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Characteristics of Proportional Flow Control Poppet Valve with Pilot Pressure Compensation

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Abstract

Electro-hydraulic proportional flow valves are widely used in hydraulic industry. There are several different structures and working principles. However, flow valves based on the existing principles usually have some shortcomings such as the complexity of the system and additional energy losses. A concept for a two-stage poppet flow valve with pilot pressure drop – pilot spool opening compensation is presented, and the linear relationship between the pilot stage and main stage, the semi-empirical flow equation are used in the electronic flow controller. To achieve the accurate control of the outlet flow, the actual input voltage of the pilot spool valve is regulated according to the actual pilot pressure drop, the desired flow rate and the given input voltage. The results show that the pilot pressure drop – pilot spool opening compensation method is feasible, and the proposed proportional flow control valve with this compensation method has a good static and dynamic performance.

KEYWORDS: Proportional flow control valve, Poppet valve, Pressure compensator

1. Introduction

Proportional flow control valves are widely used in actuators motion and speed control. This kind of valve can be divided into proportional throttle valve and flow valve. As for the throttle valve, only the valve opening is precisely controlled, and its outlet flow can be easily affected by the variation of pressure drop across the valve metering orifice. By contrast, the outlet flow of the proportional flow valve can be precisely controlled.

Broadly speaking, there are generally two types of proportional flow control valves according to the different operating principle, that is, pressure difference compensation and flow feedback /1/. The former is made up of a throttle valve and a pressure compensator, in which the compensator is in parallel or in series with the throttle valve. The compensator is used to keep the pressure drop across the throttle valve metering orifice to a constant value. This method can ensure the outlet flow approximately constant, and has been deeply studied by many researchers /2, 3/. The shortcomings of this type valve are low accuracy, high flow overshoot in its opening moments or the step change in the load pressure. Sometimes the overshoot even reaches more than 100% of the given value. Another one is the flow feedback type valve, which regards the outlet flow as testing and feedback object. For example, in the early 1980s, Lu invented a two-way proportional flow valve based on the flow-displacement-force feedback principle /4/. The shortcoming of the type valve is that the detection and feedback of the dynamic flow is difficult. In addition, traditional pressure compensator or dynamic flow sensor are usually installed in the main circuits, on one hand this layout produces additional throttling losses, on the other hand it is easy to increase the valve size and the system complexity. These shortcomings will be more serious in the large flow occasions.

In this paper, a two stages cartridge proportional flow valve based on the pilot pressure difference-displacement compensation scheme is presented. Two pressure sensors and a electronic controller are integrated in the valve. To achieve the accurate control of the flow rate, the approximate liner relation between the steady outlet flow of the main stage and that of the pilot stage, and the semi-empirical flow equation are used in the controller to fulfill the correction and compensation scheme. Compared with the traditional pressure compensation or flow feedback type proportional flow valve, the throttling loss of the proposed scheme is lower because there is no mechanical pressure difference compensator or flow sensor. As a result, this scheme can also be used in the control of large flow conditions.

2. Configuration and Mathematical Modeling

2.1. Configuration of the Proposed Flow Valve

As shown in **Figure 1**, the proposed flow valve is a two-stage one, the main stage is a Valvistor valve and the pilot stage is a single-stage servo proportional direction valve based on displacement- electrical feedback. The Valvistor valve is firstly put forward by Andersson /5/. It is a poppet valve, and a longitudinal slot is machined in its cylindrical surface to form an internal position feedback channel. Researchers have studied the performance of the Valvistor valve /6-9/. As illustrated in **Figure 1**, the inlet of pilot stage is connected with the control chamber of the main stage, and the outlet of the pilot stage is connected with the outlet of the main stage. When the input voltage of the pilot valve changes, the pilot flow Q_p changes as well, and then the control chamber pressure p_c changes accordingly. As a result, the balance of the poppet is destroyed, the poppet will move upward or downward. It will not stop until the flow rate through the feedback channel Q_s is equal to the Q_P. During this process, the poppet displacement x_m as well as the feedback channel opening (x_m+x_0) will change accordingly.

