Research on the High-speed On/off Valve with High Pressure and Large Flow Rate Based on Phase Shift PWM Control

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Abstract: - To reduce the chatter amplitude and widen the range of the effective duty cycle of the high-speed on/off valve in digital variable displacement pump systems, this paper presents a new strategy, phase shift PWM (PS-PWM) control strategy. The theoretical and simulation analysis results indicate that this strategy could improve the dynamic response speed and steady precision significantly. In addition, to improve the control precision and reduce the leakage under high pressure and large flow rate, a new type of high-speed on/off valve with two DOF of the spool is developed. The duty cycle of this valve is increased significantly, which reaches 0~90%. The floating compensation mechanism of this valve is designed to realize the wear self-compensation. So this valve can guarantee the high control precision, low leakage and high efficiency under high pressure and large flow rate.

Key-Words: - Digital variable displacement pump system, High-speed on/off valve, Duty cycle, Control precision, Phase shift PWM, Two DOF of the spool, Floating compensation mechanism, Wear self-compensation

1 Introduction

Improving the energy efficiency of hydraulic systems is one of the hot topics in mechanical engineering[1-3]. At present, the valve-controlled technology, which is widely applied to hydraulic systems, causes a lot of throttling loss, so it results in the low energy efficiency of hydraulic systems. The direct pump-controlled technology without throttling loss is a important method to solve these problems [4]. But the frequency width of the variable displacement pump system is usually too low [5], so that it cannot meet the requirements of high speed response control applications in modern industry [6]. Therefore, improving the response speed of the variable displacement pump has become the key issue to exploit its application area.

Noah D. Manring et al. [6] found that the response speed of the main pump could be improved through increasing the surge absorption capability of the variable mechanism. But the power loss would be increased since using the large displacement pilot pump. Perry Y. Li et al. [5] used the high-speed on/off valve to control the fixed displacement pump instead of using the variable displacement pump. In that case, they found the response speed and power density of pumps were both improved. However, the high-speed on/off valve they designed had some drawbacks, such as the large leakage under high

pressure. ShunFeng Ceng et al. used the same principle to construct a digital variable displacement pump system [7]. Compared to the proportional valve system and servo valve system, the throttling loss of the digital variable displacement pump system was much smaller [8]. Therefore, using the high-speed on/off valve instead of the proportional valve and servo valve can improve the efficiency of hydraulic systems dramatically. But this method will cause the high-frequency chatter and low pressure control precision, so the pressure control precision is usually less than 0.15MPa at present [5,7]. Although the control precision could be improved by increasing the PWM control frequency, the service life of the high-speed on/off valve will be shortened, it also leads to the effective duty cycle within a small range [10]. In addition, the control frequency is limited by the response speed of the valve, it cannot be improved too much. Thus the PWM control frequency is usually set to 20~60Hz now.

To improve the pressure control precision and widen the range of the effective duty cycle, this paper presents a new strategy, PS-PWM control strategy, to optimize the traditional PWM control strategy and the theoretical and simulation analysis are presented to demonstrate its effectiveness. To realize the accurate control of the outlet flow rate of the fixed displacement pump under high pressure

and large flow rate, a new type of high-speed on/off valve with two DOF of the spool is developed. In addition, the floating compensation mechanism of this valve is designed to realize the wear self-compensation.

2 Characteristic Analysis of The Traditional PWM Control

The defects of the traditional PWM control is analyzed by choosing the pressure as research object.

The traditional hydraulic circuit [5], which the high-speed on/off valve controls the outlet pressure of the fixed displacement pump, is shown in Fig. 1.
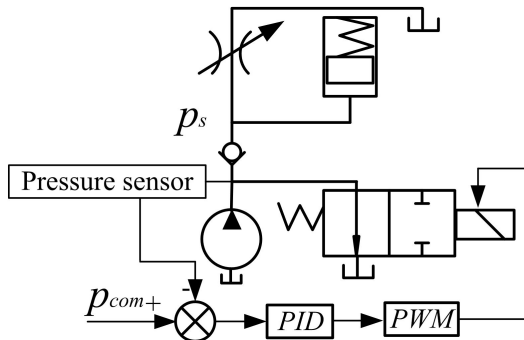

Fig. 1. Circuit of the traditional PWM control

As shown in Fig. 1, the flow rate is adjusted by regulating the duty cycle of the high-speed on/off valve, the pressure vibration is reduced through using the accumulator and the outlet pressure of the pump is controlled by the valve.

