Optimization of the Control Performance of a Novel 3/2 Water Proportional Directional Valve with a Special Position Following Servo Mechanism

Yaoyao Liao, Wenbo Zhao, Jiling Feng, and Zisheng Lian

***Abstract*—The 3/2 water proportional directional valve (PDV) is an important guarantee and also a challenge for achieving low impact, precise and safe control of the hydraulic powered roof support (HPRS) in mining industry. To address this issue, a new 3/2 water PDV is developed. The valve includes a novel three-core follow-up servo mechanism consisting of three key components: the pilot inlet valve, the pilot outlet valve, and the main inlet spool. This mechanism enables precise and reliable opening, and closing capacity of the valve. The prototype is manufactured and the performance test is conducted to improve the control performance of the new water PDV. During the test, two significant discoveries are made. Firstly, it is determined that the control chamber of the main inlet spool must be in the pre-discharge state at the initial time to ensure adequate closing capacity of the new valve during the returning process. Secondly, it is found that the continuous small-step control mode can better cope with the friction and is more effective in achieving quick and smooth following characteristics of the three valve cores in the opening process, compared with the ramp control mode.**

***Index Terms*—Directional valve, dynamic characteristics, proportional control, water hydraulics**

I. INTRODUCTION

THE hydraulic powered roof support (HPRS) plays a crucial role in coal mining operations by providing a secure environment for workers and equipment [1]. It works in tandem with the shearer and scraper conveyor to perform lifting, lowering, and pushing movements. The hydraulic system of the HPRS operates under high-pressure (31.5 MPa) and high-flow (500 L/min, $\Delta p \leq 7$ MPa) condition, with water serving as its transmission medium. Presently, each circuit is controlled by dual two-position three-way (3/2) electro-hydraulic on/off directional valves [2]. During posture adjustment, liquid supplement after leakage of the HPRS, and straightening of the scraper conveyor, precise regulation of each cylinder's position is necessary through the control valve. However, the flow of the on/off directional valve cannot be adjusted continuously and smoothly, which can result in mechanical impact [3], flow and pressure shock [4]. This can cause inaccurate positioning of the actuator [5], affecting the straightness of the scraper conveyor, the posture of the HPRS, and the coupling effect with the surrounding rocks. Furthermore, uneven load on the roof of the support may occur, leading to fractures of the HPRS. In severe cases, these fractures can cause the collapse of the roof, resulting in significant mining disasters. Many researchers have attempted to improve posture control accuracy using various algorithms [6], [7]. However, due to the inherent shortcoming of the on/off valve, which cannot continuously adjust flow, achieving good results has proven difficult. While proportional directional valve (PDV) can solve this problem [8]-[10], water PDV development lags behind oil PDV development. This is because water has much lower viscosity and is prone to leakage compared with mineral oil [11]. Additionally, the high pressure and flow of the hydraulic system requires different structure and sealing methods for the water PDV.

According to the published literature, there are two types of water PDVs: slide valve and seat-type valve. Water Hydraulics, a British company, has developed a 4/3 water PDV which is driven by a proportional electromagnet. The valve port uses a plane seal. However, there is a gap seal between the core and the sleeve, limiting the maximum control pressure to 16 MPa and the flow range to 0-30 L/min [12]. Ebara and KYB, two Japanese companies, have jointly developed a 4/3 water PDV driven by a proportional electromagnet. The valve maintains a specific gap between the valve core and sleeve to enable maximum water flow and lubrication of the friction pair, thereby reducing friction and enhancing dynamic response characteristics. However, this structure also leads to increased leakage. The water PDV can operate at a maximum pressure of 14 MPa and a maximum flow rate of 30 L/min. F. Majdic [13] and Wu [14] have also developed a 4/3 gap-sealed water PDV with pressure ratings of 16 MPa and 14 MPa, respectively.

The maximum pressure of the slide water PDVs shown above does not exceed 16 MPa. If the pressure is increased beyond this limit, the clearance must be reduced significantly. This would require advanced manufacturing techniques and

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result in much higher manufacturing costs. Furthermore, reducing the clearance could cause stiction and wear of the spool and the sleeve [15]. In comparison to slide valves, seat valves offer an effective solution to avoid these issues. Two companies, Tiefenbach and Hauhinco, have developed a 2/2 seat-type water PDV that can address these problems. The valve is powered by a proportional electromagnet, and an amplifying rod is used to increase the electromagnetic force. A displacement sensor is also used for closed-loop control to counteract excessive friction. This valve demonstrates good dynamic performance, but the amplifying rod reduces the output displacement while amplifying the force. As a result, the pressure is 32 MPa and the flow is limited to 15 L/min. Furthermore, the two companies used this valve as a pilot valve to control a 2/2 main valve, with a flow of 6000 L/min [16].

As digital technology continues to advance, a new type of seat valve for water PDVs with a digital high-speed on/off valve as the pilot stage is becoming more prevalent [17]-[19]. Park [20] has developed a 2/2 water PDV that incorporates a high-speed electromagnetic on/off valve as the pilot stage. This valve is controlled by a PWM signal to regulate the pressure in the control chamber. The flow of this valve is 14 L/min. Liu [21] and his team has developed a digital water PDV that incorporates a pilot valve to regulate the displacement and flow of the main spool. This is achieved by controlling several high-speed switching valves, allowing for a maximum flow rate of 87 L/min. Additionally, Linjama [22] has integrated numerous high-speed digital valves into a single valve house, creating a valve array that can produce varying flow rates by controlling different numbers of digital valves. However, all water PDVs that use digital valves as the pilot stage are limited to a 2/2 function, which does not meet the requirements of the HPRS. This is due to the disadvantage of 2/2 high-speed digital valves, which can cause impact loading and pressure variations, ultimately reducing the control accuracy [23].

