulic spool valve and flow channel of the fit clearance

2.2 Numerical methods

In this study, hydraulic oil containing solid particles was used as the medium for numerical modeling. The fluid phase and solid phase were assumed to be continuous fluids. An Euler-Euler solid-liquid two-phase flow model was adopted, which is commonly used to solve multi-phase flow problems with interactions between phases (Shi et al., 2024). Compared with a mixed model, this model has higher calculation accuracy. In the Euler-Euler model, each phase satisfies the continuity equation and the momentum equation. The continuity equation for the k phase is as follows:

$$\frac{\partial}{\partial t}\left(\alpha_k \rho_k\right)+\nabla\cdot\left(\alpha_k \rho_k \boldsymbol{v}_k\right)=\sum\left(\dot{m}_{pk}-\dot{m}_{kp}\right) \quad (1)$$

where $\boldsymbol{v}_k$ is the velocity of the k phase, $\dot{m}_{pk}$ is the derivation of mass transfer from the k to p phases, $\dot{m}_{kp}$ is the derivative of mass transfer from the p to k phases, t is the time, α_k is the volume ration of the k phase, and ρ_k is the density of the k phase.

The momentum equation of the k phase is as follows:

$$\begin{aligned}&\frac{\partial}{\partial t}\left(\alpha_k \rho_k \boldsymbol{v}_k\right)+\nabla\cdot\left(\alpha_k \rho_k \boldsymbol{v}_k \boldsymbol{v}_k\right)=-\alpha_k \nabla P\\ &+\nabla\cdot\overline{\overline{\tau}}_k+\alpha_k \rho_k \boldsymbol{g}+\sum\left(\boldsymbol{R}_{pk}+\dot{m}_{pk}\boldsymbol{v}_{pk}-\dot{m}_{kp}\boldsymbol{v}_{kp}\right) \quad (2)\\ &+\left(\boldsymbol{F}_k+\boldsymbol{F}_{\mathrm{lift},k}+\boldsymbol{F}_{v_{m'},k}\right)\end{aligned}$$

where $\boldsymbol{F}_k$ is the external volume force, $\boldsymbol{F}_{\mathrm{lift},k}$ is the lift force, $\boldsymbol{F}_{v_{m'},k}$ is the virtual mass force, $\boldsymbol{R}_{pk}$ is the interaction force between phases, P is the pressure shared by all phases, $\boldsymbol{v}_{pk}$ and $\boldsymbol{v}_{kp}$ are phase velocities, $\boldsymbol{g}$ is the gravitational acceleration, and $\overline{\overline{\tau}}_k$ is the pressure strain of the k phase. The value of $\overline{\tau}_k$ can be calculated as follows:

$$\overline{\overline{\tau}}_k=\alpha_k\mu_k\left(\nabla\boldsymbol{v}_k+\nabla\boldsymbol{v}_k^T\right)+\alpha_k\cdot\left(\lambda_k-\frac{2}{3}\mu_k\right)\nabla\cdot\boldsymbol{v}_k\boldsymbol{I} \quad (3)$$

where μ_k and λ_k are the shear viscosity coefficient and bulk viscosity coefficient of the k phase, respectively, and $\boldsymbol{I}$ is the unit tensor.

The volume fraction equation for each phase can be obtained from the continuity equation:

$$\frac{1}{\rho_{rk}}\left[\frac{\partial}{\partial t}\left(\alpha_k\rho_k\right)+\nabla\cdot\left(\alpha_k\rho_k\boldsymbol{v}_k\right)-\sum\left(\dot{m}_{pk}-\dot{m}_{kp}\right)\right]=0 \quad (4)$$

where ρ_{rk} is the phase reference density, that is, the volumetric average density of the k phase.

2.3 Numerical gridding and boundary conditions

The flow channel was discretized using a structured grid division method. Five layers of grids were set in the direction of clearance thickness, 400 layers in the axial direction, and 400 layers in the circumferential direction. The total number of grid cells is approximately 720,000. For the flow channel model with pressure equalization grooves, 18 layers of grids were set in the depth and width directions of the pressure equalization grooves, with the total number of grid cells being approximately 1,030,000; the gridding strategy is illustrated in Fig. 2.