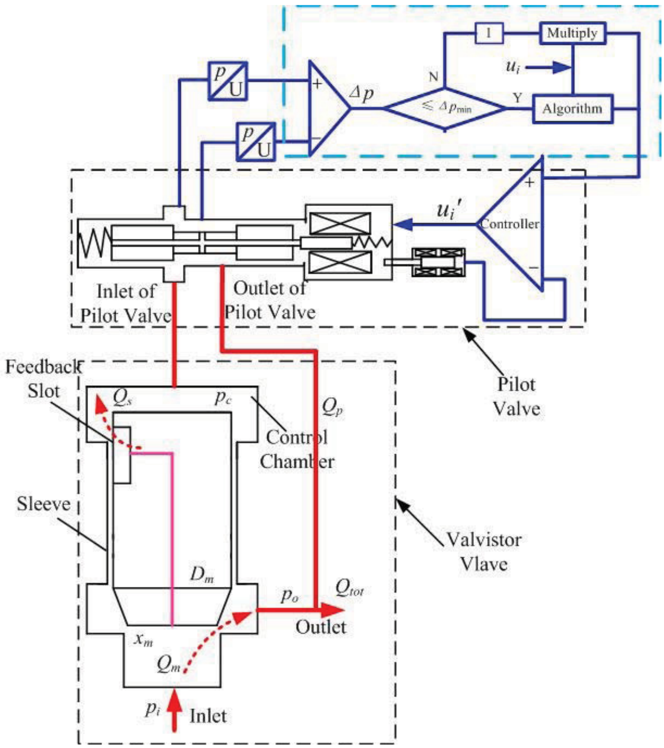


Figure 1: Configuration of the Proposed Valve

In the electronic controller, the pilot outlet flow at 1 MPa pressure drop is designed as the reference flow. The pilot pressure drop is detected by two pressure sensors and then transferred to the controller. If it is less than 1 MPa, the compensation scheme does not work, namely, the actual input voltage of the pilot stage is equal to the given one; else if the actual pilot pressure drop is equal to or higher than 1 MPa, the correction and compensation method will conduct, and the electronic controller will calculate the actual input voltage of the pilot stage according to the actual pilot pressure difference, the given input voltage, and the desired flow rate. Then the electronic controller generates an appropriate control signal to adjust the pilot spool opening.

2.2. Mathematical Modeling

The movement of the pilot valve can be written as follows,

$$F_e = m_p \ddot{x}_p + B_p \dot{x}_p + k_p x_p + 2C_d C_v w_p x_p \left(p_c - p_o\right)\cos\theta \tag{1}$$

If the pilot stage adopts displacement-electrical feedback scheme, the input force F_e can be written as,

$$F_e = k_e \left(u_i' - k_f x_p\right) \tag{2}$$

The movement of the poppet valve can be written as,

$$-2C_d C_v w_m x_m \cos\theta\left(p_i - p_o\right) - A_c p_c + A_a p_i + \left(A_c - A_a\right) p_o = M\ddot{x}_m + B_m \dot{x}_m + kx_m \tag{3}$$

The flow continuity equation of the control chamber between the main stage and the pilot valve is,

$$\frac{dp_c}{dt} = \frac{\beta_e}{V_c}\left(Q_s + A_c \dot{x}_m - Q_p\right) \tag{4}$$

The flow rate through the pilot valve is written by,

$$Q_p = C_d w_p x_p \sqrt{\frac{2}{\rho}\left(p_c - p_o\right)} \tag{5}$$

The opening of the variable orifice on the side surface of the poppet is determined by the main poppet position x_m and the underlap x_0. The flow through the slot is,

$$Q_s = C_d w_s \left(x_m + x_0\right)\sqrt{\frac{2}{\rho}\left(p_i - p_c\right)} \tag{6}$$

