Research Article

Investigation on the Characteristics of a Combination Microflow Control Valve

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Flow control valves have broad application prospects in aviation hydraulic systems. This paper proposes a combination microflow control valve (CMCV) instead of the traditional valve to optimize the performance. Influences of structural parameters of CMCV on its characteristics are numerically investigated to determine the static and dynamic characteristics of CMCV. The calculation results indicate that there is a negative feedback control between stages of the flow regulator, the orifice pressure drop is compensated, and the flow regulation deviation is reduced. The orifice area and the flow regulator valve port area have significant effects on flow characteristics. The diameter of orifice, the spring stiffness, the number of throttle holes, and the ultimate displacement of sleeve are positively correlated with the flow rate stability value of the valve. The valve port flow area gradient and initial overlap of the flow regulator affect the flow rate fluctuation range and response time of CMCV.

1. Introduction

The combination microflow control valves (CMCVs) are widely used in high-pressure and low-flow-rate hydraulic systems, such as the helicopter brake system [1]. The flow and dynamic characteristics of the valve have a significant impact on the hydraulic system’s performance [2–4]. Typically, the working pressure of the hydraulic system varies suddenly, which may cause instability of the system and components. Therefore, the CMCV with rapid response and stable performance is the key component, which largely determines the stability and performance of the entire hydraulic system.

Previously, many scholars have made significant contributions to researching the fundamental characteristics of flow control valves. Wu et al. considered the coupling relationship between the flow field and the spring system in the valve. The indirect CFD method was proposed based on the valve control equation to predict the flow rate-pressure characteristic curve of the pressure control valve [5]. Xiao et al. analyzed the flow characteristics of the digital pilot speed control valve. They used the fixed-rate pilot to control the flow rate to effectively alleviate the problem of excessive pressure loss [6]. Nie et al. studied the flow and noise characteristics of the S-type valve. They applied LES theory to study the velocity and pressure changes of the flow field and the impact on noise problems at different openings [7]. Xie et al. used the fluid mechanics theory to establish a mathematical model under rated load. Then they analyzed the relationship between its flow characteristics and certain parameters and provided a reference for the design and application of valves [8]. The relationship between the performance of the valve and certain parameters was analyzed [9]. Filo and Rajda analyzed the force of the spool mechanical model and made corresponding adjustments. It was determined that the flow field force of the directional valve can be compensated by appropriate spool valve geometry [9]. Qian et al. analyzed the influence of spring stiffness on the flow characteristics and response time of fast-response valves. The research conclusions were used in the design and flow control of the new valve [10]. Okhotnikov et al. studied the steady-state flow state and hydraulic characteristics of the flow control valve, predicted the torque and pressure drop caused by steady-state flow, and improved
the controllability of the valve [11]. The simulation models and control algorithms were used to analyze the valve flow characteristics, which provide a reference for valves structure design and flow control. Wang et al. established the linear mathematical model of the valve and used PID to control the flow and conducted real-time predictive feedback. This method effectively increased the pressure drop in the main orifice and kept the flow constant [12]. Xie et al. analyzed the static flow control performance model based on the hydraulic half-bridge principle and thus determined the displacement characteristics of the two keyholes and realized proportional flow control [13].

The relationship between the structural parameters and fundamental characteristics of the flow control valve has attracted increasing attention in recent years [14, 15]. The fluid characteristics of the valve were investigated, and the influences of the structural parameters such as valve port area, groove width, and spring stiffness on the valve characteristics were analyzed. The valve had better flow characteristics and dynamic response performance by improving the structural parameters [16, 17]. The above results indicated that the dynamic characteristics of the flow control valve are directly affected by the structural parameters. Apart from the structural parameters, several scholars have also reported many flow control valves with special spool structures [18, 19]. Lisowski et al. determined the flow characteristics of the proportional flow valve and compared the flow rate and pressure of the circular and triangular valve ports under different throttle gap widths. They improved the flow rate by modifying the valve geometry [20, 21].

Previous research has focused on the flow characteristics of different types of valves. However, there is no sufficient research to prove the influence of the microflow valve structure on the flow characteristics. This paper proposes a combination microflow control valve, which can significantly realize the integration and lightweight design of the flow control valve. From the abovementioned researches’ description, the static characteristics and dynamic response performance of the valve are directly affected by the structural parameters.