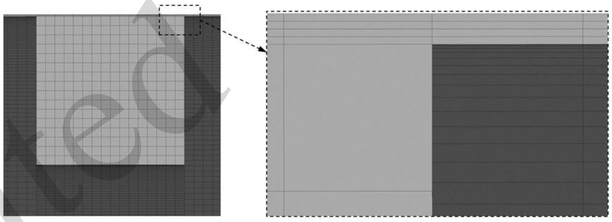


Fig. 2 Structure grid of the flow channel

The medium was a mixture of oil and solids. We set the boundary conditions as a pressure inlet and pressure outlet. The inlet pressure was set at 7 MPa, and the outlet pressure was set to the atmospheric pressure. The first phase is the oil, and we assume the second phase to be comprised of ideal spherical solid particles with a density of ρ=7500 kg/m^3 and a dynamic viscosity of μ=10^{-5} Pa·s. The solid particle diameters were set to range from 3 to 12 μm. The volume fraction of the inlet particles was set at 3%. In the Euler-Euler model, the drag force function between the solid-liquid two-phases was selected as Syamlal-O'Brien. In the phase interactions, the drag is selected as Schiller-Naumann, and the lift is selected as "none" – meaning there is no separation of phases.

In the present geometric model, the critical Reynolds number of an asymmetric annular clearance with a pressure equalization groove is the smallest, which is 400. It corresponds to a velocity of 366 mm/s, which is greater than the actual flow velocity of the fit clearance. Therefore, the fluid flow in the clearance is in a laminar state, and the viscous model adopts a Laminar model.

In this study, a pressure-based solver was employed for the numerical solution. The gradient was set to be least squares cell based, which is calculated based on the cell center. The pressure, momentum, and turbulence terms were second order upwind with higher accuracy. The discretized control equations are solved using the SIMPLE algorithm. In the numerical solution process, we consider that convergence has occurred when the residual of the monitored physical quantity is less than 10^{-6}, and the inlet and outlet pressure difference remains constant with fluctuations below 1%.

2.4 Method validation

In order to verify the accuracy of the numerical simulation, a concentric ring clearance model was established. The clearance flow obtained from this theoretical calculation was compared with that from the numerical calculation. Due to $\delta / D << 1$, the flow in the annular clearance can be considered as parallel plate gap flow. According to the principle of clearance flow, the formula for the flow of the concentric ring-shaped clearance between the valve core and the valve body is as follows:

$$Q_T = \frac{\pi D \delta^3 \Delta p}{12 \mu l} \tag{5}$$

where δ is the fit clearance between the valve core and sleeve, D is the diameter of the valve core (D=9.6 mm), Δp is the pressure difference at the inlet and outlet of the fit clearance (Δp=7 MPa), l is the length of the fit clearance (l=10 mm), and μ is the kinematic viscosity.

The flow calculated by theoretical methods was compared with that of the numerical simulation, as shown in Table 1. The maximum error between the theoretical and simulation results is 6.06%, and the fit clearance size is 5 μm. The error generated is within a reasonable range, and thus the numerical method and results are accurate and reliable.

Table 1 The flow calculated by the theoretical equation and the numerical simulation at different fit clearances

δ (μm)	Q_T (mL/min)	Q_S (mL/min)	Error (%)
5	0.33	0.35	6.06
10	2.65	2.73	3.02
15	8.95	9.44	5.47
20	21.22	21.84	2.92

We also conducted experimental verification of the simulation model, with the results shown in Table 2 (Wang et al., 2026). The maximum error between the experimental and simulation results is 7.8%, which is within a reasonable range, indicating that the simulation method is reliable.

Table 2 Comparison of flow rate for the experimental and simulation results (Wang, et al., 2026)

Opening (%)	Experimental results (L/min)	Simulation results (L/min)	Error (%)
10	3	3.1	3.3
30	10	9.2	8
50	16	14.9	6.9
70	19.5	20.2	3.5
90	23.5	25.5	7.8

3 Results and discussion

3.1 Fluid force of the valve core within an asymmetric fit clearance

This section focuses on the fluid force within the asymmetric fit clearance of the valve core. The impact of the inclination angle on the fluid force of the valve core was analyzed. The variation patterns of the fluid dynamics of the valve core with uniform grooves were studied under different inclination angles.

(1) Fluid force analysis without pressure equalization groove

The flow channel model with an initial fit clearance of 20 μm was selected for analysis. The pressure distribution on the upper and lower surfaces of the valve shoulder at different inclination angles is shown in Fig. 3. As one can see in Fig. 3a, the pressure distribution on the surface of the valve core shoulder is significantly uneven. At the same axial position, the pressure on the upper surface of the valve core shoulder is higher than that on the lower surface. From Fig. 3b, it is clear that as the inclination angle increases, the pressure on the lower surface of the valve core shoulder decreases, while the pressure on the upper surface increases. In the axial direction, at the same position, the pressure increment on the upper surface is increasing. According to the Bernoulli principle (Li et al., 2021), when the fit clearance size between the upper surface and the valve sleeve is small at the outlet, the speed decreases, and the pressure increases.