The flow through the poppet valve Q_m is written by,

$$Q_m = C_d w_m x_m \sqrt{\frac{2}{\rho}\left(p_i - p_o\right)} \tag{7}$$

$$w_m = \pi\left(D_m - \frac{x_m}{2}\right)\sin\frac{\pi}{4} = 2.22\left(D_m - \frac{x_m}{2}\right) \tag{8}$$

The total outlet flow of the proposed valve is expressed as,

$$Q_t = Q_m + Q_p \tag{9}$$

The dynamic response of the pilot stage is faster than that of the main stage. Assuming the flow force is neglected, equation (1) and equation (2) can be simplified as,

$$x_p = \frac{k_e}{k_p + k_e k_f} u_i' \tag{10}$$

Equation (3) can be simplified as equation (11) when the poppet valve is steady, both the spring and flow force of the main stage are neglected.

$$-A_c p_c + A_a p_i + \left(A_c - A_a\right) p_o \approx 0 \tag{11}$$

If there exists A_c=2A_a, then equation (11) can be simplified as,

$$p_c \approx \frac{p_i + p_o}{2} \tag{12}$$

Then the pressure difference of the pilot stage is written as,

$$\Delta p_s = p_c - p_o = \frac{\Delta p}{2} \tag{13}$$

It can be found from equation (13) that the pressure difference of the pilot stage is proportional to that of the main stage when the valve is steady. In addition, there exists Q_p=Q_s. Then equations (10), (12) and (13) are substituted into equations (5) and (6), there exists,

$$x_m = \frac{w_p k_e}{w_s\left(k_p + k_e k_f\right)} u_i' - x_0 \tag{14}$$

It can be easily found from equation (14) there is a linear relationship between the poppet displacement x_m and the actual input voltage of the pilot valve u_i'. When $\Delta p_s \le 1\mathrm{MPa}$, the compensation mechanism does not work, there exists,

$$u_i' = u_i \tag{15}$$

When $\Delta p_s > 1\mathrm{MPa}$, there exists,

$$u_i' = \frac{\sqrt{2} w_m x_0 \left(k_p + k_e k_f\right)}{\left(\frac{\sqrt{2} w_m}{w_s} + 1\right) k_e w_p} + \left(u_i - \frac{\sqrt{2} w_m x_0 \left(k_p + k_e k_f\right)}{\left(\frac{\sqrt{2} w_m}{w_s} + 1\right) k_e w_p} \right) \sqrt{\frac{1}{\Delta p_s}} \tag{16}$$

Because there exists $\frac{\sqrt{2} w_m}{w_s} \gg 1$ in equation (8), equation (16) can be simplified as,

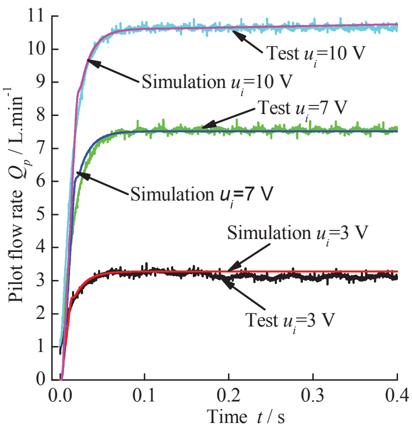
$$u_i' = \frac{w_s x_0 \left(k_p + k_e k_f\right)}{k_e w_p} + \left(u_i - \frac{w_s x_0 \left(k_p + k_e k_f\right)}{k_e w_p} \right) \sqrt{\frac{1}{\Delta p_s}} \tag{17}$$

3. Simulation and Experiment

3.1. Simulation model

The simulation model is established in software SimulationX. It contains the controller, pilot valve, main valve, oil sources and load et al. In the simulation model, a user-defined proportional valve module is used to define the pilot valve 4WRPEH6 integrated in the proportional flow valve. The comparison of the simulation results and measured results of the 4WRPEH 6 are respectively shown in **Figure 2** and **Figure 3**.