To analyze the relationship between performance and structural parameters of CMCV, the main content of this paper is arranged as follows. In Section 2, the structure of the CMCV is introduced, and its advantages are pointed out. In Section 3, the valve port area of the movable sleeve is calculated, and the mathematical and simulation model of the CMCV is established. Dynamic characteristics of the flow regulator are researched, and the accuracy of the model is verified by using static characteristics experiments. In Section 4, the influence of valve structural parameters on the fundamental characteristics is numerically investigated. The sensitive factors and laws that affect the flow characteristics and dynamic response performance of the CMCV are determined. Finally, we summarize some conclusions of this paper in Section 5.

2. Structure of CMCV

2.1. Description of CMCV. The CMCV studied in this paper comprises the check valve and flow regulator. Figure 1 shows a schematic of the CMCV. The flow regulator includes the flow control unit and orifice in series, realizing the constant flow control function. The design of the flow control unit represents a normally closed two-directional flow control valve with a fixed spool, a movable sleeve, and a spring return mechanism to keep the valve fully open in the initial position. The structure of CMCV is shown in Figures 1(a) and 1(b). Applying terms and symbols of GJB 1482-2009, the graphical symbols for the valve are drawn in Figure 1(c).

As shown in Figure 1(d), $P_1$ is the inlet pressure, $P_2$ is the pressure after the fluid passes through the valve port of the flow regulator, and $P_3$ is the outlet pressure of the valve. The movement of the sleeve realizes the flow regulator's pressure compensation function. The sleeve is mainly affected by fluid pressures $P_2$ and $P_3$, spring force $F_k$, and hydrodynamic force $F_r$. $F_r$ is negligible in this paper because its direction is related to the movement state of the sleeve.

The diameter of the orifice is constant. The values of $P_1$ and $P_3$ change with the load of the system changes. The orifice pressure drop is kept constant by adjusting the flow regulator, ensuring a constant outlet flow of the valve.

2.2. Calculation of the Opening Area. The flow control unit studied in this paper represents a flow regulator with a movable sleeve and incomplete opening, which directly adjusts the sleeve position by spring. As shown in Figure 2, the flow regulation is attained by adjusting throttle holes of the sleeve opening area near the seat. The opening area of the valve is affected by the number and position of holes and the edge relationship of the sleeve spool.

There are three kinds of initial positional relationships between holes of the sleeve and edges of the spool: The first is over-lap (the left edge partially covers the hole), where the sleeve moves to the right, and the opening area first increases and then decreases. The distance between the axis of the holes and the left edge of the spool is $l_{i1}$, $l_{i2}$, ..., $l_{in}$. The second is under-lap (the right edge partially covers the hole), where the sleeve moves to the right, and the opening area decreases, and the distance between the axis of the holes and the right edge of the spool is $l_{1r}$, $l_{2r}$, ..., $l_{mr}$. The third is zero-lap (the hole is not covered and between the left and right edges), where the sleeve moves to the right, and the valve opening decreases.

When the distance between any hole $i$ and the left edge is $l_{i} \in [0, R]$, the opening area of holes is
\[ s_i = \begin{cases} 
\frac{\pi R^2}{180} \left( \arccos \left( \frac{\left( R - l_{id} + x \right) R}{R} \right) \right) 
+ \frac{\pi R^2}{180} \left( \arccos \left( \frac{\left( X_1 - R + l_{id} - x \right) R}{R} \right) \right) 
+ \frac{\pi R^2}{180} \left( \arccos \left( \frac{\left( x + R - X_1 \right) R}{R} \right) \right) 
\end{cases} \]

\[(1)\]
When the distance between any hole \(i\) and the right edge is \(l_{ir} \in [0, R]\), the opening area is

\[
s_i = \begin{cases} 
\pi R^2 - \frac{\arccos((l_{ir} - x)/R)}{180} \pi R^2 + (l_{ir} - x)\sqrt{R^2 - (l_{ir} - x)^2} & (0 \leq x < l_{ir}), \\
\arccos((x - l_{ir})/R) \pi R^2 - (x - l_{ir})\sqrt{R^2 - (x - l_{ir})^2} & (l_{ir} \leq x < l_{ir} + R), \\
0 & (l_{ir} + R \leq x \leq x_{\text{max}}). 
\end{cases}
\]

