Design and Research on New Type of Balance Valve Suitable for Miter Gate Hoist of Ship Lock

Fangchao Yan, Tianwei Huang*, Hanzhao Li

Three Gorges Navigation Authority, Hubei Yichang, China
*Corresponding author's email: 1484210947@qq.com

Abstract. The miter gate hoist of the ship lock is the core component of the ship lock operation, and its opening and closing operation requires the gates on both sides to be synchronized and adapt to the external load changes caused by undercurrent, over-irrigation and over-discharge. A new type of pilot plug-in balance valve, designed for a 1:50 external load ratio, has been developed. We have constructed an AMESim model for the new pilot plug-in balance valve and hydraulic gate system, and conducted simulations to study its dynamic characteristics and anti-interference capabilities. The results indicate that the new type of pilot plug-in balance valve exhibits good dynamic performance. It can adapt to changes in external loads and is insensitive to them. The hydraulic hoist will not exhibit hydraulic cylinder crawling or herringbone gate drift under external loads.

Keywords: New Pilot-injected Balance Valve; Dynamic Behavior; AMESim Model; Sliding Phenomenon; Drift Phenomenon

1 Introduction

The performance of the balancing valve directly affects the reliability and stability of the entire hydraulic system operation. Unlike hydraulic balance valves commonly used in crane engineering machinery, ship lock inclined gate hoisting machinery is usually a double-acting, center-supported, bi-directional swing, horizontal installation hydraulic direct-link hoisting machinery that mainly bears horizontal loads[1-2].

The ship lock miter gate equipment is a large inertia load, the miter gate operation process requires smooth action, small impact, and the operation of the double miter gate coordination and synchronization, a lock miter gate is often subject to changes in the opening and closing of the variable load, the hydraulic hoist crawling, shaking and miter gate drift and other problems[3-4].The reverse load is large when the hermitage door is open, and the operation process requires the cylinder to be stable, the water level changes regularly throughout the year, and the repeated changes of the external load put forward higher requirements for the performance of the balance valve. At present, the high-end balance valve on the market basically relies on foreign products, and faces problems such as difficulties in purchasing spare parts and uncontrollable delivery[5-6].At the same time, the domestic balance valve can not meet the requirements of stability and fast response, and has the characteristics of small flow and high pressure loss[7-8]. In practical application, two or more balancing valves are used in parallel to meet the large flow demand of the miter gate hoist with high operating pressure, so it is urgent to develop and design a new balancing valve suitable for the miter gate hoist.

In order to solve the problem of crawling and drifting of the hydraulic hoist of the miter gate of the ship lock, a new type of pilot plug-in balance valve adapted to the operating condition of the ship lock is studied and designed.

2 The Hydraulic Hoist for the Miter Gate of the Ship Lock

2.1 System Principle

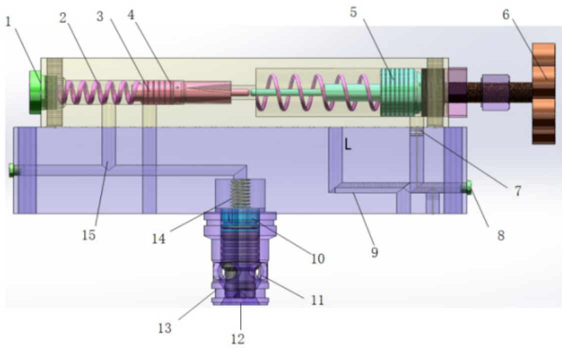
The normal operation of the hydraulic hoist for the miter gate of the ship lock is the key to ensuring the safe and efficient operation of the ship lock. The hoist has a large scale and complex structure. A high pressure and large flow stepless speed control circuit for the miter gate hoist of ship lock is formed by the combination of the high pressure and large flow piston pump and the integrated control structure of the two-way plug valve[2].