In recent years, various types of motors have been increasingly used as electromechanical converters for high-flow valves in order to achieve large stroke, high output force, and improved control accuracy [24-25]. Gong and Zhang [26] have designed a 2/2 water PDV that is directly driven by a voice coil motor. The valve port features a spherical sealing structure, resulting in excellent sealing performance. With a maximum displacement of 1 mm, maximum pressure of 16 MPa, and flow rate of 16.8 L/min, this valve offers improved performance in terms of stroke, force, and accuracy.

In summary, there are currently two types of water PDV available: the 2/2 valve and the 4/3 valve. However, neither of these valves can meet the control requirements of the HPRS. The 2/2 water PDVs use digital high-speed on/off valves as the pilot stage, which results in high pressure and flow shocks that reduce the control accuracy. Additionally, achieving the 3/2 function requires combining multiple high-speed on/off valves, which increases the control difficulty. The current 4/3 water PDVs are capable of meeting the direction switching requirement. However, gap sealing is used on the valve which carries the potential risk of leakage, thereby limiting the pressure level that can be applied. Currently, the sealing pressure of the gap does not exceed 16 MPa.

To address this issue, a novel 3/2 seat water PDV has been developed for the HPRS in this study. The key innovation of this new water PDV is the incorporation of a special three-core follow-up servo mechanism comprising the pilot inlet valve, the pilot outlet valve, and the main inlet spool. This mechanism provides very high control accuracy of the valve opening. The following sections will introduce the structure, mechanism, and performance of this newly developed valve.

II. Mechanism and Model

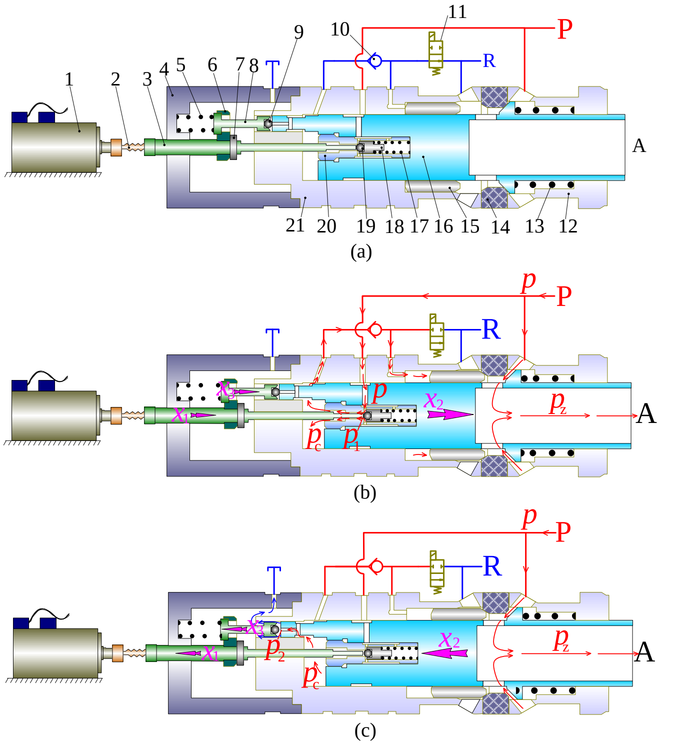
A. Structure and Components

Fig. 1 displays the structure of the new 3/2 water PDV, which comprises the three primary units found in other proportional valves: the electromechanical converter, pilot valve, and main valve. The electromechanical converter of the new water PDV is composed of the DC servo motor 1, screw nut 2, and drive rod 3. The screw nut 2 plays the role of transforming the rotary motion of the DC servo motor 1 into the linear motion of the drive rod 3. It should be noted here that there is no special requirement on the response speed of the water PDV because the HPRS moves circularly at extremely low frequency, so the DC servo motor can meet the conditions of the HPRS. The pilot valve is made up of two sub-units: the pilot inlet unit and the pilot outlet unit, both of which are integrated within the main valve. The pilot inlet unit comprises the pilot inlet valve 19(a ceramic ball), spring b 17, the spring seat 18, and the pilot seat 20. Meanwhile, the pilot outlet unit is composed of the pilot outlet valve 9 (another ceramic ball), spring a 5, and the follow rod 8. The double-hole disc connector 6 is secured onto the drive rod 3 using a lock nut 7. The follow rod 8 is then inserted through the eccentric hole of the connector 6. The pilot inlet unit is aligned with the main valve, while the pilot outlet unit is not. The main valve is composed of two spools: the main inlet spool 16 and the main outlet spool 15. To control the valve, a check valve 10 and an on/off valve 11 are utilized.

B. Control Mechanism

1) Opening process

At the beginning of the process, the pilot inlet valve 19, pilot outlet valve 9, and main inlet spool 16 are in their closed state due to the forces of their respective springs. Meanwhile, the main outlet port R is open, as depicted in Fig. 1(a). As the DC servo motor 1 rotates in the positive direction, the drive rod 3 triggers the pilot inlet valve 19 to move a certain distance to the right. This movement results in the opening of the pilot inlet port to a certain degree. Subsequently, the high-pressure liquid flows into the control chamber of the main valve, as shown by the arrows in Fig. 1(b). It firstly pushes the main outlet spool 15 to the right and closes the main outlet port R, then pushes the main inlet spool 16 and the pilot seat 20 to the right simultaneously. As a result, the P-A channel of the main inlet port gradually opens to a certain degree, while



1-DC servo motor, 2-screw nut, 3-drive rod, 4-sleeve, 5-spring a, 6-connector, 7-lock nut, 8-follow rod, 9-pilot outlet valve, 10-check valve, 11-on/off valve, 12-sleeve, 13-main spring, 14-main seat, 15-main outlet spool, 16-main inlet spool, 17-spring b, 18-spring seat, 19-pilot inlet valve, 20-pilot seat, 21-sleeve

Fig.1. Structure of the new water PDV with the special servo mechanism, (a) initial state, (b) opening process, (c) returning process

the opening of the pilot inlet port gradually reduces to zero. This process allows the main port to remain stably open. During this process, the pilot outlet valve 9 moves in sync with the main inlet spool 16 under the action of spring a 5 and the follow rod 8. This ensures that the main outlet port remains closed all the time. With each repetition of this process, the main inlet port gradually opens up further. It's worth noting that the main inlet port's opening is directly proportional to the distance that the pilot inlet valve 19 moves.