The dynamic equation of the outlet pressure of the fixed displacement pump is defined as:

$$\dot{p}_s = \frac{\beta(p)}{V_{in}}\left(Q_p - Q_t - Q_l - a_s\dot{x}\right) \tag{1}$$

Where p_s and V_{in} denote the outlet pressure and outlet volume. Q_p is the outlet flow rate, Q_p is constant if the speed of the drive shaft is fixed. Q_t and Q_l represent the flow rate through the high-speed on/off valve and load. a_s and x are the piston area and displacement of the accumulator. $\beta(p)$ is the bulk elastic modulus of the oil, it could be described as [8]:

$$\beta(p) = \frac{\beta_{oil}\left(1+10^{-5}p\right)^{(1+1/\gamma)}}{\left(1+10^{-5}p\right)^{(1+1/\gamma)} + 10^{-5}R\left(1-c_1p\right)\left(\beta_{oil}/\gamma - 10^5 - p\right)} \tag{2}$$

In which β_{oil} represents the bulk elastic modulus of the oil under atmospheric pressure and is set to $1701MPa$. R is the gas content percent of the oil. c_1 represents the coefficient of the change of bubble volume caused by the change of gas volume, which could be set to -9.307×10^6.

The change of $\beta(p)$ is influenced by the change of the pressure, it cannot be negligible under the condition of $10MPa$. If the pressure is over $10MPa$, it will change slowly and could be approximately a constant [11].

The equation of the flow rate through the high-speed on/off valve is defined as [12]:

$$Q_t = c_d A_p D\sqrt{\frac{2\Delta p}{\rho}} \tag{3}$$

Where c_d represents the flow rate coefficient of the valve port and it is set to 0.67. $A_p = \frac{\pi}{4}d^2$, d is the diameter of the valve port. D and Δp represent the duty cycle of the PWM signal and the pressure difference of the valve port. ρ is the oil density and it is set to $870kg/m^3$.

The equation of the displacement of the variable piston can be presented as:

$$p_s a_s = m_s\ddot{x} + k_s x + b_s\dot{x} \tag{4}$$

In which m_s represents the piston mass of the accumulator. k_s and b_s denote the spring stiffness of the accumulator and the damping coefficient of the piston motion.

Because the values of m_s and b_s are both small, Eq. (4) can be simplified as:

$$p_s a_s = k_s x \tag{5}$$

Eq. (5) is taken the derivation with time and it could be presented as:

$$\dot{p}_s a_s = k_s\dot{x} \tag{6}$$

Substituting Eq. (6) into Eq. (1), we can obtain it as:

$$\dot{p}_s = \frac{\frac{\beta(p)}{V_{in}}\left(Q_p - Q_l - Q_t\right)}{1 + \frac{\beta(p)}{V_{in}}\frac{a_s^2}{k_s}} \tag{7}$$

When the control system is in steady state and uses the proportional valve or servo valve, $\left(Q_p - Q_l - Q_t\right) = 0$.

If the high-speed on/off valve is adopted to control, Q_p and Q_l could be approximately constants in steady state, the PWM control frequency could be also set to a constant. t_{PWM} represents one PWM control cycle length, during one cycle, the

high-speed on/off valve has two kinds of state, fully open and fully closed. During the fully closed stage, $Q_t = 0$, so the increase of the pressure is defined as:

$$\Delta p_{up} = \dot{p}(1-D)t_{PWM} = \frac{c_d A_p \sqrt{\dfrac{2p_s}{\rho}}(1-D)t_{PWM}}{\dfrac{V_{in}}{\beta(p)} + \dfrac{a_s^2}{k_s}} \quad (8)$$

Similarly, during the fully open stage, the drop of the pressure could be presented as:

$$\Delta p_{down} = \dot{p}Dt_{PWM} = \frac{c_d A_p \sqrt{\dfrac{2p_s}{\rho}}Dt_{PWM}}{\dfrac{V_{in}}{\beta(p)} + \dfrac{a_s^2}{k_s}} \quad (9)$$