When any hole \(i\) is not covered and between the left and right edges and the distance from the right edge is \(l_{ir} \in [R, X_1 - R]\), the opening area is

\[
s_i = \begin{cases} 
\pi R^2 (0 \leq x < l_{ir} - R), \\
\pi R^2 - \frac{\arccos((l_{ir} - x)/R)}{180} \pi R^2 + (l_{ir} - x)\sqrt{R^2 - (l_{ir} - x)^2} & (l_{ir} - R \leq x < l_{ir}), \\
\arccos((x - l_{ir})/R) \pi R^2 - (x - l_{ir})\sqrt{R^2 - (x - l_{ir})^2} & (l_{ir} \leq x < 2R), \\
0 & (2R \leq x \leq x_{\text{max}}). 
\end{cases}
\]

The total opening area of the valve is

\[
s = \sum_{i=1}^{n} s_i,
\]

where \(n\) is the number of holes, \(x\) is the displacement of sleeve, \(R\) is the radius of hole, \(X_1\) is the distance between the left and right edges, and \(x_{\text{max}}\) is the stroke of the sleeve.

3. Description of Models

3.1. The Mathematical Model of CMCV. The following assumptions are accepted in the mathematical model of the valve. The mass of the spool and the leakage of the valve are negligible. The effects of flow force and coulomb friction of the CMCV are ignored.

The flow regulator keeps the flow rate of the hydraulic system stable when the inlet pressure or load pressure changes, and the simplified damping diagram is shown in Figure 3.

The static equation of the sleeve is

\[
A(P_2 - P_3) + F_S = k(x_0 + h - x),
\]

where \(A\) is the effective working area of the flow regulator; \(P_2\) and \(P_3\) are the pressures of the inlet and outlet chambers of the orifice, respectively; \(F_S\) is the steady flow force; \(x_0\) and \(k\) are the spring precompression displacement and stiffness, respectively; \(h\) and \(x\) are the preopening displacement and opening displacement of the flow regulator, respectively.

The flow rate through the valve port of the flow regulator is given by

\[
Q = C_1 A_1 \sqrt{\frac{2(P_1 - P_2)}{\rho}} = C_2 A_2 \sqrt{\frac{2(P_2 - P_3)}{\rho}} = Q_2,
\]

where \(A_1(x)\) is the flow area of the flow regulator valve port; \(\rho\) is the fluid density; \(A_2\) is the flow area of the orifice; and \(C_2\) is the flow rate coefficient of the orifice.

When the sleeve is not moving, the continuity equation of the flow rate of the valve is

\[
Q = C_2 A_2 \sqrt{\frac{2(P_2 - P_3)}{\rho}} \left[\frac{1}{1 + (C_1^2 A_1^2)/(C_2^2 A_2^2)}\right].
\]

The function of the flow regulators is realized by taking pressure compensation measures, which is a dynamic process. According to the actual force acting on the sleeve and Newton’s second law, the force balance equation of the sleeve is
Based on the above, the following inferences are obtained:

1. The outlet flow rate of the valve is mainly affected by the flow area of the flow regulator valve port \( A_1 \) and the flow area of the orifice \( A_2 \).

2. Fluctuations of the outlet flow rate \( \Delta Q_2 \) change with the change of the pressure drop between the inlet and outlet, and the change of the orifice pressure drop \( \Delta (P_2 - P_3) \) changes and forms negative feedback to the system. The sleeve gets pressure compensation \( \Delta (P_1 - P_2) \) to maintain the outlet flow rate fluctuations around a constant value.

3. \( K_{q,s} \) and \( K_{q,P_2} \) are directly proportional to flow area and flow area gradient of the flow regulator, respectively. Reducing gains is helpful to control the fluctuations of the outlet flow rate.

4. Flow pulsation affects the stability and response characteristics of the valve. The undamped natural frequency \( \omega_n \) is higher, and the flow rate is more stable, but the response is slower. According to equation (11), \( K_x \) is directly proportional to the flow area gradient, and the smaller flow area gradient will make the flow rate more stable.

### 3.2 Advantages of CMCV

From the fundamental characteristics, the CMCV may have faster response speed and accurate position control, which can be attributed to three reasons:

1. The sleeve throttling holes' profile is identical in shape to the spool's windows, but the position is staggered. It allows a very smooth change of the opening area when pressure fluctuates.