2) Returning process

When the motor 1 rotates in the opposite direction by a certain angle, the follow rod 8 and drive rod 3 move to the left by a certain distance, as depicted in Fig. 1(c). As a result, the drive rod 3 separates from the pilot inlet valve 19. At this point, the pilot inlet valve 19 is pushed to the left by spring b 17, closing the pilot inlet port. Additionally, the pilot outlet valve 9 moves leftward under the pressure of the main control chamber, thus opening the pilot outlet port. This allows the liquid to flow out along the path indicated by the arrows in Fig. 1(c). As a consequence, the main inlet spool 16 also moves to the left by a certain distance. During operation, opening of the pilot outlet port gradually decreases until it is completely closed. At that point, the opening of the main inlet spool 16 remains stable. The motor 1 continues to repeat this process, resulting in a gradual reduction of the opening and flow of the main inlet spool 16.

The new water PDV is a 3/2 valve. Double valves independently control the two ports of an actuator. One valve supplies the liquid and the other one discharges the liquid. Therefore, the on/off valve 11 is in a normally open state to ensure the control chamber of the main outlet spool 15 is in an unloaded state. Thus, the liquid in the low-pressure chamber of the actuator can return through the A-R channel. Whenever the opening of port P-A is not zero, the on/off valve 11 must be energized to ensure that the main outlet spool 15 is on the right and closes the low-pressure port of the main valve.

C. The Key Feature and Innovation of the Valve

To summarize, the new 3/2 water PDV boasts several key features and advantages:(1) Its pilot inlet valve 19, main inlet spool 16, and pilot outlet valve 9 work in unison to form a unique three-core follow-up servo mechanism. During the opening process, the pilot outlet valve 9 mirrors the movements of the main inlet spool 16, while the main inlet spool 16 follows the actions of the pilot inlet valve 19. When returning, the main inlet spool 16 tracks the pilot outlet valve 9, while the pilot outlet valve 9 follows the drive rod 3. This special mechanism provides exceptional control accuracy of the spool position, which is a significant improvement over the existing water PDVs. Additionally, the pilot inlet and outlet units are integrated within the main valve, resulting in a highly compact structure. These features make the new water PDV a standout option in the market. (2) The newly developed water PDV is of 3/2 function, expanding and enriching the existing range of water PDVs. (3) The DC servo motor 1 and screw nut 2 combination serve as an electromechanical converter that produces an output force that is unaffected by the stroke. This overcomes the inherent contradiction between solenoid force and stroke in conventional proportional valves that use proportional electromagnets, and is especially suitable for high-flow valves. (4) The pilot valve and main valve both utilize seat-type valves, which effectively eliminate the issue of limited sealing pressure that is commonly found in slide valves.

D. Model

Based on the working principle of the new water PDV, the relationship between the pilot inlet valve, main inlet spool, and pilot outlet valve can be simplified and depicted in Fig. 2.

When the main inlet spool is opened, the force exerted by the main spring is negligible in comparison to the high-pressure liquid force and can be disregarded. As a result, the

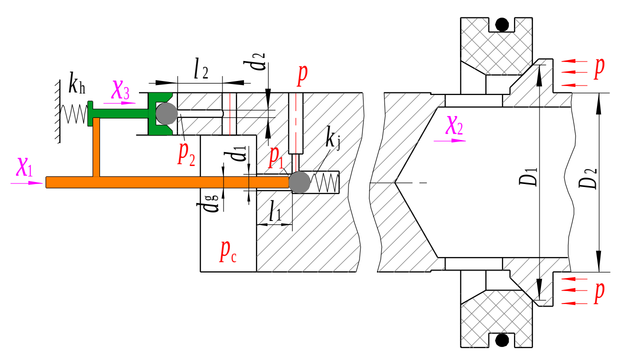


Fig.2. Diagram of the simplified three-core follow-up mechanism

opening pressure can be defined as:

$$p_c = \frac{\pi}{4}(D_1^2 - D_2^2)\cdot p / A_c \tag{1}$$

where, p_c is the pressure of the main control chamber, D_1 is the sealing tape diameter between the seat and the poppet valve. D_2 is the stem diameter of the poppet valve. p is the supply pressure. A_c is the control chamber area of the main inlet valve.

During the opening process of the main inlet spool, the flow of the pilot inlet port is:

$$q_i = C_i A_i \sqrt{2(p - p_1)/\rho} \tag{2}$$

$$A_i = \pi (d_1/2)^2 \left[a^2 + 1 - (d_b/d_1)^2 / \sqrt{a^2+1}\right] \tag{3}$$

where, $a = 2(x_1 - x_2)/d_1 + \sqrt{(d_b/d_1)^2 - 1}$. d_b is the ball diameter. d_1 is the seat diameter of the pilot inlet valve. A_i is the flow area of the pilot inlet port. C_i is the flow coefficient of the pilot inlet port. p_1 is the outlet pressure of the pilot inlet port. ρ is the liquid dengsity. q_i is the flow of the pilot inlet port. x_1 and x_2 are the displacements of the drive rod and the main inlet spool.

The flow of the annular gap in the inlet part is

$$q_1 = \frac{\pi d_1 (d_1 - d_g)^3}{96 \mu l_1}(p_1 - p_c) \tag{4}$$

$$q_i = q_1 \tag{5}$$

where, q_1 is the flow of the annular gap in the inlet part. d_g is the diameter of the drive rod. μ is the liquid viscosity. l_1 is the length of annular gap in the inlet part.

The flow of the damping hole in the outlet part is

$$q_2 = \frac{\pi d_2^4}{128 \mu l_2}(p_c - p_2) \tag{6}$$

where, d_2 is the diameter of thedamping hole in the outlet part. l_2 is the length of fixed damping hole in the outlet part. q_2 is the flow of the fixed damping hole in the outlet part.