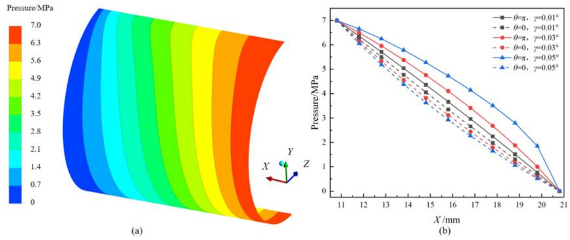


Fig. 3 Pressure of the upper and lower surfaces of the valve core shoulder at different inclination angles. (a) Pressure distribution, (b) Pressure curves

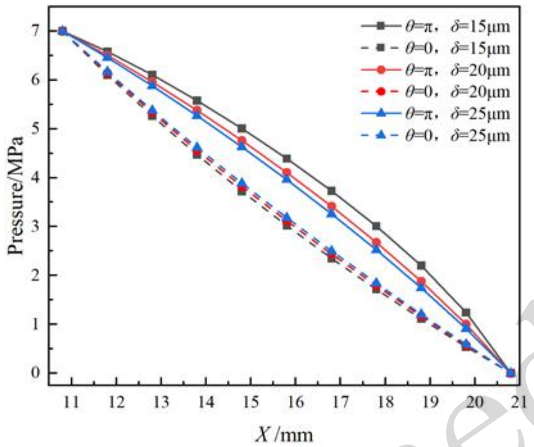


Fig. 4 Pressure curves of the upper and lower surfaces of the valve shoulder for different clearances

Next, the flow channel model with an inclination angle of 0.03° is selected for analysis. Under different valve core shoulder clearance values, the pressure curves of the upper and lower surfaces are shown in Fig. 4. As the clearance increases, the pressure on the upper surface of the valve core shoulder decreases, while the pressure on the lower surface increases. This results in a reduction in the radial fluid force acting on the valve core. However, the increased valve core clearance leads to increased leakage. Therefore, it is necessary to comprehensively consider leakage and radial fluid force to achieve optimal design of the valve core clearance.

The distribution of pressure difference between the upper and lower surfaces of the valve shoulder under different clearance sizes and inclination angles is shown in Fig. 5. The pressure difference is positively correlated with the inclination angle and negatively correlated with the clearance. The pressure difference with a clearance of 15 μm and an inclination angle of 0.03° is equal to the pressure difference with a clearance of 25 μm and an inclination angle of 0.05°.

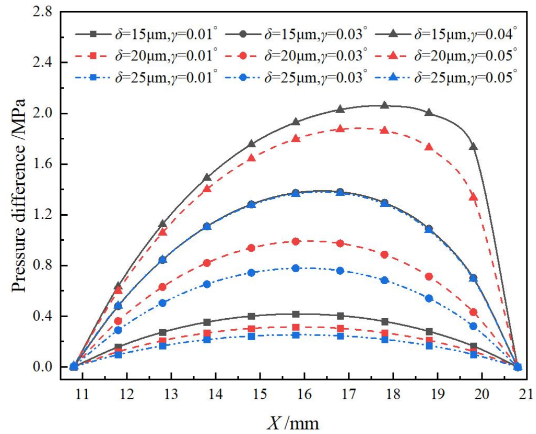


Fig. 5 Pressure difference curves of the upper and lower surfaces

Table 3 shows the radial forces acting on the valve core shoulder within the fit clearance. The radial fluid force increases with the inclination angle, and decreases as the fit clearance increases. The increased radial fluid force further increased the inclination of the valve core, making it more prone to contact with the valve sleeve, thereby causing sticking problems.