2. The spool incorporates two sets of circumferential grooves cut on both sides of the windows. The purpose of the grooves is to lower coulomb friction in the spool-sleeve assembly and prevent stagnating of the movable sleeve due to the small radial clearance between the parts.

3. The combined structure reduces the mass and dimension of the valve, and the movable sleeve replaces the traditional spool, which decreases the viscous force, inertial force, and friction.
ΔP1 Δ (P1 – P2) ΔQ1

ΔP3

Kp

Δ(P2 – P3)

1/m j + C j + K = Ksx

AS

ΔQ2

Kq

Figure 4: Schematic of the mathematical model of the flow regulator.

Table 1: Differences between traditional flow valve and CMCV.

| Difference                  | Traditional flow valve | CMCV |
|-----------------------------|------------------------|------|
| Manufacturing and assembly  | Complex                | Simple |
| Type                        | Slide                  | Cartridge |
| Structure                   | Scattered              | Compact |
| Mass                        | Heavier                | Lighter |
| Sleeve                      | Fixed                  | Movable |
| Spool                       | Movable                | Fixed |
| Response speed              | Slower                 | Faster |
| Inertial force, viscous, friction force | Greater | Lesser |

3.3. Simulation Model of CMCV

3.3.1. Simulation Parameter Setting. According to the main structure and working principle of the CMCV, the simulation model is established by AMESim, as shown in Figure 5. The CMCV model is built using the hydraulic component design library. In contrast, other hydraulic components such as the pump, relief valve, motor, and filter are built using the standard hydraulic library.

The main component of the model is the flow regulator, and the flow path is equipped with fixed orifice and control edges. The logic signals of model control edges are determined by the number and the specific position of throttle holes. The left and right edges control the positions of the flow regulator valve ports. The displacement sensor detects the displacement of the sleeve and inputs the value into the logic control module. Logic signals adjust the opening area of the sleeve by the comparator. The simulation parameters are following the test conditions and the specific structure of the CMCV, presented in Table 2.

3.3.2. Model Validation. In order to verify the accuracy of the simulation model, experimental investigations on the CMCV were conducted to measure the static performances, and the hydraulic circuit of the test apparatus is presented in Figure 6 and Table 3. The globe valves K1, K2, and K3 are used to control the on-off of the measuring fluid circuit, and the relief valve F1 is adjusted to ensure that the fluid supply pressure range is 0~21 MPa. Adjust the outlet relief valve F2 to change the load pressure. When the ambient temperature is (25 ± 10)°C, control the hydraulic pump flow to be higher than 2.8 L/min, adjust F1 to make the valve inlet A pressure up to 6 MPa, adjust F2 to make the valve outlet B back pressure to be tested be 0.5 MPa, 1 MPa, 1.5 MPa, 2 MPa, and 6 MPa, and detect and record the output flow of B.

Figure 7 shows the flow rate of port B comparison between the experimental results and the calculated dates. The simulated characteristic curve and stable values agree with the experimental ones, and the maximum relative error is approximately 5.2%. Hence, the simulation dates agree well with the experimental results, and the established simulation model can be considered scientific and reasonable.

4. Results and Discussion

Based on the above, the internal flow characteristics of the flow regulator have a great influence on the function of the CMCV. To further obtain the influence law of the structural parameters of CMCV on the flow and response characteristics, the following main structural parameters of the valve are analyzed: the diameter of orifice d1, spring stiffness k, number n2, and diameter d2 of throttle holes, ultimate displacement of sleeve lmax, and number of the throttle holes axis n1 and number of effective edges n, represent the position and initial overlap of throttle holes, respectively.

4.1. Influence of the Orifice Diameter. The orifice diameter is a critical CMCV parameter that creates a pressure drop between the flow regulator sleeve outlet and the spring chamber. Three diameter values for orifice (d, 1.55 mm, 1.6 mm, and 1.65 mm) are investigated in this research. All other structural parameters are the same as those presented in Table 1 to avoid accidental results. As shown in Figure 8, the inlet pressure has the characteristics of a ramp function, where the pressure increases uniformly over time.

The dotted line in Figure 9 shows the variation in the flow rate of the valve with increasing orifice diameter. It can be seen that as the inlet pressure increases, the initial flow rate increases and remains stable after reaching a certain valve. The stable value increases with increasing d1.