The flow of the pilot outlet port is

$$q_o = C_o A_o \sqrt{2 p_2/\rho} \tag{7}$$

$$A_o = \pi (d_2/2)^2 \left[a^2 + 1 - (d_b/d_2)^2 / \sqrt{a^2+1}\right] \tag{8}$$

where, $a = 2(x_2 - x_3)/d_2 + \sqrt{(d_b/d_2)^2 - 1}$. A_o is the flow area of the pilot outlet port. C_o is the flow coefficient of the pilot outlet port. q_o is the flow of the pilot outlet port. x_3 is the displacement of the follow rod.

What's more,

$$q_o = q_2 \tag{9}$$

Motion equation of the pilot outlet valve can be expressed as

$$k_h(s_x - x_3) - p_2\frac{\pi d_2^2}{4} - f_f = m_3 \frac{d^2 x_3}{dt^2} \tag{10}$$

where, t is the time. s_x is the spring pre-compression of the pilot outlet valve. k_h is the spring stiffness of the pilot outlet valve. f_f is the friction force on the follow rod.

The continuity equation of the main control chamber is

$$q_1 = A_c \frac{dx_2}{dt} + \frac{(V_0 + A_c x_2)}{\beta}\frac{dp_c}{dt} + q_2 \tag{11}$$

where, V_0 is the initial volume of the control chamber. β is the water bulk modulus.

For the main inlet spool,

$$m_2 \frac{d^2 x_2}{dt^2} = F_p - F_w - F_m - c_f \frac{dx_2}{dt} - k_z(s_z + x_2) \tag{12}$$

where, m is the mass of the main inlet spool. s_z is the pre-compression of the main inlet spool. c_f is the viscous friction coefficient of the main inlet spool. k_z is stiffness of the main valve. F_p , F_w , F_m is the pressure force, flow force and friction force on the main inlet spool respectively.

Flow of the main inlet port can be written as

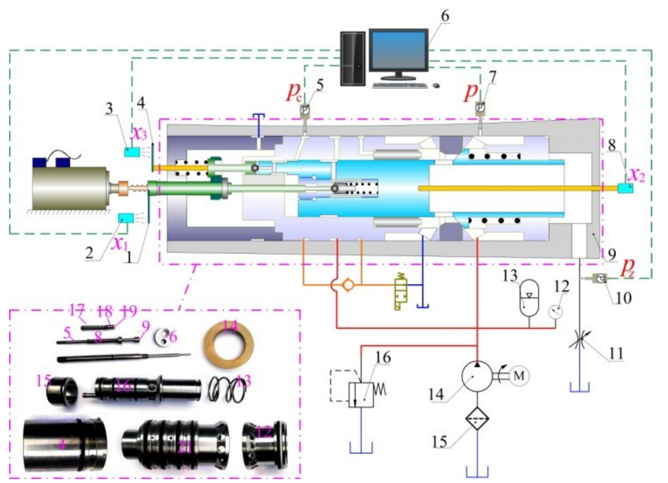
$$Q = C_q A(x_2)\sqrt{2(p - p_z)/\rho} \tag{13}$$

where, c_q is the flow coefficient of the main inlet port. $A(x_2)$ is the flow area of the main inlet port. p_z is the pressure of port A.

III. Performance Test

A. Experimental Setup

The key and distinctive aspect of the high-pressure and high-flow water PDV analyzed in this paper is the three-core follow-up servo mechanism, which plays a crucial role in determining the valve's performance. Therefore, the primary indicators that need to be measured are the displacements of the drive rod, follow rod, and main inlet spool. Fig. 3 illustrates the testing system used for the high-flow water PDV. To measure the displacements of the drive rod and follow rod, two laser displacement sensors are employed with a range of 50 mm and an accuracy of 0.1 μm, while an LVDT is used to measure the displacement of the main inlet spool, with a range of 10 mm and an accuracy of 0.2 μm. In addition, a pressure sensor is used to measure the pressure of each valve port. A set of accumulators provide high flow to the tested prototype. The designed valve's nominal pressure is 31.5 MPa, but due to limitations in experimentation and funding, the maximum



1,4-sheet, 2,3-laser displacement sensor, 5,7,10-pressure transducer, 6-data acquisition computer, 8-LVDT, 9-tested prototype, 11-trottle valve, 12-pressure gauge, 13-accumulator, 14-pump, 15-filter, 16-relief valve