At present, the hydraulic hoist of the miter gate of a ship lock can not adapt to the large change of water level, including the crawling of the miter gate hoist and the drifting of the miter gate closing[3].The crawling phenomenon of the hoist gate refers to the alternating movement of the hydraulic cylinder during movement, where the displacement jumps and stops, or moves faster and slower. Once the operation of the hoist miter gate of the ship lock appears crawling and drifting, it will cause the miter gate to be eventually mislocated, the water is unstable, the closing signal cannot be triggered, and the automatic operation process of the ship lock will be interrupted. In serious cases, it will also lead to the guide wheel card jamming, damage the metal structure of the miter gate, reduce the efficiency of safe operation, and bring about the risk of safe operation.

In this paper, a new type of pilot-injected balance valve is designed to adapt to the operating conditions of ship locks, in response to the failure phenomenon of the balancing valve due to the failure of the balancing valve to hold the speed during the fluctuation of the outer load of the sluice gates. The proposed high flow balance valve for ship locks can directly increase the diameter of the main valve component by enlarging the main valve core, thereby increasing the flow of the high flow balance valve for ship locks. It has the characteristics of easy adjustment and can better meet the high flow demand of ship lock opening and closing machines with high operating pressure. It also has the feature of low pressure loss.

The Principle of a New Type of Pilot Plug-in Balance Valve. The structure of a new type of pilot plug-in balance valve is shown in Figure 1. The control mode is pilot type, the connection and installation mode is cartridge type, the main spool is innovatively designed auxiliary spool on the basis of the spool structure of two-way cartridge valve. The valve body has four oil ports, namely load port B, port A connected to the electromagnetic directional valve, pilot control port X on the end cap, and pilot drain port L located on the control piston valve body. A. B ports are all located on the main valve body. The tail of the pilot valve core is machined with threads, which can be used for installing overcompensation damping. A quick closing valve port is designed between the pilot valve core and the reset spring. The quick closing valve port is in a normally open state when the pilot valve core is not open, and in a normally closed state when the pilot valve core is open. The pilot valve core and the control piston are in a coaxial position, and there is elastic contact between the control piston and the pilot valve core. The large end of the control piston is designed with threaded holes for installing bypass damping. The control piston is reset by a reset spring. The control piston part has been designed to be detachable, allowing for the replacement of reset springs with different stiffness. The pilot control oil port is located at the right end of the valve body, and inlet damping is installed on the sealing surfaces of the upper and lower valve bodies.

Structural innovation has been carried out on the main valve core of plug-in valve, with the addition of a throttling groove at the front end to achieve throttling effect. The throttling groove is evenly distributed at the front end of the main valve core. The auxiliary valve core is connected to the main valve core, and a feedback valve port is designed on the auxiliary valve core to enhance the overcompensation effect. Eight small holes with a diameter of 1mm are designed on the inner side of the shoulder as flow channels. The feedback valve port communicates with the hydraulic oil at port B, and after a certain pressure loss, it enters the spring chamber. The hydraulic oil at load port B acts on the shoulder, and its force is consistent with the opening direction of the main valve port. After the main valve is opened, the B port and A port are connected, and the overcurrent area of the main valve port determines the size of the main valve flow.



1-Plug 2-Reset spring 3-Pilot valve core 4- Pilot valve port 5-Control piston 6-Adjusting scre 7-Pilot inlet damping 8- pilot port 9- Recycle port 10-Auxiliary valve core 11-Load port (B port) 12-A port 13-Main valve sleeve 14-Main valve spring 15-Overcompensation damping

Fig. 1. Structure of a new type of pilot plug-in balance valve.

3 The Mathematical Modeling of Balance Valve

Balance valve static characteristics are mainly analyzed when the balance of the valve components movement to reach equilibrium state when the main spool and the pilot part of the force.

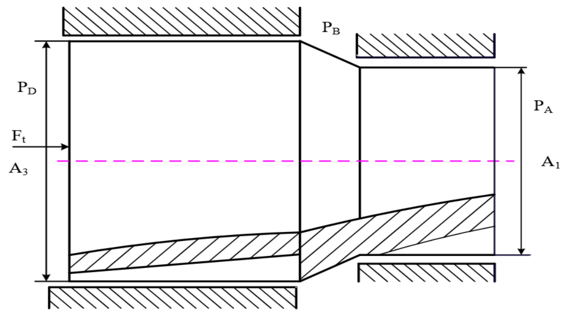


Fig. 2. Force analysis of the main valve.

