Research article

Flow force research and structure improvement of cartridge valve core based on CFD method

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ABSTRACT

The hydraulic force has a great negative effect on the cartridge poppet valve system. Based on the law of momentum, the calculation formula of flow force of outflow poppet valve is modified, and a new valve core structure is designed. The compensation effect of the improved main valve core structure on the hydraulic force is discussed; secondly, CFD simulation is carried out to obtain the influence rules of these parameters on hydrodynamic forces. According to the analysis, the influence of main valve core arc structure on hydrodynamic force compensation under different opening degrees is also studied; then the optimal parameters of the arc structure are obtained through analysis. AMEsim system simulation model and test-bed are built to verify the hydrodynamic formula and simulation results. The experimental results verify that the new valve core structure has a good hydrodynamic compensation effect.

Keywords:
Flow force compensation
CFD simulation
Compensation characteristics
Momentum theorem

1. Introduction

The magnitude and direction of hydrodynamic force and the influence of hydrodynamic force on the working reliability, operating flexibility and dynamic and static characteristics of valve, and visual calculation of internal flow field have been basic research problems in the field of fluid transmission and control technology, and also the key problems that must be solved in the development of high-performance hydraulic pressure control valves [1]. As the main structural form of cartridge valve [2], the core valve is found that the existing theory needs to be further studied and solved. Zhao [3] has analyzed and calculated the transient hydrodynamic force, steady hydrodynamic force, pressure and flow rate of the conical valve with conical valve seat and valve core. Zhang et al. [4] studied the radial hydrodynamic force under the control surfaces of the valve and obtained lateral force on spool increases with the pressure, and could reach to the maximum. Lv used CFD simulate fluid flow field and studied fluid characteristics under different fluid conditions [5]. Based on the Fluent flow field simulation software, Qian carried out numerical simulation and analysis on hydrodynamic force on valve core and pressure distribution on valve core under conditions of outflow and inflow of poppet valve. Research has proved that the hydraulic force is proportional to the valve port pressure difference; when valve port pressure difference is greater than 2.5 MPa, negative pressure appears on surface of valve core, and gas etching occurs at valve port; when the valve port opening is 1 mm, the steady-state hydrodynamic force is the largest; under other conditions being equal, the hydraulic force flowing out of the poppet is greater than the hydraulic force flowing in [6, 7]. Si establishes the fluid characteristic model of cone valve including transient and steady-state hydrodynamic forces. Based on this model, the finite element method is used to deduce the conclusion that the diameter of spool has little influence on hydrodynamic force. The correctness of the established model is verified by experimental comparative study, which provides a favorable criterion for the design of cone relief valve [8, 9, 10, 11]. Gao studied the effects of inlet velocity, outlet pressure, opening size, and lift angle on the cavitation strength of poppet valves [12]. S. Hayashi [13] et al. modeled and simulated the instability of the pilot poppet valve circuit and carried out experimental verification. Javad taghinia seyedjalali [14] simulated the finite element fluid model of three types of seat valves, and gave the analysis of valve core flow field at bottom; Fu [15] et al. Carried out the numerical model of hydraulic poppet valve in the laminar flow state of two-dimensional incompressible inviscid flow of poppet valve, gave the distribution law of liquid pressure and velocity in poppet valve cavity, also obtained the total force on spool by integrating the pressure at spool bottom; Yao [16] also used CFD technology to compare and analyze the flow characteristics and stress on main spool of new main valve sleeve structure with

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circular through hole of the traditional plug-in proportional throttle valve and the main valve sleeve with waist through hole. It is pointed out that the main valve sleeve with waist through hole structure can increase flow capacity of poppet valve, and help to improve radial unbalanced force on the main valve core. Yu [17] qualitatively discussed the direction of steady-state hydrodynamic force by using momentum theorem and Bernoulli equation. Zhang [18] expounded the basic concept and calculation method of hydrodynamic force of hydraulic valve from the perspective of dynamics.