According to Eq. (8) and Eq. (9), we can conclude that the steady pressure has a fixed variable when the high-speed on/off valve is used to control the system pressure. In order to improve the pressure control precision, the value of the variable should be as small as possible. The change of the pressure could be decreased by decreasing the spring stiffness of the accumulator k_s, or increasing the piston area of the accumulator a_s or increasing the outlet volume of the pump V_{in}. But in this case, the response speed of the pressure control will be reduced at the same time, so it affects the precision of the pump-controlled system. To avoid the influence on dynamic response, the value of t_{PWM} should be reduced, which means the PWM control frequency should be increased. When the high-speed on/off valve adopts different PWM control frequency, the response characteristics of the pressure are shown in Fig. 2.
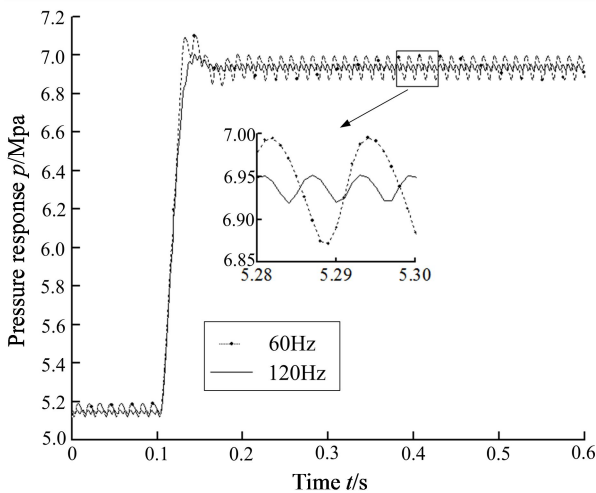

Fig. 2. Response characteristics of the pressure under different PWM control frequency

Fig. 2 shows that the chatter amplitude of the steady pressure depends on the PWM control frequency directly if the high-speed on/off valve uses the traditional PWM control strategy.

3 Optimization on PWM Control Strategy

Traditional high-speed on/off valves control the electromagnet to create suction force according to a series of pulse electrical signals, then this force makes the spool move forward and reverse. Therefore, the flow could alternately circulate and block at the valve port [13]. The duty cycle and PWM signal are both generated by the axial motion of the spool. That is to say, the two signals can be controlled by controlling the single DOF motion of the spool.

The most efficient measure to improve the control precision of the outlet pressure is increasing the PWM control frequency of the high-speed on/off valve. But in this case, the on/off times of a single high-speed on/off valve are increased under the traditional PWM control strategy, so the service life of the valve will be shortened and the range of the effective duty cycle will be narrowed. The response frequency of the high-speed on/off valve is about 166Hz at present, it means that the range of the effective duty cycle is only 20%~80%. To improve the control precision of the system and widen the range of the effective duty cycle, the PS-PWM control strategy is presented in this paper.

3.1 Theoretical Analysis

Considering the service life of a single high-speed on/off valve has effects on hydraulic systems, the single high-speed on/off valve is replaced by several identical high-speed on/off valves which are connected in parallel.
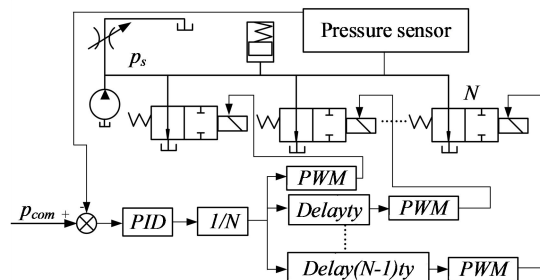

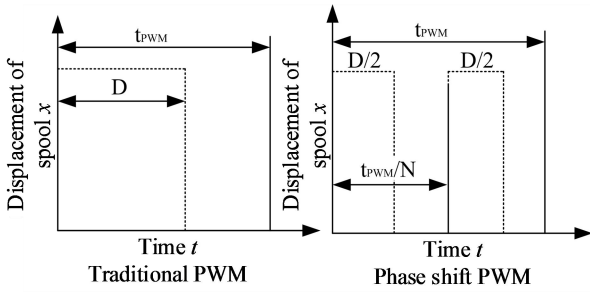Fig. 3. Phase relationship of each valve under PS-PWM control

Fig. 4. Displacement of the spool under PS-PWM control

The phase relationship of each valve and displacement of the spool under the PS-PWM control are shown in Fig. 3 and Fig. 4. The pressure error signal is divided into N parts through PID processing, this signal is transferred to the first high-speed on/off valve without time delay. Then it is delayed t_y and transferred to the second valve, where $t_y = t_{PWM} / N$. In the same way, it is delayed $(N-1)t_y$ and transferred to the Nth valve. The PS-PWM control strategy can increase the PWM control frequency, the improved times is the number of the high-speed on/off valves. Since the on/off times of each high-speed on/off valve are not increased, the service life of the single valve is not affected.