The force analysis of the main valve core is shown in Figure 2. The A port of the main valve is connected to the reversing valve. When testing the single characteristic and the influence of back pressure on the balance valve, the back pressure pressure is around 2MPa, and the back pressure PA of the A port is ignored.

The main valve core is subjected to pressure PD from the spring chamber on A3, reset spring force Ft, load port pressure PB, and hydraulic force Ff. The force balance equation is:

$$P_B(A_3 - A_1) + P_A A_1 - P_D A_3 - F_t - F_f = 0 \tag{1}$$

In the formula, A_3 is the big end area of the main valve core, and A_1 is the small end area of the main valve core

The force of pilot valve core and control piston is shown in Figure 3. The pilot valve core and control piston can be regarded as a whole for force analysis.

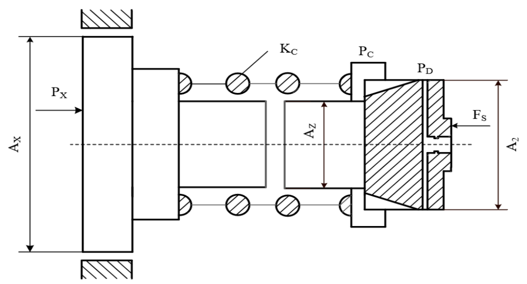


Fig. 3. Analysis of the overall force on the pilot valve and control piston.

After simplification, this whole is subjected to the pilot chamber pressure PD, the reset spring force FS, the main valve spring chamber pressure PC, and the spring force Fr of the control spring. The static pressure balance equation is as follows:

$$P_x A_x + P_c(A_2 - A_z) - P_D A_2 - F_s - F_c = 0 \tag{2}$$

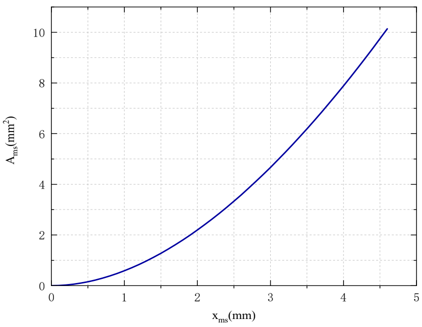
The spring force is expressed as:

$$F_s = K_s(x_s + x_{ms}) + F_{s0} \tag{3}$$

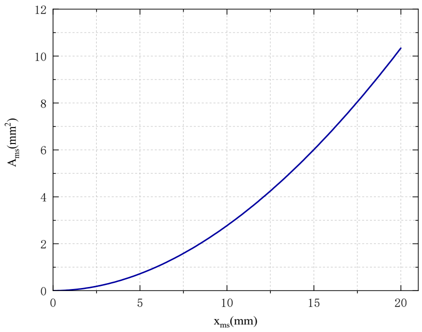
$$F_c = K_c x_p + F_{c0} \tag{4}$$

4 Simulation Analysis of Balance Valve for Ship Lock Herringbone Gate Opening and Closing Machine

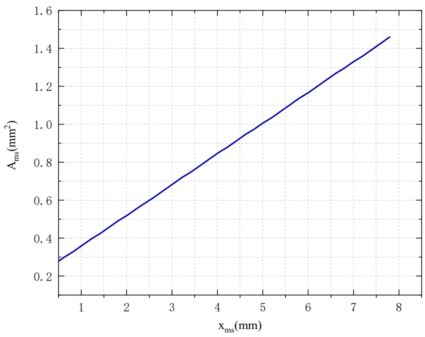
4.1 Modeling of Valve Ports for A New Type of Pilot Plug-in Balance Valve



(a)Flow area and displacement curve of the main valve port



(b)Flow area and displacement curve of the pilot valve port



(c)Flow area and displacement curve of the feedback valve port

Fig. 4. Overflow area curve of each valve port.