It can be seen from **Figure 2** that the test values and simulation results of the step outlet flow of 4WRPEH 6 are basically the same. The static characteristic of the outlet flow rate is shown in **Figure 3**, the simulation results and experiment results are also basically the same. But it should be pointed out that the tested results completely deviate from the simulation results in **Figure 3** when the input voltage is less than 1.5V. This is because that the flow meter used in the test rig has a dead zone. As a result, the output of the flow meter does not change when the flow is less than a certain value. From **Figure 2** and **Figure 3**, it can be concluded that the user-defined proportional valve module of the 4WRPEH 6 in the simulation model is feasible.



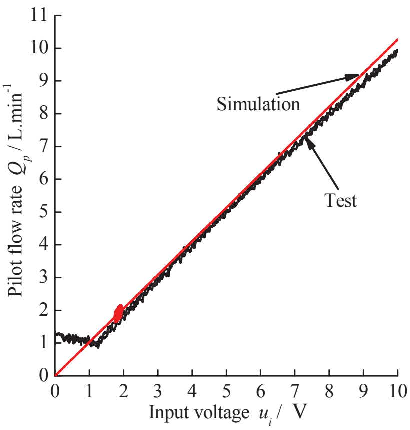


Figure 2: Step Response of Pilot Valve **Figure 3:** Static Characteristic of Pilot Valve

3.2. Experimental Verification

Experimental rig is established, in which the rated flow rate of Valvistor valve is 175 L/min at 1 Mpa pressure difference, and the pilot stage is REXROTH 4WRPEH 6. The inlet, outlet pressures of the Valvistor valve; and the pressure of the control chamber are respectively measured by pressure sensors; and the total outlet flow is measured by Parker gear flow meter (0-150L/min). The oil source is offered by variable displacement piston pump SYDFEE-11/71RN00. The poppet displacement is measured by linear displacement sensor. Flow controller is designed by means of dSPACE unit.

Figure 4 shows the comparison of simulation results and experiment results of the proposed valve with compensation scheme when the inlet pressure of main stage takes a step change. In **Figure 4**, the measured inlet pressure p_i of main valve is imported into the simulation model to calculate for the Q_t, and it can be found that the oscillation amplitude and adjustment transition time of simulation results are smaller than measured results, but the experimental results are in agreement with the simulation results from the trend of change.

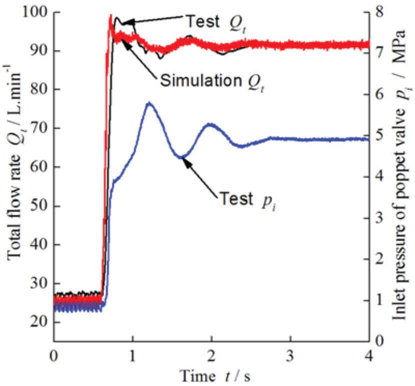
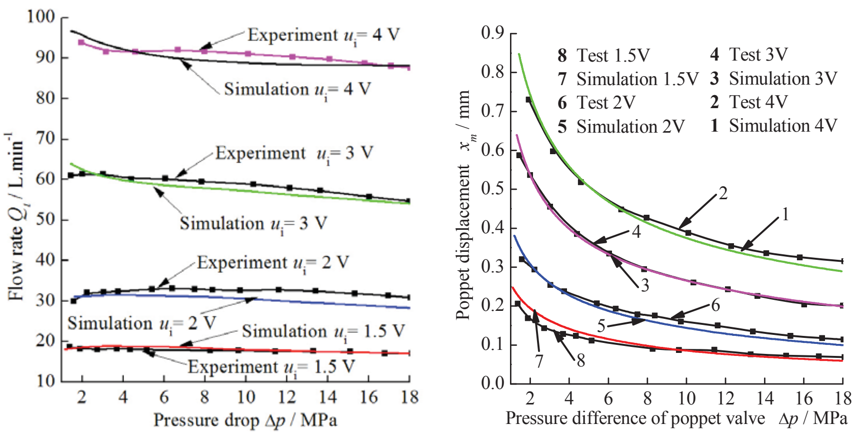