Fig.3. Testing system of the high-flow water PDV

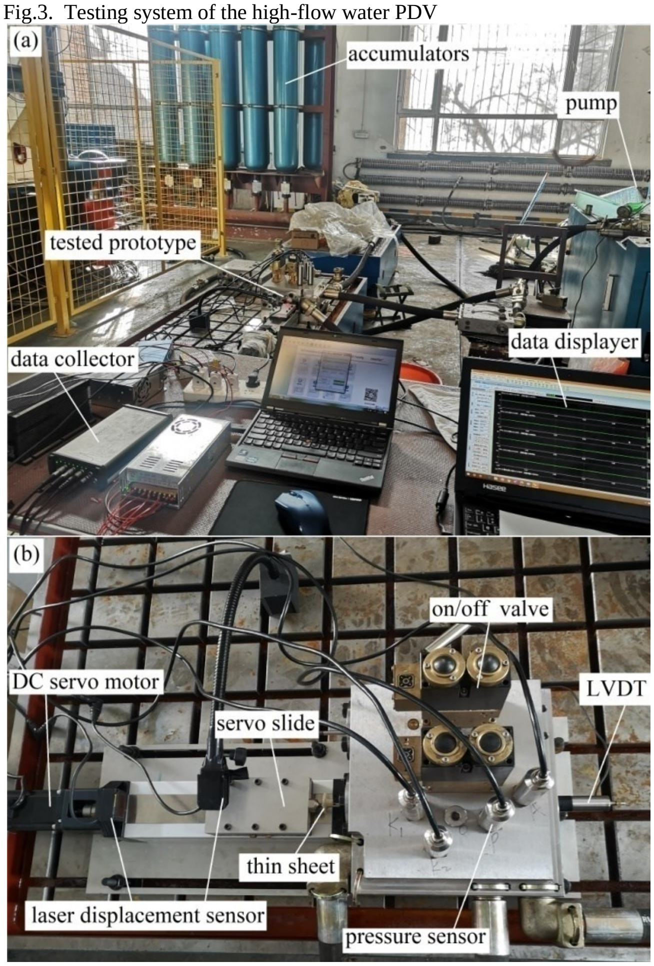


Fig.4. On-site testing pictures, (a)full scene, (b)tested prototype

pressure that the experiment can achieve is 16 MPa. Once the experimental prototype is designed and manufactured, the testing process begins. The left bottom corner of Fig. 3 shows the various key parts of the new valve in Fig. 1. The marked numbers are consistent with those in Fig. 1. It should be noted that the material of the prototype needs to have anti-rust function because of water medium, so the materials of the valve cores is 9Cr18 which is stainless steel. The valve seat is made of PEEK and the sealing rings are made of nitrile butadiene rubber. The surface roughness of all moving valve cores is 0.4 µm. Fig. 4 is the real on-site testing pictures.

B. Result Analysis

(1) Step response performance

By adjusting the new water PDV to the initial state as depicted in Fig. 1(a), the servo motor is then utilized to implement a reciprocating step drive on the drive rod with a reciprocating stroke ranging from 1 mm to 4 mm. Fig. 5 illustrates the step test characteristics of the primary inlet spool under the four strokes. The results demonstrate that the main inlet spool exhibits a rapid response speed, with an opening time of approximately 100 ms when the stroke is 4 mm and a shorter returning time. Nevertheless, during the returning process, the main inlet spool experiences difficulty in fully returning to the initial position in all four stroke conditions, with deviations that are almost identical. The

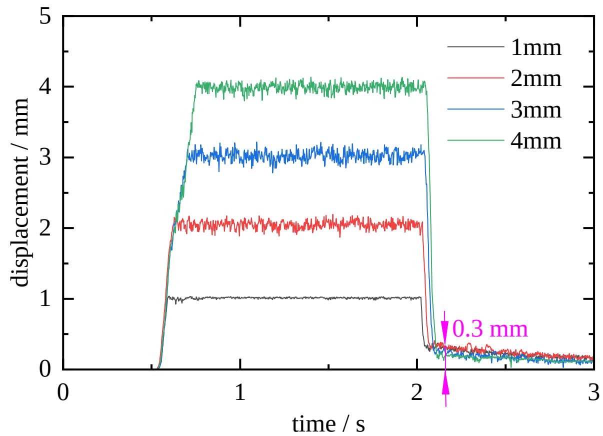


Fig.5. Step test characteristics of the main inlet spool, reciprocating stroke of the drive rod ranges from 1 mm, 2 mm, 3 mm and 4 mm

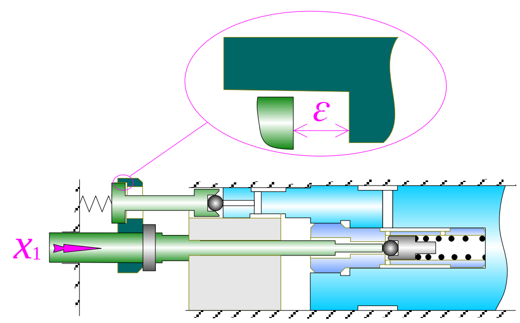


Fig.6. Schematic diagram of unavoidable installation gap

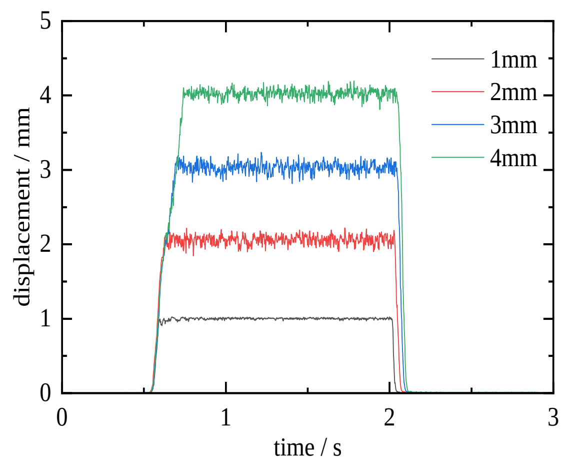


Fig.7. Step test characteristics of the main inlet spool, returning travel distances of the drive rod are 1.3 mm, 2.3 mm, 3.3mm and 4.3 mm

maximum deviation value observed is 0.3 mm. The cause of this phenomenon is depicted in Fig. 6. To ensure the proper installation of the three-core follow-up mechanism based on the relevant knowledge of the dimension chain in mechanical design, a certain gap exists between the connector and the follow rod. During the returning process, the drive rod must initially retract the connector to the left by a distance of ε=0.3 mm to compensate for the gap before the main inlet spool can track the follow rod. In other words, to restore the main spool to its original position, the drive rod must move at least 0.3 mm further than the main inlet spool during the returning process. It is important to mention that this gap value can only be determined through testing. Fig.7 displays the step test characteristics of the main inlet spool under different returning distances of the drive rod, namely 1.3 mm, 2.3 mm, 3.3 mm, and 4.3 mm. As shown in the figure, the main inlet spool can return to its initial position under all four conditions, provided that the returning distance of the drive rod is 0.3 mm greater than that of the main inlet spool. It is noted that in Fig. 5 and Fig. 7 the oscillation amplitude of the displacement signal at 1 mm is lower than the other three cases. It is mainly due to the bigger elastic oscillation of the long and thin measuring rod of the LVDT, which is caused by the bigger opening and higher turbulent flow.