Cartridge valve is in widespread use in many engineering fields due to its strong flow capacity, high-frequency response and low hydraulic resistance. Therefore, great attention should be paid to the study of compensation for hydrodynamic force of cartridge valves. This study mainly studies the outflow cone valve, through the modification of the hydrodynamic formula and the CFD simulation of the main valve core structure, successfully realizes the structural compensation of flow force through the correction of hydrodynamic formula and CFD simulation.

2. Analysis of steady-state flow force theory

The flow coefficient of the cone valve is an indicator to measure the flow capacity of the valve. A large value of the flow coefficient indicates that the flow capacity of the valve is large and the pressure loss when the fluid flows through the valve is small, the flow coefficient [7] can be expressed by the following formula (1):

\[ C_d = \frac{Q}{A(x)\sqrt{\frac{2\Delta P}{\rho}} \sqrt{f}} \]  

(1)

\( \Delta P \) is inlet outlet pressure difference, \( Q \) is flow, \( \rho \) is density of medium, \( A(x) \) is orifice flow area [7], as shown in the following formula (2):

\[ A(x) = \pi dx \left[ 1 - \frac{x \sin (2\alpha)}{2d} \right] \sin \alpha \]  

(2)

\( x \) is opening value of spool, \( \alpha \) is the half angle of top angle of the poppet spool, \( d \) is the diameter of inlet. Combining formula (1), (2), and (3) can be obtained:

\[ C_d = \frac{Q}{\pi dx \left[ 1 - \frac{x \sin (2\alpha)}{2d} \right] \sin \alpha \sqrt{\frac{2\Delta P}{\rho}}} \]  

(3)

The essence of flow force is that the pressure distribution of spool bottom is different from that in the static state of the valve. The parts with different pressure distribution and static pressure distribution of spool bottom are the main areas causing the flow force of spool. According to the selection principle of control volume, when deriving the calculation formula by using the momentum theorem, the main area of pressure change must be included when selecting the control volume. For outflow poppet valve, the control volume selected in the derivation of the traditional formula takes the throttle port as the interface, including only the bottom of the valve core at the upstream of the overflow section. When the poppet valve is at rest, the pressure change in the area where the static pressure at the bottom of spool is the outlet pressure is small, and the area where the static pressure is the inlet pressure has changed greatly. Therefore, the selection of control volume needs to include the bottom of spool corresponding to the inlet pressure area when the valve is at rest, that is, most of the valve core area, including the upstream and downstream positions of the throttle port. In this paper, the rotating body of ACDQPHGMB around axis HP is selected as the control volume, including the upstream and downstream areas of the throttle, and the control volume is divided into zone I and zone II as shown in Figure 1 to derive and modify the hydrodynamic formula under the external flow

Figure 1. Structural diagram of poppet valve orifice

Figure 2. Disassembly drawing of cartridge cone valve.

Figure 3. Flow channel 3D model of cartridge valve.
condition. Momentum Eqs. (4) and (5) of zone I and zone II are as follows:

\[ \rho Qv_2 \sin \theta - \rho Qv_1 = F_1 - F_z - F_i \]  
(4)

\[ \rho Qv_2 \sin \theta - \rho Qv_1 = F_z - F_2 - F_i \]  
(5)

It is specified that the upward direction is positive, \( v_1 \) is the average flow velocity on the division cutting surface of I and II, \( v_1 \) is the valve inlet speed, \( v_2 \) is the valve outlet speed, \( \theta \) is the angle between the valve outlet speed and the horizontal direction, \( F_1 \) is the force acting on the liquid at the inlet section, \( F_2 \) is the force acting on the liquid at the outlet section, \( F_z \) is axial force at the bottom of spool of zone I facing the liquid, \( F_2 \) is axial force at the bottom of spool of zone II facing the liquid, \( F_i \) is the force on the liquid at the overflow section.