In order to accurately and effectively build simulation models, it is necessary to obtain the flow area displacement curves of each valve port. Draw a 3D model based on the design dimensions, calculate the flow area data of each key valve port according to the actual dimensions, and use Origin to draw the flow area displacement curve of the valve port. The main valve port, feedback valve port, and pilot valve core of the new type of pilot plug-in balance valve are all irregular valve ports. The flow area displacement curves of each valve port were calculated and shown in Figure 4.

4.2 Simulation Model of New Pilot Plug-in Balance Valve AMESim

Based on the working principle and three-dimensional model dimensions of the new pilot plug-in balance valve, the HCD simulation model of the new pilot plug-in balance valve was established using the HCD library in AMESim software, as shown in Figure 5.

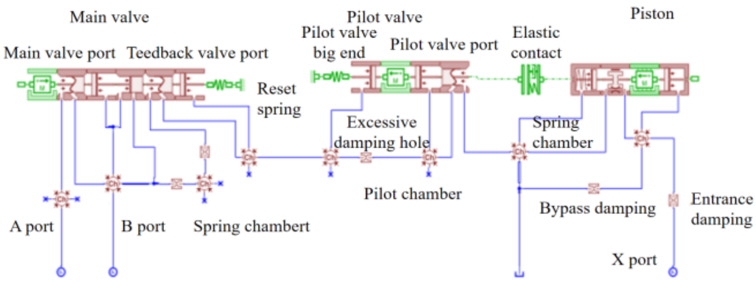


Fig. 5. Balance valve simulation AMESim model.

The new type of pilot plug-in balance valve HCD model mainly consists of a main valve core, an auxiliary valve core, a pilot valve core, a control piston, an overflow valve, a damping hole, a feedback spring, a reset spring, a pressure source, etc.

The working principle of a balance valve is complex, and there are various influencing factors. In order to facilitate modeling and identify important influencing parameters, it is necessary to ignore certain external factors and make certain idealized assumptions. In an ideal state, simplified dynamic equations for each moving component and chamber are established based on Newton's second law and flow continuity equation, and the main parameters of the balance valve simulation model are obtained, as shown in Table 1.

Table 1. Main parameter of balance valve simulation model.

Parameter Variables	Numerical Value
Hydraulic Oil Density kg/m^3	860
Hydraulic Oil Kinematic Viscosity cP	46
Hydraulic Oil Volume ModulusOf Elasticity MPa	1700
Hydraulic Oil Discharge Coefficient	0.7
Main Valve End Diameter mm	13.5

Main Valve Big End Diameter mm	17
Main ValveCore Spring N/mm	10
Main Valve Core Quality g	100
Pilot Valve Core Big End mm	14
Pilot Valve Core Small End mm	(12, 3)
Pilot Valve Core Reset Spring N/mm	10
Pilot Valve Core Quality g	120
Piston Big End mm	28
Piston Small End mm	3
Control Spring N/mm	450

Static Characteristics Analysis of a New Balance Valve.

Single Characteristic. When simulating a single characteristic, control port X is not set to control pressure (Px=0MPa), port B (load port) is connected to an oil tank or the pressure source is set to 0MPa, and the pressure at port A is set to linearly change from 0-4MPa. When drawing a one-way function diagram, the pressure at port A is taken as the x-axis, and the flow rate at port B is taken as the y-axis. The one-way pressure difference flow rate characteristic curve is drawn as shown in Figure 6.

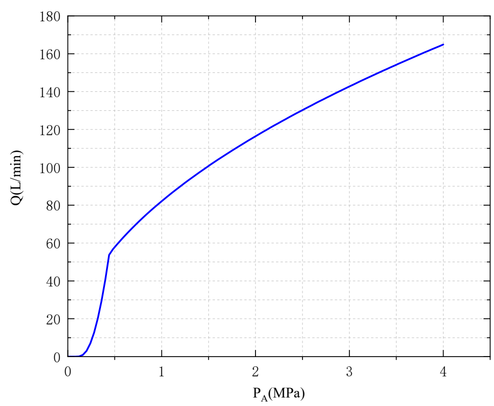


Fig. 6. Unidirectional pressure differential flow characteristic curve.