Table 3 The radial fluid force acting on the valve core at different inclinations

Clearance/Angles	0.01°	0.03°	0.05°
15 μm	21.14 N	65.76 N	/
20 μm	15.99 N	48.75 N	83.96 N
25 μm	12.83 N	38.81 N	65.23 N

(2) Fluid force analysis with pressure equalization groove

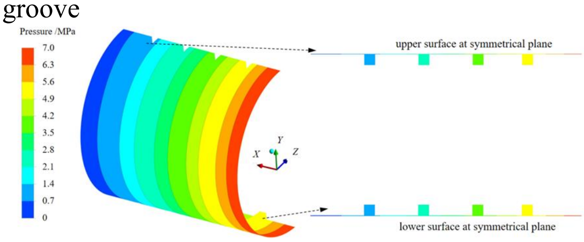


Fig. 6 Pressure distribution of the valve core shoulder surface with PEG

Establishing a pressure equalization groove (PEG) on the valve core shoulder is the mainstream improvement measure to overcome the uneven pressure distribution within the fit clearance. Fig. 6 shows the pressure distribution of the valve core shoulder surface with PEG. The pressure distribution on the upper and lower surfaces of the valve core shoulder along the circumferential direction is more uniform compared to the case without the PEG in Fig. 3, especially at the position of the PEG. However, in the position without the PEG (inlet and outlet), the fit clearance between the upper and lower surfaces of the valve core shoulder is asymmetrical. Moreover, the pressure on the upper surface at the same position is greater than that on the lower surface. It can thus be concluded that the PEG effectively divides the pressure within the fit clearance into multiple pressure reduction zones.

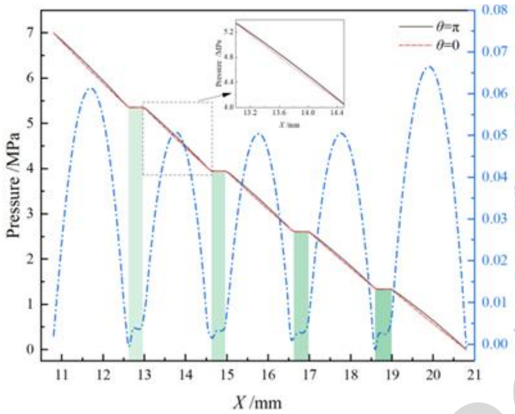


Fig. 7 Pressure curve of the valve core shoulder surface with PEG

Fig. 7 presents the pressure curve on the upper and lower surfaces of the valve core shoulder at a clearance of 20 μm and an inclination angle of 0.03°. As seen, the presence of the PEG results in a similar pressure distribution on the upper and lower surfaces of the valve core shoulder. The maximum pressure difference between them occurs at the inlet, and is 0.075 MPa. Compared to the case without the PEG, the pressure difference is reduced by a factor of about 100. Meanwhile, the existence of the PEG divides the uneven pressure distribution within the clearance into multiple segments. At the position of the PEG, there is almost no pressure difference.

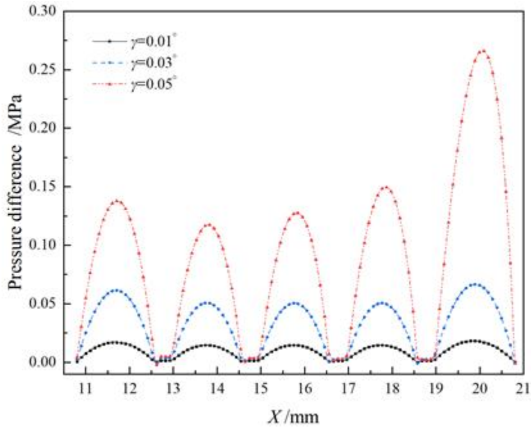


Fig. 8 Pressure difference curves with PEG at different inclination angles

Furthermore, the pressure difference curves of the valve core shoulder surface with PEG at different inclination angles are shown in Fig. 8. As the inclination angle increases, the pressure difference in the PEG region remains almost unchanged, while the pressure difference in the region without PEG increases. At this point, the pressure difference at the outlet is significantly greater than that at the inlet.

The radial fluid force of the valve core shoulder at a clearance value of 20 μm is shown in Table 4. It can be seen that the radial fluid force of the valve core shoulder with PEG is significantly less than that of the valve core shoulder without PEG; the reduction in force was over 93%. This effectively reduced the inclination degree of the valve core.

Table 4 The radial fluid force of the valve core shoulder at different inclination angles

Angle	0.01°	0.03°	0.05°
Without PEG	15.99 N	48.75 N	83.96 N
With PEG	0.70 N	2.32 N	5.41 N

3.2 The distribution of solid particles within the asymmetric fit clearance

This section focuses on flow characteristics of oil with solid particles within the asymmetric fit clearance. An investigation of the aggregation laws of solid particles under conditions with PEG and without PEG was performed.