Considering the response delay property of the high-speed on/off valve has effects on the dynamic response [14], the delay time should be modified to compensate the response delay of the high-speed on/off valve. It means that except the delay time of the first high-speed on/off valve cannot be compensated, the other (N-1) valves could be compensated by decreasing the delay time. Therefore, the delay time are modified as:

$$t_y = t_{PWM} / N - t_d \qquad (10)$$

In which t_d represents the response delay time of the high-speed on/off valve.

Since the effective duty cycle of the traditional high-speed on/off valve is only 20%~80%, the range that the valve can response to the error signal accurately is small. After adopting the PS-PWM control strategy, the effective duty cycle is divided into N parts, the effective duty cycle that each high-speed on/off valve needs to response is decreased. Thus if $N \geq 2$, the maximum of the effective duty cycle can reach to 100%, the range of the duty cycle that the system can control is increased. However, the minimum of the effective duty cycle will be also increased in this case, the qualification is:

$$D \geq 20N\% \qquad (11)$$

on the other hand, because the limit of the response speed and the minimum of the effective duty cycle, the number of the high-speed on/off valves, N, should match the response characteristics. That is to say, if the response frequency of the high-speed on/off valve is assumed to $x_h Hz$, it is required as:

$$\frac{t_{PWM}}{N} \geq \frac{1}{x_h} \qquad (12)$$

If the limitations mentioned above are satisfied, it is guaranteed that the high-speed on/off valve from fully open to fully closed position could be realized in one PWM control cycle. Considering the dead zone of the duty cycle has influence on the system control, the value of t_{PWM} should be increased, but too large value will increase the chatter amplitude. There is not an accurate rule to choose the PWM control frequency, the frequency is concentrated in 20~60Hz now [10].

If the duty cycle can satisfy Eq. (11) when the system is in steady state, the PS-PWM control strategy mentioned above could be adopted. But if the duty cycle is too small, Eq. (11) cannot be satisfied, the PS-PWM control strategy should be modified. The example of $N = 2$ is used to illustrate the improvement measure.

When $N = 2$, the range of the effective duty cycle is 40%~100%, the dead zone is 0%~40%, the range of the dead zone is too large obviously. Thus except the two high-speed on/off valves with the large valve port (the area of the valve port is set to A_{p1}), two valves with the small valve port (the area of the valve port is set to A_{p2}) are used at the same time, where

$$A_{p2} = 0.2 A_{p1} \qquad (13)$$

When the effective duty cycle exceeds 40%, the pressure error signal is transferred to the two high-speed on/off valves with the large valve port. When the effective duty cycle is less than or equal to 40%, the signal is transferred to the two high-speed on/off valves with the small valve port. It can be obtained from Eq. (3):

$$c_d A_{p1} D_1 \sqrt{\frac{2\Delta p}{\rho}} = c_d A_{p2} D_2 \sqrt{\frac{2\Delta p}{\rho}}$$

Substituting Eq. (13) into the equation above, the effective duty cycle can be presented as:

$$D_1 = 0.2 D_2$$

In which D_1 and D_2 are the effective duty cycle of the high-speed on/off valve with the large valve

port and effective duty cycle of the high-speed on/off valve with the small valve port.

At this moment, the minimum of the effective duty cycle is expanded to 8%. So when the error signal is small, the effective duty cycle could be expanded by decreasing the area of the valve port. The range of the effective duty cycle under different control strategies is shown in Table 1.

Table 1 Effective duty cycle under different control strategies

Control strategy	Range of the effective duty cycle(%)
Traditional PWM control	20~80
PS-PWM control	8~100
Variable quantity (%)	+53.3

As shown in Table 1, if the PS-PWM control strategy is adopted, the minimum and maximum of the effective duty cycle would be widen concurrently, the increase reaches 53.3%. At the same time, the whole system life is extended.