We can see that the one-way opening pressure of the balance valve is around 0.1MPa to 0.2MPa. The maximum flow rate of the balance valve main valve reaches 165L/min, which is relatively small due to the small flow area design of the main valve port. The pressure difference at the main valve port is between 0.1MPa and 0.2MPa, indicating that the one-way opening pressure of the balance valve is low, energy consumption is low, and the throttling effect is good.

Flow Control Characteristic Curve. The flow control characteristics of a balance valve refer to the relationship between the main valve flow rate and the pilot control pressure when the pilot control port pressure changes linearly when the load port pressure is at different constant values. The simulation of flow control characteristics is divided into ideal no back pressure and back pressure.

Under ideal no back pressure conditions, that is, when the pressure at port A is 0MPa, the pressure at load port B is set to 5MPa, 7MPa, 9MPa, and 11MPa respectively, and the pressure at control port Px is set to linearly change from 0 to 5MPa. The relationship between the main valve flow rate Q and the pilot pressure Px is shown in Figure 7.

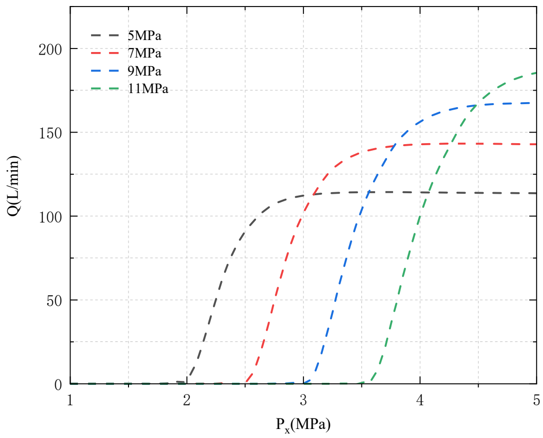


Fig. 7. Ideal No Back Pressure Flow Characteristic Curve.

As shown in Figure 7, when the load port pressure is constant, the flow control characteristics are divided into three stages. When the pilot pressure is lower than a certain opening pressure, the main valve does not open, and the flow rate through the main valve is 0L/min; In the second stage, when the pressure reaches between the opening pressure and the saturation pressure, the flow rate through the main valve increases linearly with the pilot control pressure. In the third stage, when the pressure at the pilot port reaches saturation pressure, the flow rate of the main valve remains unchanged. From the simulation results, it can be seen that the control range of the new pilot plug-in balance valve can reach 1.55MPa to 4.5MPa, with a large pressure regulation range and good balance valve regulation performance.

If the A port is set with a back pressure of 1.2 MPa, simulate the flow control characteristics again with the same ideal no back pressure setting, and obtain the relationship between the main valve flow Q and the pilot control pressure Px as shown in Figure 8. From the above figure, it can be seen that applying a back pressure of 1.2MPa significantly increases the saturation flow rate of each load pressure of the balance valve.

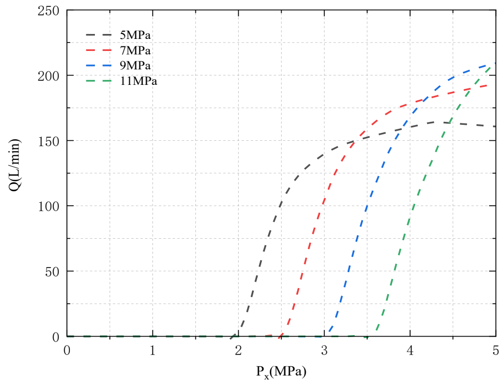


Fig. 8. Flow characteristic curve under back pressure.

For example, when the load port pressure is 5Mpa, the ideal condition flow rate without back pressure is around 115L/min. After setting the back pressure, the saturated flow rate increased to around 165L/min due to the influence applied. Similarly, the saturation pressures of 7MPa, 9MPa, and 11MPa at the load port have all increased. However, as the load pressure increases, the saturation flow rate decreases relatively. It can also be seen from the figure that the back pressure has little effect on the opening pressure of the flow characteristics.