Combining Eqs. (4) and (5), the momentum Eq. (6) of the liquid in the whole control volume is obtained as follows:

\[ \rho Qv_2 \sin \theta - \rho Qv_1 = F_1 - F_z - F_i - F_2 \]  
(6)

\( F_i \) is the static pressure of the valve core subjected to the liquid as shown in Eq. (7), \( F_s \) is total axial force of spool as shown in Eq. (8). \( F_p \) is hydrodynamic force on spool as shown in Eq. (9).

\[ F_i = p_1 \pi \left( d_f^2 - d_s^2 \right) / 4 + p_2 \pi d_s^4 / 4 \]  
(7)

Among them, \( d_f \) is the valve core diameter, \( d_s \) is the through hole diameter of the valve seat.

\[ F_s = F_i + F_2 \]  
(8)

The flow force on the valve core is:

\[ F_p = F_z - F_z = F_i = F_2 - F_2 \]  
(9)

\[ = \rho Qv_2 \sin \theta - \rho Qv_1 - F_2 + F_2 + p_1 \pi \left( d_f^2 - d_s^2 \right) / 4 + p_2 \pi d_s^4 / 4 \]
Figure 6. Simulation results. (a) Axial force on valve core surface; (b) Flow force.

Figure 7. Pressure nephogram. (a) Original structure; (b) $L = 0.3$ mm; (c) $L = 1.1$ mm; (d) $L = 1.5$ mm.
The hydraulic force can be obtained by combining Eq. (7) and Eqs. (1), (2), and (3), as shown in the following formula (10):

\[
F_p = \frac{F_s}{C_0} \frac{F_{II}}{C_0} \frac{F_{I}}{C_0} \frac{\rho}{C_{18}} \frac{\pi dx}{C_{19}} \frac{\sin \left(2 \alpha \right)}{2 \sqrt{\Delta P}} \frac{\rho_s v_1^2 \sin \theta}{C_0} \frac{\rho_C \pi dx}{C_{18}} \frac{1}{C_0} \frac{x}{C_{19}} \frac{\sin \alpha}{\sqrt{2 \Delta P}} \frac{\pi d_f}{C_{17}} \frac{4}{C_0} \frac{\pi d^2_f}{C_{17}}
\]

In the calculation formula (10), all items are the flow parameters on inlet and outlet boundaries, and the flow parameters on the surface can be considered uniformly distributed. From the calculation formula (10) of the axial steady-state hydraulic force of the hydraulic poppet valve spool obtained from the above analysis, it can be found that the steady-state hydraulic force is coupled by the structural parameters, opening, inlet and outlet speed, axial component of outlet speed and other working parameters of the hydraulic valve.

3. Geometric model of poppet valve

Cartridge valve structure refers to the components composed of valve core, valve sleeve and auxiliary springs and seals. The valve core has simple structure, sensitive action and good sealing performance. Its function is relatively simple. It mainly realizes the opening or closing of the fluid circuit. When it is combined with the ordinary hydraulic control valve, it can control flow state of the system oil. The internal dimensions of the components, including the shape of the valve core, the shape and size of the matching part between the inner hole of the valve sleeve and the valve core, and the size of the spring, shall be determined by the designer. In this paper, the parametric modeling of the standard cartridge valve with half cone angle of 30° and diameter of 20 mm is carried out by
using the software UG. Figure 2 shows physical disassembly of cartridge valve, and Figure 3 shows the flow channel model of the cone valve.

4. Valve core geometry and runner model

4.1. Valve core geometry

Figure 4 is the new poppet spool designed in this paper. Based on the original valve core, the surface of upstream and downstream sections of the valve element is designed as a convex arc surface; the downstream arc-shaped spool surface changes the shape of the flow passage, so that the liquid flow direction flowing through the circular arc structure can be changed without changing the oriifice area of each opening of the spool and affecting the flow coefficient of poppet valve, enlarging the included angle between outlet speed and horizontal line, that is, increasing the axial component of the outlet speed. Based on formula (10), the hydrodynamic force can be reduced; at the same time, the circular arc structure changes the area and position of the high-speed region of the valve core when the poppet valve is closed, and L is the distance between point n and point m, β is the angle between tangent direction and radial direction of arc starting point.