(1) The distribution of solid particles without PEG

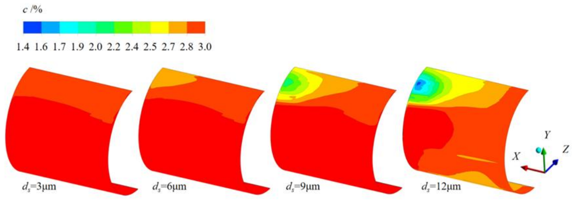


Fig. 9 The distribution of the particle volume fraction (δ=15 μm, γ=0.01°, c_0=3%)

The flow channel model at δ=15 μm and γ=0.01° was selected for analysis, and the minimum clearance size is 13.72 μm. The distributions of the particle volume fraction (c) within the fit clearance under different solid particle diameters are shown in Fig. 9, when the inlet particle volume fraction (c_0) is 3%. The existence of the valve core inclination angle makes the upper part of the valve core shoulder have a smaller size, resulting in particles accumulating on the lower surface. As the diameter of the solid particles gradually increases, the particle volume fraction is lower in the middle area of the outlet, and higher in the surrounding area. This is because the particle diameters gradually approach the clearance size, resulting in a low concentration of particles in the middle area. This is particularly evident when the solid particle diameter is 12 μm, where the minimum particle volume fraction reaches 1.4%.

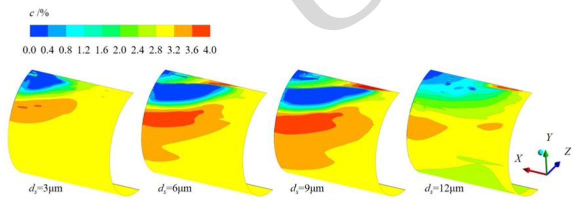


Fig. 10 The distribution of the particle volume fraction (δ=15 μm, γ=0.03°, c_0=3%)

Similarly, the distribution of the particle volume fraction (c) within the fit clearance at γ=0.03° is depicted in Fig. 10. The minimum clearance size is 8.11 μm. When the particle diameter is greater than 6 μm, particles will aggregate on the upper surface of the valve core shoulder. The volume fraction of particles is approximately 4%, which is greater than the volume fraction of the inlet. Near the minimum fit clearance size, the volume fraction of solid particles is 0. When the particle diameter is greater than the clearance size, the particles will be unable to enter; this is the sensitive sticking area. When solid particles continue to accumulate in the fit clearance, it will seriously hinder the movement of the valve core, resulting in sticking. When the inclination angle is 0.05°, the minimum size of the clearance is 0.85 μm, which is smaller than the particle diameter (therefore this scenario doesn't lead to sticking issues).

(2) The distribution of solid particles with PEG

The distribution of solid particles in the clearance with PEG at different inclination angles is shown in Fig. 11. Compared with the distribution of solid particles in clearance without PEG, the solid particles are mainly located at the position of the PEG. Moreover, the particle accumulation slows down in the sensitive sticking area. As the size of the solid particles increases, the particle volume fraction at the position of the PEG also increases. When the inclination angle is 0.01°, the maximum particle volume fraction increased from 3.55% at a d_s of 3 μm to 5.7% at a d_s of 6 μm. Varying inclination angles have different effects on the particle aggregation of the PEG. As the inclination angle increases, the clearance size changes, resulting in an increase in the particle volume fraction at the PEG.

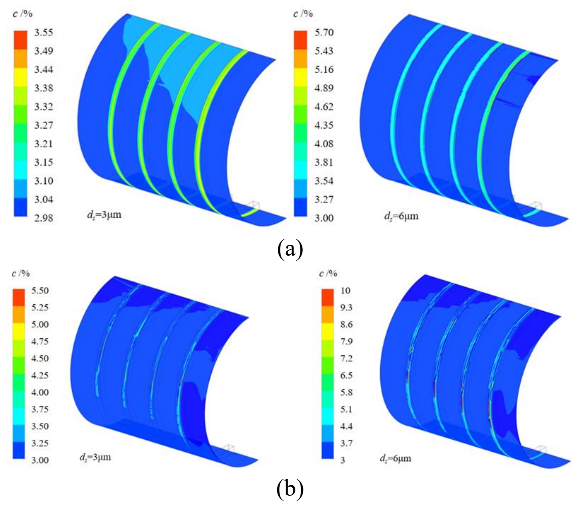


Fig. 11 The distribution of the particle volume fraction with PEG. (a) γ=0.01°, (b) γ=0.03°

3.3 Optimization design of the PEG

Based on the above analysis, the valve shoulder with PEG can reduce the radial fluid force, as well as decrease the valve core inclination degree. At the same time, it is beneficial to the aggregation of particles in the PEG, indicating that the PEG can effectively reduce the probability of valve core sticking. In this section we propose a triangular pressure equalization groove with an arc-shaped bottom (Tri-PEG). The optimal structural parameters of the Tri-PEG are determined through the response surface analysis method and the multi-objective genetic algorithm.