In fact, with increasing d1, the fluid in the sleeve outlet will flow easier into the spring chamber inlet. It will increase the pressure build-up rate in the spring chamber inlet and induce an increase in the pressure drop across orifice. However, it can be seen from Figure 9 that the pressure drop across the orifice (ΔP2,3) is stable at 5 × 10⁻³ MPa. This is because ΔP2,3 saturates in the valve structure studied in this paper.
Based on the above, increasing $d_1$ is one of the better ways to increase the flow rate of the valve but has no significant influence on $\Delta P_{2,3}$.

4.2. Influence of the Spring Stiffness. In order to analyze the influence of spring stiffness independently, the spring pre-compression force is kept constant. Figure 10 shows the pressure-flow characteristic of the valve at the spring stiffness of 1–5 N/m. As can be seen, with the increasing spring stiffness, the initial flow rate increases and remains stable after reaching a certain valve. The stable value increases with increasing spring stiffness. The spring stiffness $k$ of valve has little influence on $\Delta P_{2,3}$.

4.3. Influence of the Throttle Holes Diameter. The diameter of throttle holes $d_2$ is also a critical parameter. Three diameter values for orifice were investigated in this study ($d_2$, 1.0 mm, 1.2 mm, and 1.4 mm). All other structural parameters are the same as those presented in Table 1 to avoid accidental results.

Figure 11 shows the variation in the flow rate of the valve with increasing throttle holes diameter $d_2$. It can be seen that as the inlet pressure increases, the initial flow rate increases and remains stable after reaching a certain valve.

The stable value of flow rate and $\Delta P_{2,3}$ remain constant when $d_2$ is increasing. It is since the displacement of the sleeve is negligible of the CMCV, the change of opening area and flow area gradient is too small to cause no significant variation in the process of pressure regulation.

Table 2: List of simulation parameter.

| Parameter | Value          |
|-----------|----------------|
| $d$       | 1.5~1.7 mm     |
| $d_1$     | 1.2~1.6 mm     |
| $d_2$     | 1.0~1.4 mm     |
| $P_1$     | 0–21 MPa       |
| $P_3$     | 0 MPa          |
| $n_h$     | 1~5            |
| $n_p$     | 1~5            |
| $Q$       | 2.8 L/min      |
| $k$       | 1–5 N/m        |
| $\rho$    | 69°            |
| $\rho$    | 0.86 g/cm³     |
| $m$       | 0.067 g        |
| $n_c$     | 0~2            |
| $l_{max}$ | 1.2~1.8 mm     |
4.4. Influence of Throttle Holes Number. Figure 12 shows the pressure-flow characteristic and pressure drop $\Delta P_{2,3}$ with different numbers of throttle holes ($n_h, 3, 5, \text{ and } 7$). As can be seen, with the increasing inlet pressure, the initial flow rate increases and remains stable after reaching a certain valve. The stable value increases with increasing $n_h$.

In fact, with increasing $n_h$, the opening area of the sleeve increases, and the fluid will more easily go through throttle holes. $\Delta P_{2,3}$ saturates in the valve structure studied in this paper.

Table 3: Primary sensors and model details.

| Description                          | Main features                                      |
|--------------------------------------|----------------------------------------------------|
| Motor                                | B35, 2.2 kW, speed 0–1400 r/min                    |
| Flowmeter                            | VC 0.2, range 0–16 L/min                           |
| Pressure sensor                      | HM20, range 0–250 bar, setting time 1 ms, accuracy 0.5% |
| Data acquisition                     | DEWE-501,32 analogue channels, accuracy 0.1%       |
| Relief valve                         | DBD6, range 0–210 bar                              |
| Globe valve                          | AF6, range 0–300 bar                               |

Figure 7: Comparison of flow rate outputs values between simulation and experimental results.

Figure 8: Pressure in CMCV inlet.

Figure 9: Flow rate and pressure drop across orifice under different $d_1$.

Figure 10: Flow rate and pressure drop across orifice under different $k$.

Based on the above, increasing $n_h$ is a better way to increase the flow rate of the valve but has no significant influence on $\Delta P_{2,3}$. 