4.2. Meshing

The flow field model is divided into grids in Fluent [19], and the grid is densified at the throttle, and the boundary layer is set, as shown in Figure 5.

4.3. Simulation conditions and parameter settings

The critical Reynolds number of poppet valve port is 20–100 [20, 21], in actual operation, the Reynolds number of the poppet valve is far greater than its critical Reynolds number [22, 23], so the fluid flow state in the cone throttle valve is turbulent. Considering the requirements of simulation method for computer and calculation accuracy, the viscous model in Reynolds average method is selected for steady-state hydrodynamic numerical analysis in this paper. In fluent, the double equation of viscosity model has standard $k$ – $ε$ model, RNG $k$ – $ε$ Model and realizable $k$ – $ε$ Model three models. RNG $k$ – $ε$ model adds correction parameters, considers the influence of small vortex motion on fluid motion, and improves the accuracy of numerical calculation. Therefore, RNG $k$ – $ε$ model in turbulence is selected in this article, the steady-state hydrodynamic force of conical throttle valve is numerically analyzed by model.

The transport equation [24] of RNG $k$ – $ε$ model is shown in Eqs. (11) and (12):

$$
\frac{∂}{∂t}(\rho k) + \frac{∂}{∂x_i}(\rho ku_i) = \frac{∂}{∂x_j}
\left[\left(\rho + \frac{μ_t}{σ_t}\right)\frac{∂k}{∂x_j}\right] + G_k + G_b - ρε - Y_M + S_k
$$  (11)

$$
\frac{∂}{∂t}(ρε) + \frac{∂}{∂x_i}(ρεu_i) = \frac{∂}{∂x_j}
\left[\left(μ + \frac{μ_t}{σ_t}\right)\frac{∂ε}{∂x_j}\right] + C_{1ε}\frac{ε}{k}(G_k + C_aG_b) - C_{2ε}\frac{ε^2}{k} + S_ε
$$  (12)

where $G_k$ represents the average velocity gradient caused by the turbulent kinetic energy production term mean; $G_b$ represents the turbulent kinetic energy generated by buoyancy; $Y_M$ represents the contribution of wave expansion to the total dissipation rate in compressible turbulence; $C_{1ε}$, $C_{2ε}$ and $C_a$ refer to model constants; $σ_t$ and $σ_ε$ represent the turbulent Prandtl numbers for $k$ and $ε$: $S_k$ and $S_ε$ stand for user-defined source terms. The following values are used in Fluent:

$$
C_{1ε} = 1.42, C_{2ε} = 1.68, C_a = 0.09, σ_t = 1.393, σ_ε = 1.393.
$$

Where $μ_t$ is turbulent viscosity, which can be expressed as Eq. (13):

$$
μ_t = μC_0\frac{k^2}{ε}
$$  (13)
This paper sets fluid medium density in the valve chamber to $870 \text{ kg/m}^3$ and dynamic viscosity to $0.04 \text{ Pa}\cdot\text{s}$ [25, 26]. Considering that the hydraulic force of cartridge valve is related to the pressure difference, and according to the working environment of the cartridge valve, the inlet pressure of the valve is $2 \text{ MPa}$, the outlet pressure is $1 \text{ MPa}$, and the pressure difference is $1 \text{ Mpa}$.