3.2 Simulation Analysis

The PS-PWM control strategy is compared to the traditional PWM control strategy, the effective duty cycle is assumed to exceed 40%. That is to say, the valve with the small valve port is not used. The high-speed on/off valves are the HSV series of HongLin Machinery Company, the diameter of the valve port $d = 1.32 mm$. In addition, the opening time is set to less than 3.5ms, the closing time is less than 2.5ms and the response frequency exceeds 160Hz. According to Eq. (10) and the empirical value, the number of the high-speed on/off valves is set to $N = 2$, the PWM control frequency is set to 60Hz. The two control strategies are analyzed by simulation software AMEsim.
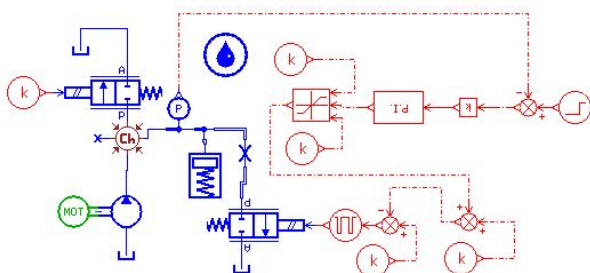

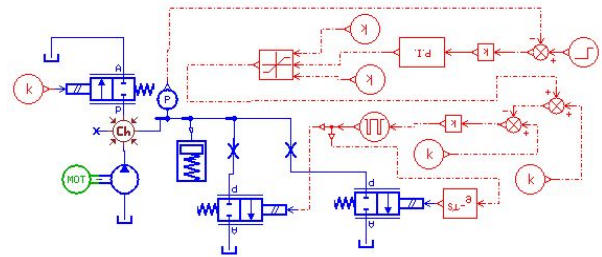Fig. 5. Circuit of the traditional PWM control

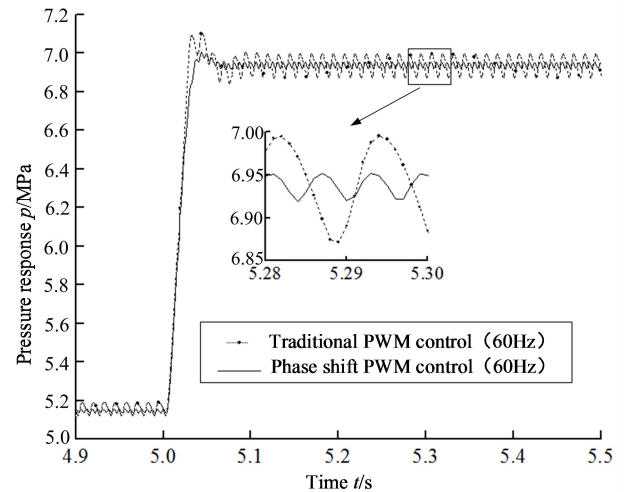Fig. 6. Circuit of the PS-PWM control when N=2

Fig. 7. Control pressure response of the traditional PWM control and the PS-PWM control

As shown in Fig. 7, the overshoot of the pressure dynamic response is reduced from 0.15MPa to 0.05MPa, the steady precision is improved from 0.125MPa to 0.03MPa and the response time is reduced from 0.08s to 0.05s. Therefore, the PS-PWM control strategy can not only reduce the overshoot of the dynamic response but also improve the steady precision, the improved times of the steady precision is more than the number of the valves. We can see from the results that the steady precision is increased fourfold, in other words, the relationship between the number of the valves and the improved times of the steady precision is near to 2^N.

3.3 Analysis of The PWM Control Frequency

Considering the service life and use occasions of the high-speed on/off valve, the PWM control frequency is usually set to 20~60Hz [5] [10], but the steady precision will be less than 0.15MPa under this condition, so there is chatter in steady state. According to the analysis of the PWM control frequency in [14], only when the frequency reaches about 100Hz, the chatter amplitude is near to zero in

steady state. However, if the PWM control frequency reaches 100Hz, the service life would be affected and the control will be distorted under the traditional PWM control.

In order to let the high-speed on/off valve play the advantage of the small throttling loss and reduce the chatter at the same time. According to the technical indicators of the high-accuracy pressure control in industrial control systems and existing pressure sensors, the chatter amplitude should be less than 0.05MPa. When the amplitude is less than 0.05MPa, the control performance of the traditional PWM control strategy and PS-PWM control strategy is shown in Table 2. Where the service life of the valve under the traditional PWM control is set to 1.

Table 2 Performance parameters of different control strategies with low chatter amplitude

	Traditional PWM	PS-PWM
PWM control frequency (Hz)	120	60
Controllability	Distortion	Controllable
Range of the duty cycle (%)	20~80	8~100
Service life	1	2

As shown in Table 2, the traditional PWM control strategy has many problems, such as the small range of the duty cycle and the control distortion which is caused by the too high PWM control frequency. These problems could be solved preferably by using the PS-PWM control strategy, the PWM control frequency could only be set to 60Hz.