4.5. Influence of the Ultimate Sleeve Displacement.

Figure 13 shows the pressure-flow characteristic and pressure drop $\Delta P_{2,3}$ within increasing sleeve ultimate displacement ($l_{\text{max}}$, 1.4 mm, 1.5 mm, and 1.8 mm). As can be seen, as $l_{\text{max}}$ increases, the initial flow rate and pressure drop across orifice increase. This means that the position distribution of the valve holes greatly influences the flow characteristics under the narrow structure, with different positions of the throttle holes. The flow area gradient of the valve port varies greatly. When the sleeve is moved to different positions, the opening area of the valve port is different and causes a stable flow differently.

Based on the above, to increase the stable flow rate of the CMCV, the ultimate displacement of the sleeve should be appropriately increased within the allowable range of the structural space.

4.6. Influence of the Throttle Holes Position.

The position of throttle holes, which acts to create the flow area gradient of the valve port, is a critical parameter. Three values of the throttle holes axis ($n_p$, 1, 3, and 5) were investigated in this study. The distribution of throttle holes axis is shown in Figure 14: single-axis ($n_p = 1$), three-axis ($n_p = 3$), and five-axis ($n_p = 5$) equidistantly distributed holes. To avoid accidental results, number and diameter of throttle holes are constant.

Figure 15 shows the flow area as a sleeve displacement function under three types of valve holes distributions. The curve slope is the flow area gradient in the case of the single-axis distribution of throttle holes; the flow area gradient is small, while the other two types of distribution modes are large and similar.

Establish models of the valve with three types of holes position distributions under 21 MPa rated pressure and 0 MPa backpressure, input different pressure step signals in load port, and the flow rate fluctuation and response time analysis are carried out. Flow rate curves and the statistical results are shown in Figure 16 and Table 4. It is shown that the valve with a single-axis distribution of throttle holes has higher response speed and less overshoot than three-axis and five-axis distributions. Nevertheless, it takes more time to reach a stable flow rate and has more flow regulation deviation. Because the flow area gradient of the valve is small, the opening area changes less under the same displacement of the valve sleeve, and the flow regulation gradient is small so that it can reach the required position faster than other valves under the same step pressure. The higher overshoot and more extended response time can be seen with reducing load pressure. It is a typical underdamped system with a flow pulsation.

Based on the above, it can be concluded that properly adjusting the throttle holes position distribution can maintain a relative balance between the dynamic response
Figure 14: The number of throttle holes axis $n_p = 1$: (a) $n_p = 1$, (b) $n_p = 3$, and (c) $n_p = 5$.

Figure 15: Flow rate and pressure drop across orifice under different $n_p$.

Figure 16: Flow rate across orifice under different $n_p$: (a) $n_p = 1$, (b) $n_p = 3$, and (c) $n_p = 5$. 
speed and the flow regulation deviation, and the deviation can be reduced by sacrificing part of the response time.

4.7. Influence of the Throttle Holes Initial Overlap. The initial overlap of throttle holes greatly influences the flow area gradient of the valve port. The covering relationship between the throttle holes and the number of initial effective control edges \( n_c \) can be seen from Figure 17, which is roughly divided into three types: no control edges \( n_c = 0 \), single-control edge \( n_c = 1 \), and dual-control edges \( n_c = 2 \). To avoid accidental results, the sleeve structure is determined.

Figure 18 shows the opening area as a sleeve displacement function under three types of the covering relationship. The curve slope is the flow area gradient. When the control edge influences the initial overlap of throttle holes, the opening area increases slightly first and decreases with increasing the sleeve displacement. The throttle holes are covered by single-control edge; when the valve works, the valve sleeve moves to the right and opens completely. The increased opening area is more significant than the reduced, and the total opening area has an upward trend. After an absolute displacement, the increased opening area begins to be less than the reduced opening area, and the total opening area begins to decrease. The throttle holes are covered by dual-control edges, affected by the throttle holes position distribution, and the opening area is not reaching the maximum; the range of opening area is too small.

Establish models of the valve with three types of holes position distributions under 21 MPa rated pressure and 0 MPa backpressure, input different pressure step signals in load port, and the flow rate fluctuation and response time.
Flow rate curves and the statistical results are shown in Figure 19 and Table 5. It is shown that the valve with no control edge has higher response speed and less overshoot than single and dual edges. Because the valve has a larger flow area gradient, it can reach the desired position more quickly than the others under the same step pressure. However, it has more flow regulation deviation. With increasing pressure drop, higher overshoot and longer response time can be seen. Based on the above, it can be concluded that properly adjusting the initial overlap of throttle holes can maintain a relative balance between the dynamic response speed and the flow regulation deviation, and the deviation can be reduced by sacrificing part of the response time.