(2) Ramp response performance

The water PDV is typically utilized to prolong the valve's opening and closing time, which helps in reducing mechanical impact and hydraulic shock, while improving the control accuracy of the actuator. In most cases, the PDV might be driven by the ramp signal.

Fig. 8 displays the test results of the new water PDV under four distinct ramp inputs starting from the initial position. The ramp signal's control time varies between 100 ms, 300 ms, 500 ms, and 700 ms, respectively. All four conditions have an opening distance of 4 mm and a returning distance of 4.3 mm. Generally, the hydraulic shock value decreases with the increasing control time. The results shown in Fig. 8(a) indicate that with a control time of 100 ms, the main inlet spool can fully return to its initial position, and the pressure in the control chamber drops to zero rapidly. However, for control times of 300 ms, 500 ms, and 700 ms, the main inlet spool is unable to return entirely to its initial position, resulting in deviations of 0.1 mm, 0.2 mm, and 0.2 mm, respectively. Furthermore, residual pressure remains within the control chamber under these conditions. This phenomenon can be explained by the fact that when the control time t is set to 100 ms, which is similar to the step response time of the main inlet spool, the speed at which the main inlet spool returns to its initial position is very fast. As a result, the liquid within the control chamber is rapidly discharged. However, when the control time is set to 300 ms, 500 ms and 700 ms, the time is much longer than the step response time of the main inlet spool. As a result, the main inlet spool follows the drive rod and moves at a slower speed. Additionally, the main inlet spool is affected by the friction resistance of the seals. Especially, the viscous friction coefficient c_f, which is indicated in (12) of the model, is bigger due to the much lower viscosity water. Furthermore, there may be some inevitable eccentricity during the machining and assembly of the drive rod, follow rod and the main inlet spool, which can increase the friction resistance that is included in the item F_m in (12) of the model. The combined effects of these multiple factors result in a slower moving speed of the main inlet spool when the friction force is significant and the control time is long. This can lead to insufficient closing ability at the end of the returning process. Furthermore, the opening of the pilot outlet port is small, making it difficult to discharge the liquid and depressurize the control chamber. As a result, the returning movement of the main inlet spool becomes obstructed. As

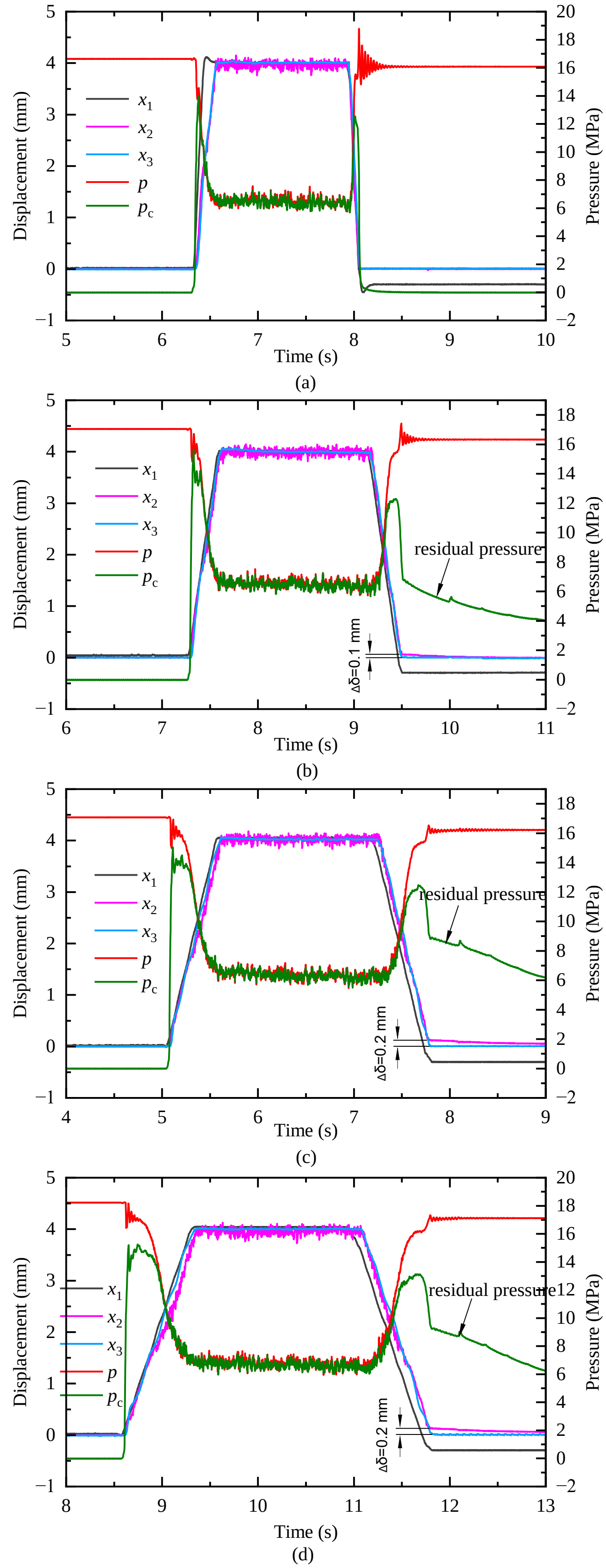
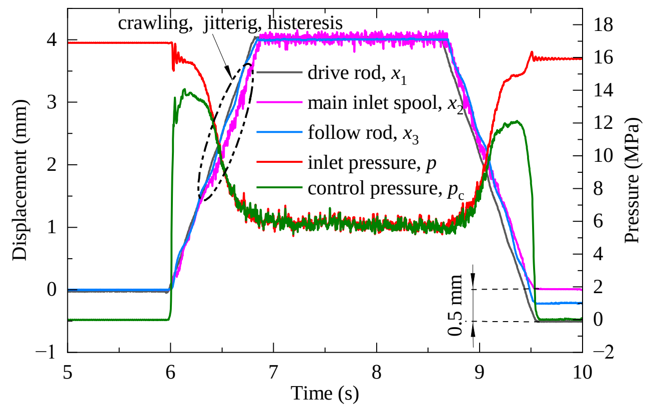