5. Simulation results and analysis

5.1. Compensation effect of parameter $L$

5.1.1. Compensation effect of parameter $L$ of fixed valve core opening on flow force

Set spool opening $D = 0.3 \text{ mm}$, $H = 0.1 \text{ mm}$, $\beta = 60^\circ$ unchanged, $L$ changes within the range of $0.3$–$1.5 \text{ mm}$, every $0.4 \text{ mm}$ is a measuring point. CFD simulation of cartridge valve flow channel model with flange structure and original valve core structure was carried out. According to the simulation results shown in Figure 6(a) and Figure 7, the pressure in the upstream area of the valve element changes little, always $2 \text{ MPa}$, and the pressure on the arc surface of the valve element in the downstream area gradually increases with the increase of $L$, so the axial force on the bottom surface of the valve element gradually enlarges with the increase of $L$, according to the calculation formula (9), the hydraulic force value shown in Figure 6(b) can be calculated, and the hydraulic force gradually reduces with the increase of $L$.

According to velocity nephogram analysis in Figure 8, when the arc size is from original to $L = 0.3 \text{ mm}$, velocity downstream of the flow passage of the orifice and the flow channel decreases slightly, and the high-speed area of the fluid becomes smaller. Based on Bernoulli equation, flow rate increases, pressure decreases, as the same time, the axial force of liquid on the spool decreases, so the hydrodynamic force decreases; when changing from $L = 0.3 \text{ mm}$ to $L = 1.1 \text{ mm}$ and $L = 1.5 \text{ mm}$, the velocity of fluid in the flow channel further decreases, and affected by the arc structure, the outflow direction of fluid in the downstream shifts to the horizontal direction, and the partial velocity of fluid in the vertical direction decreases. With the increase of $L$, the high-speed region of fluid gradually spreads from the orifice to the downstream area away from the valve core surface. According to Bernoulli equation, the axial acting pressure of the liquid on the spool enlarges, which leads to gradual decrease of the hydraulic force, and the hydraulic force compensation effect is better; Therefore, with the increase of the size of the circular arc $L$, the better the radial diversion of the liquid and the better the compensation effect of the hydrodynamic force.

5.1.2. Influence of different opening on $L$ hydrodynamic compensation

For the arc size $H = 0.1 \text{ mm}$, $\beta = 60^\circ$, simulate the valve cores with parameters $L$ of $0.3 \text{ mm}$, $0.7 \text{ mm}$, $1.1 \text{ mm}$ and $1.5 \text{ mm}$ at the opening of $0.1$–$0.5 \text{ mm}$, simulate and calculate the steady-state hydrodynamic force in various opening, which are clear in Figure 9. The flow force of the original cone spool enlarges linearly with growing opening value, which is in accordance with derived theory; the flow force of the innovative spool is measurably smaller than flow force of original spool; when $L$ is $0.3$–$1.1 \text{ mm}$, with the increase of opening degree, the area of overflow section increases, the flow through overflow section increases in unit...
time, the circular arc structure of poppet valve core acts on more fluid, the influence of the arc surface of the valve element on the radial direction of the fluid increases, and the amount of hydrodynamic compensation enlarges with increasing opening degree; when L is 1.5 mm and the opening is small, the compensation effect is poor. This is because firstly, the orifice area is too small and the flow is too small. Secondly, the blocking effect of too large L on low flow fluid is too large, resulting in the high-speed area of fluid close to the valve core and I area away from the arc structure, so the compensation effect is reduced.

5.2. Compensation effect of parameter H

5.2.1. Compensation effect of parameter H of fixed valve core opening on hydrodynamic force

Set spool opening D = 0.3 mm, L = 1.1 mm, β = 60° unchanged, H changes within the range of 0.1–1.3 mm, every 0.3 mm is a measuring point. According to the simulation results in Figure 10(a), the stress on upstream and downstream surface of spool first increases and then decreases with the increase of H, the hydraulic force first enlarges, then decreases with increasing L in Figure 10(b). Figure 11 shows the pressure cloud diagram of upstream and downstream surface of spool. It can be seen that when H = 0.1 increases to H = 1, the stress on upstream and downstream surface of the spool enlarges continuously, resulting in the increase of hydraulic force.