4 High-speed on/off Valve with Two DOF of The Spool

When the outlet flow rate of the pump is small, the accurate control of the outlet pressure could be realized by using the traditional high-speed on/off valve with single DOF of the spool and the PS-PWM control strategy. But with the increase of the outlet flow rate, the hydraulic circuit mentioned in section 2 will not be adaptive because the limit of the surge absorption capability of the traditional valves. The main ways to improve the surge absorption capability of the high-speed on/off valve include: (1) the structure of pilot control [7,15], (2) the structure of rotary valve [16] and (3) the

structure of relative motion of the spool and valve sleeve [13]. The performance parameters of these structures are shown in Table 3, the surge absorption capability is calibrated by the pressure difference of 4MPa.

Table 3 Performance parameters of different structures of the high-speed on/off valve

	Opening time (ms)	Closing time (ms)	Surge absorption capability(L/min)
Pilot control	7	12	107
Rotary valve	<0.5	<0.5	40
Relative motion of the spool and valve sleeve	0.78	1.10	35

As shown in Table 3, if the structure of pilot control is adopted, the surge absorption capability could be increased but the response speed will be decreased [15]. The structure of relative motion improves the surge absorption capability and the response speed dramatically, but it is worse than the structure of rotary valve. Different from the other structures, the opening and closing time of the rotary valve are very close and both less than 1ms. Meanwhile, the surge absorption capability is large. The spool of the structure of pilot control is single DOF, so it is limited by the inertia force during control process. In addition, with the increase of the diameter, the PWM control frequency is reduced [17]. In order to guarantee the steady precision, the frequency cannot be too low. As the too low frequency will decrease the effective duty cycle, which means the working range is narrowed. Therefore, the high-speed on/off valve with single DOF of the spool cannot be applied to the occasion under large flow rate. According to the structure of rotary valve that was proposed by Haink C. Tu et al. [16], the spool had two DOF, the rotational motion and the axial motion. The rotational motion of the spool generates the PWM control signal and the axial motion controls the duty cycle. In other words, the PWM control frequency is only related to the rotational speed. Thus the PWM control frequency is stable, cannot be affected by the change of the flow rate. Furthermore, since the opening and closing time are symmetric and short, it has many advantages, such as the fast response and stable control. However, this kind of high-speed on/off
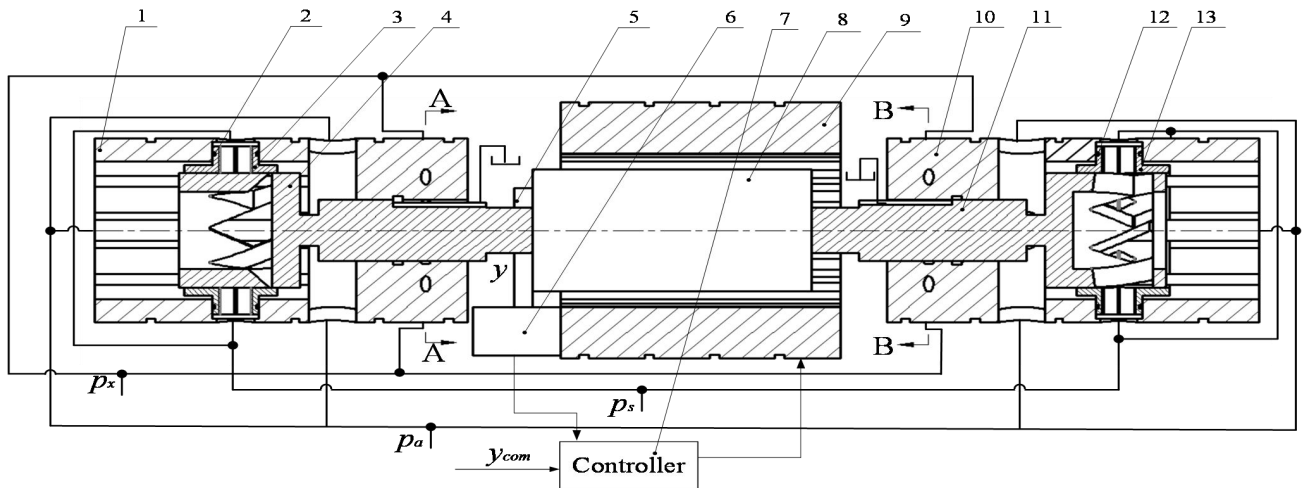
Fig. 8. High-speed on/off valve with low leakage and large flow rate

1, 10 - left valve sleeve   2, 13 - compensation mechanism   3, 12 - sealing ring   4, 11 - spool
5 - moving part of the displacement sensor   6 - static part of the displacement sensor   7 - controller
8 - secondary of the electromagnet   9 - primary of the electromagnet

valve also exists some problems, like the big leakage and low efficiency [18]. Therefore, this paper presents a new type of high-speed on/off valve with low leakage and large flow rate, the structure is shown in Fig. 8.