Fig.8. Test performance of the main inlet spool under different ramp signals with the returning distance 4.3 mm, (a)t=100 ms ,(b)t=300 ms, (c)t=500 ms,

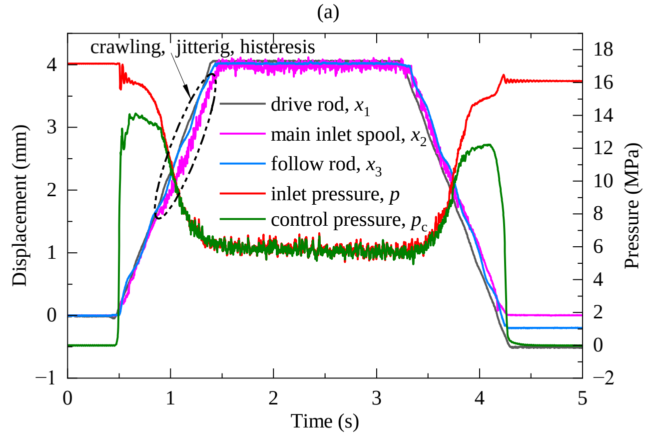
(d)t=700 ms; x_1-drive rod, x_2-main inlet spool, x_3-follow rod, p-inlet pressure, p_c-control pressure

shown in Fig. 8(b)-(d), the pressure in the control chamber decreases slowly, and the main inlet spool cannot return to its initial position in a timely manner.

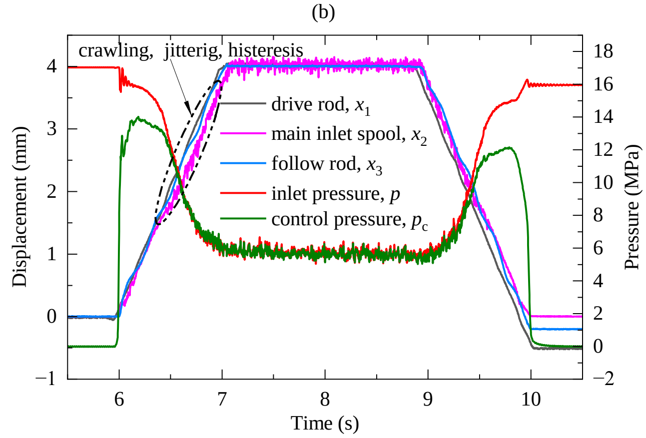
To address this issue, the drive rod can be pulled slightly to the left at the end of the returning process, which will release pressure in the control chamber. As the duration of control time increases, the effects of this phenomenon become more pronounced. Therefore, control times of 800ms, 900ms, and 1000ms have been chosen for analysis. Additionally, the returning distance has been increased by 0.2mm to account for these effects, resulting in a total returning distance of 4.5mm for the drive rod. The test performances of the main inlet spool under these conditions will be evaluated. The findings are



(a)



(b)



(c)

Fig.9. Test performance of the main inlet spool under different ramp signals with the returning distance 4.5 mm, (a)t=800 ms ,(b)t=900 ms, (c)t=1000 ms

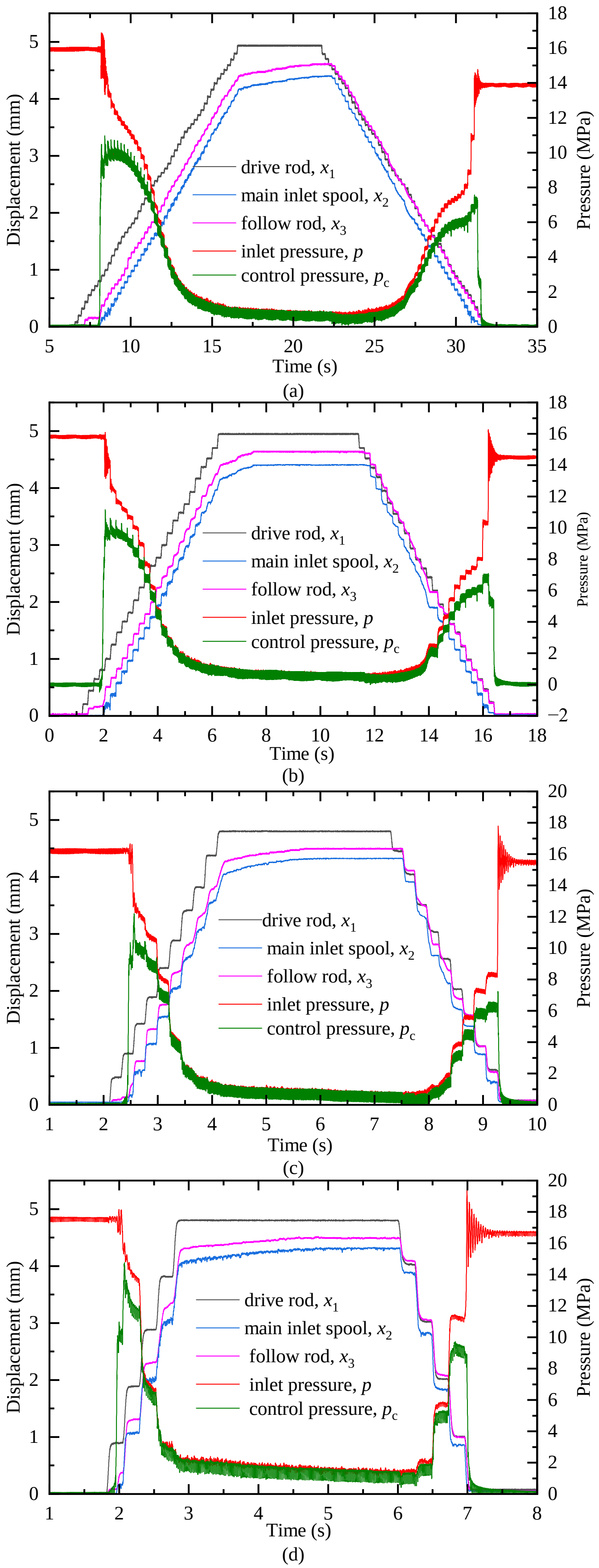


Fig.10. Continuous small-step test performance with the reciprocating stroke 4.5 mm, (a)0.1 mm / step ,(b)0.2 mm / step, (c)0.5 mm / step, (d)1 mm / step