Figure 4: Comparison of Simulation Result and Test Result of the Proposed Valve with Compensation Scheme under Step Change of the Input Pressure



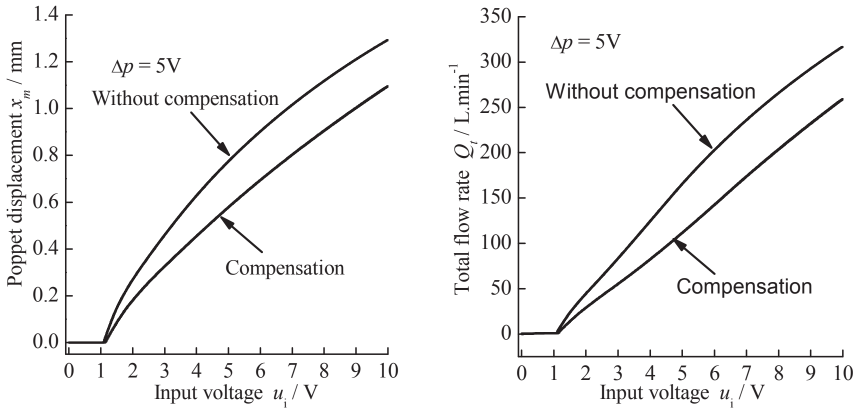
(a) Q_t versus Δp (b) x_m versus Δp

Figure 5: Comparison of the Simulation and Tested Results of the Proposed Valve with Compensation Scheme in the Steady State

Figure 5(a) and **Figure 5(b)** respectively show the comparisons of the total outlet flow Q_t and poppet displacement x_m between simulation and experiment results of the proposed valve with compensation scheme when the load is steady. It can be found that the experimental results are in agreement with the simulation results. All in all, the above simulation and experimental results show that the theoretical derivation in section 2.2 and simulation model of the proposed valve with compensation scheme are reliable.

3.3. The Simulation Analysis of Steady State Characteristics

Figure 6(a) and **Figure 6(b)** respectively show the steady state control characteristic of the proportional throttle valve and the proposed valve. These figures show that both the x_m and Q_t have dead zones when the pilot input voltage u_i is less than 1.2 V, which is related to the flow capacity of the pilot valve, underlap x_0 and gradient flow area w_s. In addition, **Figure 6(a)** and **Figure 6(b)** also show that both x_m and Q_t of the proposed valve are less than that of the throttle valve under the same working condition; and the linearity of x_m and Q_t of the former are better than that of the latter. Accordingly, on one hand the proposed scheme can contribute to the improvement of the steady control performance of the flow valve, one the other hand the range of outlet flow will be reduced relatively.

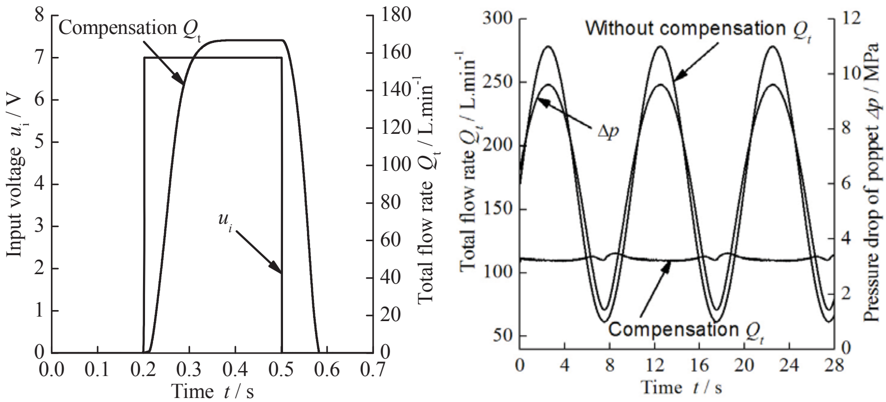


(a) x_m versus ui (b) Q_t versus u_i

Figure 6: The Steady Control Performance of the Proposed Flow Valve and the Proportional Throttle Valve