4.1 Floating Compensation Mechanism

The symmetric structure ensures this valve is uniformly forced and makes the momentum of the inflow and outflow counteract, it can reduce the axial hydraulic force significantly, so the axial driving force could be reduced. The external drive is using the pilot constant voltage source p_x to pass the valve sleeve holes in A-A and B-B cross section and impact the spool. Then the rotational motion of the spool generates the PWM signal. This driving mode guarantees the rational speed of the spool is stable under different loads, realizes the spool bracing and lubrication. Furthermore, the problems of using the motor to drive, such as complicated structure, could be avoided.

To realize the wear self-compensation and improve the efficiency of the rotary valve under high pressure, according to the characteristics of the valve port, the floating compensation mechanism is designed and the structure is shown in Fig. 9 and Fig. 10.
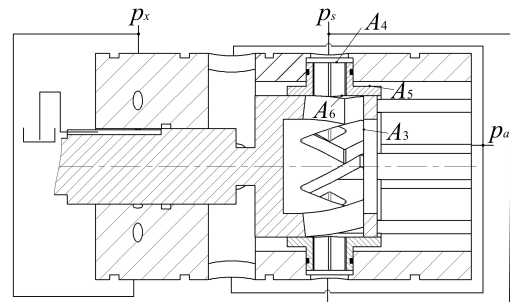

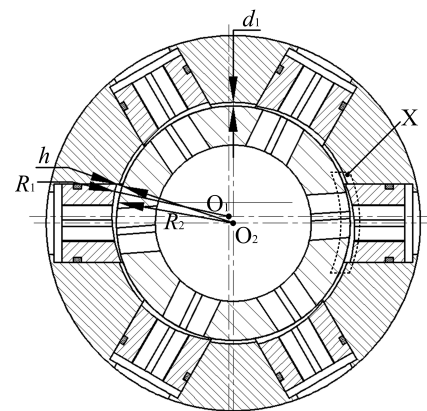*Fig. 9. Schematic diagram of the floating compensation mechanism*

Fig. 10. Structure of the compensation mechanism

There are the convergent-divergent gaps between the six floating compensation blocks and the spool, so the rotational motion of the spool will create a layer of dynamic pressure oil film to

support the load. Different from the journal bearing, it only has the contraction, thus the distribution of the dynamic pressure is near to the sliding support of inclined plane. On the other hand, the high-pressure oil will enter into these gaps and create the hydraulic reverse thrust to balance the press force.

The force analysis of the floating compensation mechanism is achieved and the resultant force is presented as:

$$F_m = p_s A_4 + p_a A_5 - p_{ave} A_6 \qquad (14)$$

Where p_{ave} represents the average pressure of the lower surface of the floating compensation mechanism, it is influenced by the rational speed of the spool and size of the mechanism.

Because the resultant force $F_m > 0$ is always satisfied, the compensation mechanism could guarantee the sealing of the high-speed on/off valve under different working conditions.

To decrease the friction of the axial motion and improve the PWM control frequency, the interfaces between the spool and the valve sleeve, the interfaces between the spool and the compensation mechanism should be under oil lubrication. The condition of generating the oil film is [19]:

$$\frac{\mu v_\omega}{p_{aveb}} < C \qquad (15)$$

In which μ and v_ω represent the viscosity of lubricant and the relative velocity. $v_\omega = \omega r$, where ω is the rational speed of the spool, the design value is $126 rad/ s$, r is the radius of the spool. p_{aveb} denotes the pressure of the average load of the compensation mechanism, $p_{aveb} = \dfrac{F_m}{A_6}$.

When the rotational motion of the spool meets Eq. (15), the supporting mechanism of the spool and the seal are both under oil lubrication. Therefore, the friction resistance of the rotational and axial motion will be decreased significantly. Owing to the oil lubrication state, the friction coefficient f is fixed, the displacement control precision of the spool could be improved. It means that the control precision of the duty cycle could be improved.