presented in Fig. 9, which demonstrate that when the returning distance of the drive rod is set to 4.5mm, the pressure in the control chamber can be effectively discharged. Under all three conditions, the main inlet spool can return to its original position completely. However, during the opening process, noticeable jittering, hysteresis, and crawling phenomena are observed on the follow rod and main inlet spool. This is mainly due to misalignment between the pilot outlet unit and the main valve, resulting in the axis of the follow rod being eccentric with that of the guide hole on the main inlet spool due to errors during machining and installation. This leads to additional friction force which is contained in the item f_f in (10) of the model on the follow rod, causing insufficient following capacity when it moves slowly with the main inlet spool. Consequently, the pilot outlet port fails to close in a timely manner, leading to insufficient pressure in the control chamber. This, in turn, causes the jittering, hysteresis, and crawling phenomena on the main inlet spool and follow rod during the opening process.

(3) Continuous small-step response performance

The ramp control mode may experience crawling, jittering, and hysteresis due to friction. To overcome the effects of friction in proportional valves, a flutter signal is often added to the control signal. As a solution to this problem, the continuous small-step control mode has been adopted. Based on the research mentioned above, to achieve symmetrical control of the opening and returning process, it is necessary to pull the drive rod to the left by an additional 0.5 mm from the initial position shown in Fig. 1(a), ensuring that the control chamber remains in the pre-discharge state. This new position serves as the new zero point for both the DC servo motor and the water PDV. To implement the continuous small-step control mode, the reciprocating stroke of the drive rod is set to 4.5 mm. According to Fig. 10, the new water PDV was tested across four different step distances: 0.1mm, 0.2mm, 0.5mm, and 1mm. The figure demonstrates that the drive rod, main inlet spool, and follow rod all perform well across the four conditions, with no crawling or hysteresis resulting from friction during the opening and returning process. Additionally, the smaller the step distance, the lower the pressure shock observed at port P and in the control chamber, resulting in smoother pressure. In conclusion, the 0.1 mm small-step control mode results in minimal pressure pulsation at each valve port and accurate, timely position tracking by the three-core follow-up servo mechanism. This eliminates undesirable dynamic characteristics and ensures that the displacement of the main inlet spool is nearly continuous and smooth.

IV. Discussion

Table I presents a technical comparison between the newly developed valve and the existing water PDVs. Both the 2/2 and 4/3 water PDV fail to meet the HPRS's control demands. The former uses the digital high-speed on/off valve as the pilot stage, causing pressure and flow shocks that impair control accuracy. Meanwhile, achieving the 3/2 function necessitates combining multiple high-speed on/off valves, resulting in increased control complexity. On the other hand, the latter satisfies the direction switching requirements, but its valve design employs gap sealing, posing a potential leakage risk and limiting the applicable pressure level (currently capped at 16 MPa). The new water PDV developed by the authors in this study is a 3/2 seat-type valve equipped with a special high-precision three-core follow-up mechanism. This design is specifically tailored to handle both high pressure and high flow, making it an ideal match for the HPRS's control demands. In this study, the control performance of the new valve is improved, but the friction force still exists, which has wear problems. In order to make the new valve have a long service life in practical applications, reasonable technical measures to reduce friction force need to be studied in the near future.

V. CONCLUSION

In this study, a highly compact 3/2 water PDV with a special high-precision three-core follow-up mechanism was proposed to achieve low-impact, precise, and safe control of the HPRS by optimizing the valve's control characteristics. The focus of the study is the position tracking characteristics of the three-core follow-up mechanism. An experimental prototype was manufactured and tested, leading to two important findings. Firstly, the initial position of the DC servo motor and the three-core follow-up mechanism is crucial the the water PDV's performance. It should ensure that the control chamber of the main valve is in a pre-discharge state to ensure reliable closing ability of the main inlet spool during its returning process. Secondly, additional friction caused by machining and installation errors can easily cause crawling and hysteresis phenomena on the valve cores under the ramp control mode. These issues can be effectively overcome by using a continuous small-step incremental control mode.

The newly developed highly compact 3/2 water PDV and its control strategy offer a new approach for the precise, digital, and safe control of the HPRS, particularly under high-pressure and high-flow conditions. This expands and enriches the range of water PDV types and makes it applicable to a wide range of industrial fields. Given the extensive promotion and application of the new water PDV in industrial fields, our future work will primarily focus on improving and optimizing its structure and processing technology to reduce friction, enhance service life, and ensure reliability.

TABLE I
TECHNICAL COMPARISON OF THE PRESENT WATER PDV

Inventor	Spool and sleeve	Valve port	Main valve	electromechanical converter	Drive type	Pressure/Mpa	Flow/L/min
Water Hydraulics	gap	plane	4/3	proportional electromagnet	direct drive	16	30
Ebara、KYB	gap	gap	4/3	proportional electromagnet	direct drive	14	30
Tiefenbach、Hauhinco	–	Seat valve	2/2	proportional electromagnet	direct drive	32	15
	–	Seat valve	2/2	high speed on/off valve	pilot operated	32	6000
Park	–	Seat valve	2/2	high speed on/off valve	pilot operated	14	11
Liu	gap	gap	4/3	high speed on/off valve	pilot operated	1.5	87
Gong	–	Seat valve	2/2	voice coil motor	direct drive	16	17
Liao, et al (this study)	**seal**	**poppet / ball valve**	**3/2**	**servo motor+screw nut**	**pilot operated**	**31.5**	**500**

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