(1) Optimizing the design of the PEG

Our previous study revealed that the Tri-PEG has a more significant effect on particle retention compared to uniform rectangular grooves (Hang et al., 2026). It can effectively prevent particles from aggregating in the area without the PEG. Therefore, in order to enhance the dirt-retaining capacity of the PEG and reduce the leakage volume of the clearance, a PEG consisting of a triangular cross-section and an arc-shaped bottom was designed as shown in Fig. 12. Meanwhile, the structural dimensions of the original valve core shoulder were maintained.

Response surface methodology (RSM) was used to optimize the structural parameters of the Tri-PEG. The accuracy of the fitted response surface is significantly influenced by the distribution of the test sample points. To ensure that each selected design point is representative and to improve computational efficiency, the Latin Hypercube Sampling (LHS) method (Aljubran et al., 2020) is used to design the test samples. The sample type is CCD sampling, and a total of 15 sets of test design points were generated. The design parameters and results are presented in Table S1 and Table S2.

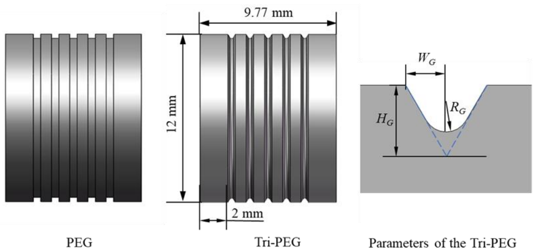
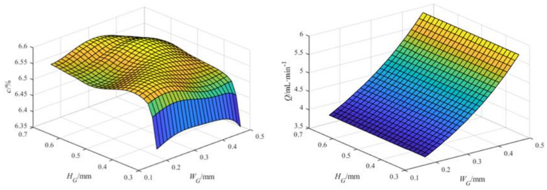


Fig. 12 Structural diagram of Tri-PEG

(2) Response surface analysis

The response relationship between the design parameters and the particle volume fraction and the clearance leakage is obtained through fitting of the above simulation data. Fig. 13a shows how the particle volume fraction of the Tri-PEG's bottom varies with W_G and H_G at R_G=0.15 mm. The particle volume fraction increases sharply as H_G varies from 0.3 mm to 0.4 mm, while W_G has little effect on the particle volume fraction. As H_G increases, the volume of the PEG expands, and the residence time of the fluid in the PEG is prolonged, making it easier for particles to be captured by vortices in the groove. However, after H_G reaches a certain value, the flow field inside the groove tends to stabilize and the vortex structure becomes fully developed. Further increasing H_G weakens the effect of particle capture in the groove. So, as H_G varies from 0.4 mm to 0.7 mm, the resulting changes are relatively gentle. At this time, changing W_G will alter the cross-sectional shape of the groove, affecting the secondary flow intensity and the lateral transport of particles, and thereby affecting the volume fraction of particles in the groove. As W_G increases, the particle volume fraction first increases and then decreases. When W_G is constant, the effect of H_G on particle concentration slightly differs. When W_G=0.2 mm, the particle volume fraction shows a trend of first increasing, then decreasing, and then increasing again with greater HG. For W_G=0.3 mm, the particle volume fraction first increases and then decreases with an increase of H_G, and a maximum value region forms. When W_G is 0.45 mm, the particle volume fraction first increases and then stabilizes. From the above, it can be concluded that when H_G and W_G are both set to the middle value, the particle volume fraction at the Tri-PEG's bottom reaches a maximum.

Fig. 13b depicts the leakage of the clearance with Tri-PEG, which varies with W_G and H_G at R_G =0.15 mm. The leakage volume increases exponentially with greater W_G, rising from a minimum of 3.52 mL/min to 6 mL/min. This is because the leakage mainly depends on the axial clearance; as W_G increases, the effective flow area of the fluid leakage increases, making leakage more likely. The effect of H_G on the effective flow area of the fluid leakage is relatively small. Therefore, the impact of H_G on leakage is minor.