According to the velocity Cloud Figure 12, compared with the original structure, the outflow velocity direction of the liquid guided by the arc structure is more inclined to the horizontal direction when H = 0.1–1 mm, which lead to the decrease of the velocity component in the vertical direction. With the increase of H from 0.1 mm to 1 mm, the area of the high velocity region of the liquid gradually decreases, and the average velocity gradually decreases with the increase of H, according to the Bernoulli equation, the decrease of the fluid velocity leads to the increase

![Figure 12. Velocity nephogram. (a) Original structure; (b) H = 0.1 mm; (c) H = 0.7 mm; (d) H = 1 mm.](image-url)
of the fluid pressure on the valve core, so the hydraulic force is reduced; when $H$ is 1.3 mm, the arc structure is too far away from the throttle port, and the arc area becomes very small, which reduces the diversion effect of improved structure on the liquid, so the compensation effect is not as good as when $H = 1$ mm.

5.2.2. Influence of different opening on $H$ hydrodynamic compensation

When circular arc size $L = 1.1$ mm, $\beta = 60^\circ$, simulate the valve core with parameter $H$ of 0.1 mm, 0.4 mm, 0.7 mm, 1 mm and 1.3 mm at opening of 0.1–0.5 mm. The hydrodynamic compensation effect under different opening is observed. Figure 13 shows that under the opening degree of 0.1–0.5 mm, the hydrodynamic force of spool with different $H$ values is obviously smaller than the original spool structure; with the increase of the opening, the larger the $H$ value, the better the hydraulic force compensation effect.

Comprehensively considered, when $H$ is 1 mm, the arc compensation effect is the best.

5.3. Compensation effect of parameter $\beta$

5.3.1. Influence of arc parameter $\beta$ of fixed valve core opening on flow force

Set spool opening $D = 0.3$ mm, $L = 1.1$ mm and $H = 1$ mm unchanged, $\beta$ changes within the range of $60^\circ$–$75^\circ$, every $5^\circ$ is a measuring point. According to the simulation results in Figure 14(a) and pressure cloud Figure 15, the stress on upstream and downstream surface of spool

![Figure 13. Hydraulic force of valve element with different opening and $H$.](image)

![Figure 14. Simulation results. (a) Axial force on valve core surface; (b) Flow force.](image)

![Figure 15. Pressure nephogram. (a) Original structure; (b) $\beta = 60^\circ$.](image)
increases with $\beta$ and flow force changes little with the increase of $L$, as shown in Figure 14(b).

According to the velocity cloud Figure 16, the increase of the arc angle basically has no effect on the fluid outflow velocity angle and the area of the fluid high pressure area. Therefore, the arc structure radian $\beta$ has little effect on hydraulic force compensation.

5.3.2. Influence of different opening on $\beta$ hydrodynamic compensation

When the arc size $L = 1.1$ mm and $H = 1.0$ mm, simulate the valve cores with parameters $\beta$ of $60^\circ$, $65^\circ$, $70^\circ$ and $75^\circ$ respectively under the opening of $0.1$–$0.5$ mm, and the hydrodynamic compensation effect under different opening volume is observed. Figure 17 shows under different opening volume, different $\beta$, the flow force of improved valve core is obviously smaller than that of the original structure; with increase of opening, the compensation effect relative to the precursor structure is better and better; change $\beta$ It has little effect on the compensation effect of flow force.

Considering the convenience of mass production and processing of valve core, $\beta = 60^\circ$ is the optimal size.

5.4. Verification of mesh independence

In this paper, the quadrilateral grid is used to discretize the poppet valve calculation model, and the number of grid elements will inevitably affect the results obtained by Fluent. Therefore, in order to verify the mesh independence of the numerical simulation model of the conical throttle valve, the mesh models of 0.3 million, 0.6 million, 0.8 million and 1.4 million are selected for simulation. Figure 18 shows the verification results. It can be seen from the figure that when the mesh quality is 0.3 million, the simulation results are quite different from those of 0.6 million, 0.8 million and 1.4 million, but there is no obvious difference between the simulation curves of 0.6 million, 0.8 million and 1.4 million. It shows that the result of 0.8 million mesh selected in this paper is accurate and reliable.