4.2 Characteristic Analysis of The Flow Rate Control

The structure of the unilateral spool is shown in Fig. 11, where ω is the rational speed of the spool and v represents the velocity of the axial motion.
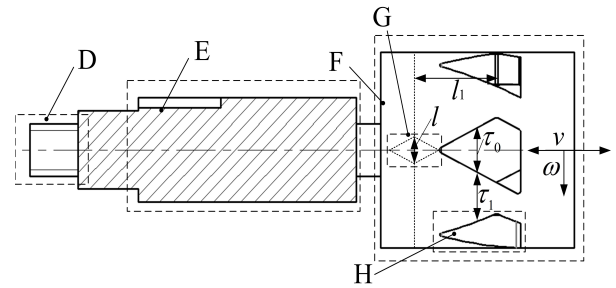

Fig. 11. Flow rate distribution principle of the new type of the high-speed on/off valve with low leakage and large flow rate

D - location part   E - rotation driving part
F - flow rate distribution part of the spool
G - projection of the valve port of the floating compensation mechanism
H - trimming triangular open-apertures of the spool

There are six trimming triangular open-apertures H in the unilateral spool and these open-apertures correspond to the rhombic valve ports of the compensation mechanism G. The rotational motion of the spool generates the PWM control signal, as the rotational speed $\omega = 126 rad / s$, the PWM control frequency is :

$$f = 20 \times 6 = 120 Hz$$

The circumferential arc length of G is set to l, the opening time t_{on} and the closing time t_{off} are presented as:

$$t_{on} = t_{off} = \frac{l}{\omega}$$

Because the opening time equals to the closing time, and the rational speed is constant, the defects of the control system that the response characteristics are different under different working conditions caused by the high-speed on/off valve with single DOF can be avoided.

The whole spool has 12 rhombic open-apertures, so the equation of the total flow rate is presented as:

$$q = 12 c_d A_{pd} D \sqrt{\frac{2(p_s - p_a)}{\rho}} \qquad (16)$$

Where A_{pd} represents the area of a single rhombic open-aperture and $A_{pd} = A_p$, that is to say, it is same with the maximum opening area of the high-speed on/off valve with single DOF of the spool.

The axial motion is driven by the electromagnet or the linear motor, this motion controls the duty cycle. The displacement of the spool is l_1, the proportion of the circumferential

open is the duty cycle D, thus the duty cycle is described as:

$$D = \frac{\tau_0}{\tau_0 + \tau_1} \qquad (17)$$

The minimum of the duty cycle is 0% and the maximum is 90%. These values are mainly limited by the strength of the spool, so the values could be improved by increasing the material strength.

The performance parameters of the traditional high-speed on/off valve and high-speed on/off valve we developed are shown in Table 4.

Table 4 Performance parameters of different high-speed on/off valves

	Single DOF of the spool	Two DOF of the spool	Change (%)
Duty cycle (%)	20~80	0~90	+50
Flow rate (L/min)	20	240	+1100
PWM control frequency (Hz)	60	120	+100

As shown in Table 4, the PWM control frequency and the range of the duty cycle could be improved concurrently by using the high-speed on/off valve with two DOF of the spool. Therefore, the contradiction of the traditional high-speed on/off valve in this aspect is eliminated.

According to the analyses above, when the PWM control frequency reaches 120Hz, the chatter amplitude of the steady pressure is very small and the steady precision is near to the precision grade of the proportional valve. As shown in Table 4, we can see that the high-speed on/off valve with two DOF of the spool has the characteristics of the proportional valve. Furthermore, it also has its own advantage, like the low throttling loss. Therefore, the flow rate characteristics of this valve could be called the high-speed on/off proportional flow rate characteristics and this valve could be called the high-speed on/off proportional valve.

5 Conclusion

This paper used the high-speed on/off valve to control the outlet pressure of the fixed displacement pump instead of using the variable displacement pump. Then the traditional PWM control strategy was analyzed to present the equation of the chatter amplitude. To decrease the chatter amplitude, the PS-PWM control strategy was presented. This strategy could increase the dynamic response speed, steady precision and whole life of hydraulic systems significantly. Meanwhile, the range of the effective duty cycle reaches 8%~100% by adopting the valves with different areas of valve ports, in other words, the range is increased 53.3%.

In order to ensure the hydraulic circuit that the high-speed on/off valve controls the outlet pressure of the fixed displacement pump could be used under high pressure and large flow rate, the new type of high-speed on/off valve with two DOF of the spool was developed. This valve could improve the range of the duty cycle and PWM control frequency concurrently. In addition, the floating compensation mechanism of this valve could realize the wear self-compensation and reduce the friction resistance of the rotational and axial motion.

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