5. Conclusions

As an innovative point of this paper, the paper first proposes the combination microflow control valve and realizes the improvement of static and dynamic response-ability, accuracy, and stability of the hydraulic system. Numerical simulations investigate the influence of structural parameters on the fundamental characteristics of CMCV. The main conclusions can be drawn as follows:

1. The orifice area $A_2$, the flow regulator valve port area $A_1$, and the orifice pressure drop $(P_2 - P_3)$ significantly influence the static characteristics of CMCV, which are reflected in the form of coupling effect. The flow rate of the valve is increased by increasing the orifice pressure drop and reducing area ratio ($A_2/A_3$).

2. The diameter of orifice, the spring stiffness, and the number of throttle holes have significant influences on the flow rate of the CMCV. The ultimate displacement of sleeve significantly affects the flow rate of the valve by directly varying the pressure drop of orifice. The flow rate of the valve decreased when they decreased.

3. Because of the structural constraints of the sleeve, the diameter of throttle holes effects on the pressure drop and flow rate are not as significant as other analyzed parameters.

4. The position and the initial overlap of throttle holes have similar influences on the flow area gradient of the valve port. Their increase caused the flow rate fluctuation range and response time to decrease, but flow regulation deviation increases.

The CMCV can be widely used in aerospace, narrow space equipment, and other directions. Based on the analysis of the paper, find the sensitive factors and rules that affect the fundamental characteristics of the CMCV. The valve structure and research content proposed in this paper will lay the foundation for further developing high-pressure, microflow, and fast-response flow valves and provide new ideas for further developing other flow valves.

### Table 5: The response characteristic of the CMCV with different $n_c, q_{0w}$.

| Pressure step (MPa) | Rise time (ms) | Maximum overshoot (L/min) | Stable time (ms) | Stable flow rate (L/min) |
|--------------------|----------------|---------------------------|-----------------|----------------------------|
|                    | $t_{01}$ | $t_{11}$ | $t_{21}$ | $q_{0max}$ | $q_{1max}$ | $q_{2max}$ | $t_{02}$ | $t_{12}$ | $t_{22}$ | $q_{0}$ | $q_{1}$ | $q_{2}$ |
| 9                  | 1.58    | 1.59    | 1.63    | 5.89    | 6.08    | 6.43    | 10.15   | 10.21   | 10.27   | 2.61    | 2.64    | 2.72    |
| 12                 | 2       | 2.01    | 2.05    | 5.41    | 5.58    | 5.89    | 10.16   | 10.18   | 10.28   | 2.61    | 2.64    | 2.71    |
| 15                 | 2.81    | 2.83    | 2.84    | 4.82    | 4.96    | 5.26    | 10.18   | 10.23   | 10.35   | 2.58    | 2.62    | 2.70    |
| 18                 | 5.18    | 5.21    | 5.29    | 3.98    | 4.09    | 4.28    | 10.24   | 10.29   | 10.41   | 2.55    | 2.58    | 2.65    |

Note: 0 in the subscript of $t$, $t_s$, $q_{max}$, $q_s$ represents "no control edges", 1 represents "single control edge", 2 represents "dual control edges".
**Data Availability**

The data used to support the findings of this study are included within the article.

**Conflicts of Interest**

The authors declare that there are no conflicts of interest.

**Authors’ Contributions**

The authors Yuqi Wang, Xinhui Liu, Jinshi Chen, Yafang Han, Siyuan Liu, and Dongyang Huo have presented the proposed study. The contributions of the authors are as follows: Yuqi Wang and Xinhui Liu conceptualized the study; Yuqi Wang and Jinshi Chen were responsible for the methodology; Yafang Han developed the software; Siyuan Liu and Dongyang Huo validated the study; Yuqi Wang and Xinhui Liu performed formal analysis; Yuqi Wang and Jinshi Chen were responsible for original draft preparation; Xinhui Liu reviewed and edited the manuscript; Jinshi Chen was responsible for project administration; and Xinhui Liu was responsible for funding acquisition. All authors have read and agreed to the published version of the manuscript.

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