6. Experiment and verification

The test bench is composed of pressure sensor, data acquisition card, driving device, displacement sensor, pressure gauge, pilot valve and
hydraulic pump. Figure 19 shows the schematic diagram of test principle, and test platform is shown in Figure 20.

In addition, in order to verify the accuracy of the hydrodynamic derivation formula, based on actual performance and dimensions of experimental poppet valve, the hydraulic valve is customized by using the hydraulic component design library (HCD) in AMEsim, and the hydraulic system model with the same test conditions is built. The deduced poppet valve hydrodynamic formula is brought into the built AMEsim model, the boundary condition parameters of the simulation are the same as the test process, and the hydrodynamic value is calculated by simulation. Figure 21 shows the system simulation model.

The valve core with the optimal structure $H = 1 \text{ mm}$, $L = 1.1 \text{ mm}$ and $\beta = 60^\circ$ is tested and simulated. The experimental, simulation and calculation results are shown in Figure 22. The experimental curve is in good agreement with the calculation curve and simulation curve, which verifies the correctness of the hydrodynamic formula of the poppet valve and the correctness of fluent simulation.

For purpose of better comparing hydraulic force compensation effect of innovative valve core structure, set the ratio of hydraulic force and opening as the compensation coefficient $\delta$, $\delta$ can be expressed as Eq. (14):

$$\delta = \frac{\Delta F_p}{\Delta D}$$  \hspace{1cm} (14)

The smaller the ratio of hydraulic force and opening, the better the compensation effect and the more stable the poppet valve. The experimental values in Figure 18 are solved to obtain the compensation coefficients under different opening degrees in Table 1. With the increasing value of the opening degree, the absolute value of compensation coefficient of the innovative spool becomes smaller and smaller; when the opening is 0.5 mm, the absolute value of the improved spool
compensation coefficient 40.3 is 64.14% lower than the absolute value of the unmodified spool compensation coefficient 112.4, and the arc structure has the best compensation effect.

7. Conclusions

According to the law of momentum, this paper deduces and modifies the hydrodynamic force formula, and proposes a structure for compensating the hydrodynamic force of the cartridge cone valve. The flow force of the original valve core under different flow rates and valve core positions is obtained, and several parameters that may affect the hydrodynamic characteristics are studied. Then, the influence of each parameter on the hydrodynamic characteristics is explored:

1. As the increase of the opening degree of the valve core, the arc surface structure of the designed valve core has better compensation effect;
2. Keep $H$ and $\beta$ unchanged, when the spool opening value is small, the hydrodynamic compensation effect first enlarges and then reduces with the increase of $L$; when the opening value of the spool is large, the hydrodynamic compensation effect is greater with the increasing value of $L$;
3. Keep $L$ and $\beta$ unchanged, the hydrodynamic compensation effect first increases and then decreases with the increase of $H$.
4. Keep $H$ and $L$ unchanged, the hydrodynamic compensation effect basically has no change with the increase of $\beta$.

Based on the optimal size of circular arc structure, the independence of mesh generation is verified, and a test-bed is built for verifying the reliability and accuracy of simulation results; the AMESim system simulation model is built for verifying the accuracy of derived hydraulic floor formula, which is helpful to the future research on hydrodynamic force of the external flow cone valve. The compensation coefficient is defined and calculated. Research results show that under the optimal size, the hydrodynamic compensation effect of the valve core increases with the increase of the opening degree. When the opening degree is 0.5 mm, the absolute value of the compensation coefficient decreases by 64.4%, and the compensation effect is the best. Therefore, the method proposed in this paper is feasible. The research results can provide a theoretical basis for the structural optimization of cartridge cone valve, the innovative valve core structure in this paper is helpful to improve the control accuracy of the conical valve, reduce the energy loss of the hydraulic system, and make it more suitable for the increasingly complex hydraulic system.
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