3.4. The simulation analysis of dynamic characteristics

Figure 7(a) shows the Q_t step response of the proposed proportional flow valve with compensation scheme. It can be found that the adjustment time of Q_t is about 160 ms without overshoot and oscillation during the ascending stage; and the adjustment time is about 82.8 ms during the falling stage of the input signal, which is only about the half of the adjustment time when the valve is in the opening phase. As a result, it can be concluded that the proposed proportional valve possesses the ability to quickly cut off the outlet flow. This phenomenon is related to the structure of the Valvistor valve. When the pilot valve is closed, the control chamber pressure p_c of the Valvistor valve rise rapidly, until it is equal to the inlet pressure p_i; and due to the control chamber area is larger than the inlet chamber area, thus the poppet closes quickly.



(a) Step input signal (u_i = 7V) (b) Flow rate – pressure difference

Figure 7: Dynamic response of the proposed valve with the compensation scheme

Figure 7(b) shows the Q_t of the proposed flow valve and that of the proportional throttle valve when the load changes dynamically. It can be found that the Q_t of the proposed valve is roughly constant and the fluctuation range is about 5 L/min when the pressure difference periodically changes and the amplitude is 10MPa. But the Q_t of the proportional throttle valve will fluctuate periodically with the periodical change of pressure difference, and the range is up to 260 L/min. Therefore, the proposed flow valve possesses the high flow control accuracy under the condition of dynamic load.

4. Conclusion

Both the simulation and experimental results show the proposed proportional flow valve is feasible, and can realize the precise control of the outlet flow. In addition, it possesses good static and dynamic characteristics.

The proposed flow valve can be applied to the large flow control condition because it achieves flow control without a mechanical component to adjust the pressure drop across the metering orifice. As a result, the additional energy losses resulting from the pressure compensator is almost zero. In addition, the sensors will improve the intelligence and performance of the valve.

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6. Nomenclature

k	Spring constant of the poppet valve	N/mm
k_e	Electronic gain the proportional solenoid	N/V
k_f	Displacement Sensor Gain	Mm/V
k_p	Spring constant of pilot valve	N/mm
m_p	Mass of the pilot spool	kg
p_c	Pressure of the control chamber	N/mm^2
p_i	Inlet pressure of the poppet valve	N/mm^2
p_o	Outlet pressure of the pilot valve	N/mm^2
t	Time	s
u_i'	Actual input voltage of the pilot valve	V
w_m	Gradient flow area of poppet orifice	mm
w_p	Gradient flow area of pilot valve orifice	mm
w_s	Gradient area of the feedback slot	mm
x_0	Underlap of feedbcak slot on the poppet	mm
x_p	Displacement of the pilot spool	mm
x_m	Displacement of the poppet	mm
A_a	Cross-sectional area of poppet (small side)	mm^2
A_c	Cross-sectional area of poppet (large side)	mm^2
B_m	Damping coefficient of poppet	N.s.mm^{-1}
B_p	Damping coefficient of pilot valve	N.s.mm^{-1}

C_d	Discharge coefficient of the pilot valve orifice	
C_v	Flow velocity coefficient	
D_m	Diameter of the poppet valve (large side)	mm
F_e	Force produced by proportional solenoid	N
Q_m	Flow rate of the poppet	mm^3/s
Q_p	Pilot flow rate	mm^3/s
V_c	Volume of the pilot circuit	mm^3
θ	Jet angle of the orifice	°
β_e	Fluid bulk modulus	N/mm^2
ρ	Oil destiny	kg/mm^3