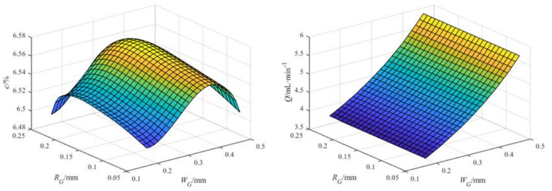


(a) (b)

Fig. 13 Variation of particle volume fraction and leakage volume with W_G and H_G (R_G=0.15 mm). (a) Particle volume fraction, (b) Leakage volume

Fig. 14a presents how the particle volume fraction of the Tri-PEG's bottom varies with W_G and R_G at H_G=0.5 mm. The particle volume fraction first increases and then decreases with an increase in W_G. When W_G is constant, the particle volume fraction shows a trend of increasing first and then decreasing with greater R_G. This indicates the existence of a critical R_G value, where the flow field and particle dynamics in the groove are optimally matched. When R_G exceeds this critical value, the centrifugal force in the groove weakens and the ability to capture particles decreases. For R_G in the range of 0.22 to 0.25 mm, there is an obvious decreasing trend. From these results, we can conclude that when W_G is approximately 0.3 mm and R_G is approximately 0.2 mm, the particle volume fraction reaches its maximum value.

Fig. 14b shows how the leakage volume of the fit clearance for Tri-PEG varies with W_G and R_G at H_G=0.5 mm. When R_G is constant, the leakage volume increases exponentially as W_G increases, which is consistent with the earlier results. When W_G is constant, R_G has relatively small influence on the leakage volume. Considering these findings, it is clear that the main factor influencing leakage is W_G, while H_G and R_G have a relatively small impact on leakage. Choosing a smaller W_G can thus promote a smaller leakage rate.

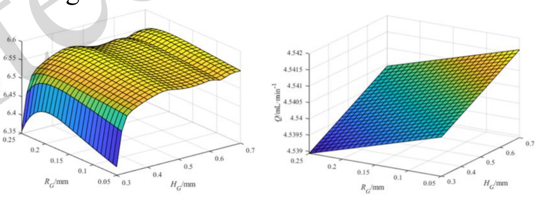


(a) (b)

Fig. 14 Variation of particle volume fraction and leakage volume with W_G and R_G (H_G =0.5 mm). (a) Particle volume fraction, (b) Leakage volume

In Fig. 15a, it is shown how the particle volume fraction of the Tri-PEG's bottom varies with H_G and R_G at W_G =0.315 mm. If R_G is constant, the particle volume fraction first increases and then stabilizes as H_G increases. For H_G in the range of 0.3 to 0.4 mm, the change in the particle volume fraction is significant; this is consistent with the trend from Fig. 13a. When H_G is 0.3 mm, the effect of R_G on the particle volume fraction is consistent with the above. After H_G exceeds 0.4 mm, the change in particle volume fraction is minimal, indicating that the effects of H_G and R_G on particle volume fraction cancel each other out. Considering these results, one can conclude that when H_G is approximately 0.4 mm and R_G takes on the middle value, the particle volume fraction reaches a maximum.

Fig. 15b depicts how the leakage of the fit clearance for Tri-PEG varies with H_G and R_G at W_G =0.315 mm. Clearly, the leakage volume is linearly correlated with R_G or H_G; specifically, the leakage volume is negatively correlated with R_G and positively correlated with H_G. However, the influence of both factors on the leakage volume is very small, varying between 4.534 and 4.546 mL/min. This is consistent with the previous conclusion. From the above, it can be deduced that choosing a smaller H_G and a larger R_G can minimize the leakage.



(a) (b)

Fig. 15 Variation of particle volume fraction and leakage volume with H_G and R_G (W_G=0.315 mm). (a) Particle volume fraction, (b) Leakage volume

(3) Determination of the parameters of the PEG

The optimization objective is to minimize the leakage and maximize the particle volume fraction at the Tri-PEG. The influence of the design parameters on the leakage and the particle volume fraction exhibits different positive and negative correlations. Therefore, a multi-objective genetic algorithm was employed to balance the optimal solutions among various responses and find the Pareto solution (Deb et al., 2020). The optimized parameter values of the Tri-PEG's structures obtained from this approach are shown in Table 5. Compared with the rectangular PEG structure, the optimal parameter combination of the Tri-PEG reduces the leakage by 12% and increases the particle volume fraction by 6%. Therefore, the optimized Tri-PEG structure can control the leakage volume of the fit clearance and the degree of dirt accumulation at the bottom of the groove. Accordingly, it can reduce the possibility of hydraulic spool valve sticking.

Table 5 The results of the multi-objective optimization

Variable	PEG	Tri-PEG	Percentage
W_G (mm)	0.265	0.215	-
H_G (mm)	0.3	0.392	-
R_G (mm)	-	0.2	-
Q_S (mL/min)	4.26	3.76	-12%
c_s (%)	6.27	6.65	6%

4 Conclusions

This study involved a systematic investigation of the mechanism of valve core sticking formation from the perspective of the unbalanced radial force and solid particle intrusion into the fit clearance. The effects of inclination angle, clearance size, and particle diameter were analyzed. We proposed a novel PEG structure, and further adopted multi-objective optimization to obtain optimal structural parameters.

Through simulations, it was revealed that the radial fluid force is positively correlated with the inclination angle, and negatively correlated with the clearance. When the particle diameter reaches a certain value, the volume fraction of solid particles in the sensitive sticking area is relatively small. The introduction of a PEG into the valve core is beneficial for reducing the uneven distribution of pressure in the fit clearance, and can slow down the aggregation of particles in the sensitive sticking area. The particles are mainly concentrated in the PEG, which helps to reduce the occurrence of valve core sticking.

The groove width (W_G) of the triangular PEG with an arc-shaped bottom (Tri-PEG) was revealed to be the main factor affecting the leakage at the fit clearance. The groove depth (H_G) and arc-shaped radius (R_G) have positive and negative effects on the leakage rate, respectively, but their impacts are relatively small. Compared with the rectangular PEG structure, the optimal Tri-PEG structure reduces the leakage by 12% and increases the volume fraction of particles in the groove by 6%. It can therefore control the leakage at the valve core clearance and the degree of particle concentration at the bottom of the groove.

Moreover, the optimized Tri-PEG structure effectively reduces the unbalanced radial force by homogenizing the pressure distribution in the fit clearance, while simultaneously acting as an efficient particle trap to mitigate sticking risk. However, it should be noted that the optimization results and conclusions of this study were obtained under specific simulation operating conditions. If there are significant changes in the working fluid and operating conditions, then the optimal structural parameters of the Tri-PEG structure may change.

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Author contributions

Zhen-hao LIN designed the research. Zhen-hao LIN processed the corresponding data. Zhen-hao LIN, Yu-wei WANG and Zhe-hui MA wrote the first draft of the manuscript. Tian-xiao ZHANG helped to organize the manuscript. Zhi-jiang JIN and Jin-yuan QIAN revised and edited the final version.

Conflict of interest

Zhen-hao LIN, Yu-wei WANG, Zhe-hui MA, Tian-xiao ZHANG, Zhi-jiang JIN, and Jin-yuan QIAN declare that they have no conflict of interest.

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Electronic supplementary materials

Figs. S1 and S2

中文概要

题　目：液压滑阀阀芯均压槽非对称间隙流动分析及抑制卡滞的优化结构

作　者：林振浩[1,2]，王雨薇[1]，马哲辉[1]，张天晓[1]，金志江[1,3]，钱锦远[1,4]

机　构：[1]浙江大学，能源工程学院特种装备研究所，中国杭州，310027；[2]阀源智能科技（杭州）有限公司，中国杭州，310058；[3]浙江大学，温州研究院，中国温州，325036；[4]浙江大学，流体动力基础件与机电系统全国重点实验室，中国杭州，310058

概　要：液压滑阀是航空航天液压系统中的关键控制部件。然而，复杂的工作环境会导致阀芯卡滞，从而限制此类阀门的性能。这会阻碍液压油的精确控制，降低液压系统的稳定性，从而导致航空航天系统发生严重事故。不平衡的径向力和固体颗粒侵入配合间隙是导致这种粘附的主要因素。为了更好地理解这些问题，在这项研究中，我们数值模拟了阀芯间隙内的流体动力学和颗粒行为，并分析了倾斜角度、间隙尺寸、颗粒直径和均压槽（PEG）特性的影响。揭示了阀芯卡滞的机理，发现PEG对不平衡径向力和颗粒侵入具有抑制作用。此外，我们提出了一种弧形底部的三角形均压槽（Tri-PEG）优化结构。通过多目标优化确定结构参数，以间隙处的泄漏最小化和Tri-PEG底部的颗粒体积分数最大化为优化目标。最佳参数为0.2 mm的弧形半径、0.392 mm的槽深和0.215 mm的槽宽。与矩形PEG相比，Tri-PEG最优结构的泄漏量减少了12%，颗粒浓度增加了6%。总的来说，这些研究发现为缓解滑阀卡滞提供了重要参考。

关键词：滑阀；流量特性；多目标优化；固液两相流；航空航天液压系统；阀门卡滞