Compensation Characteristics. The compensation characteristic of a balance valve is the relationship between the flow rate at the main valve port and the load pressure when the pilot pressure is set to a constant value and the load pressure at port B changes linearly. The pilot pressures are set at 4.0MPa, 4.3MPa, 4.8MPa, and 5.1MPa, respectively. The load port pressure is set to linearly change from 0-15MPa, with a simulation time of 10 seconds. The overcompensation characteristic curve under no back pressure is shown in Figure 9.

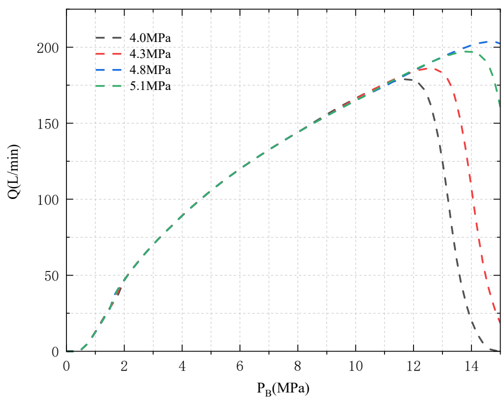


Fig. 9. Compensation characteristic curve without back pressure.

When the pilot pressure is constant, the curve basically satisfies the 1/2 power of the flow formula. As the load gradually increases, the pilot pressure remains constant, the main valve port tends to close, and the flow rate decreases. If the flow rate at the highest point is defined as Qmax and the flow rate at the lowest point is defined as Qmin, then the compensation amount is

$$\xi = \frac{Q_{max} - Q_{min}}{Q_{max}} \tag{5}$$

Taking the pilot pressure of 4.3MPa as an example, the compensation amount is 85%, and the compensation inflection point is around 12.4MPa. As the pilot pressure increases, the compensation decreases and the inflection point of the curve becomes increasingly backward.

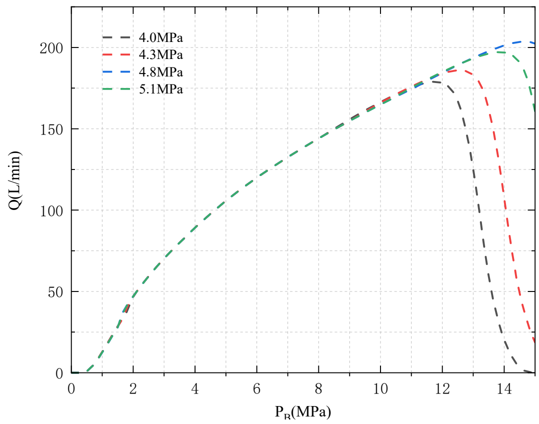


Fig. 10. Compensation characteristics at a back pressure of 2MPa.

From Figure 10, We can see that when the A port is designed with a back pressure of 2MPa, all other setting conditions remain unchanged, and the compensation characteristics under back pressure can be obtained. Similarly, the compensation amount can be calculated as 87% when the pilot pressure is 4.3MPa under back pressure. The turning point of compensation has been slightly advanced.

Dynamic Characteristics Analysis of the New Type of Balance Valve. To verify that the dynamic characteristics of the new type of balance valve can adapt to the operating conditions of the ship lock's herringbone gate opening and closing machine, it is necessary to connect the loading cylinder. Taking the herringbone gate opening condition as an example, an AMESim model of the hydraulic opening and closing machine system is built. During testing, the external load is reduced by 1:50, which means the external load is reduced by 1/50 of its original size. The same hydraulic cylinder parameters are also reduced in the same proportion. The system parameters are shown in the Table 2.

Table 2. Parameter of system

Parameter Variables	Numerical Value
Hydraulic cylinder diameter mm	122
Hydraulic cylinder piston diameter mm	80
Effective displacement of hydraulic cylinder m	2.5
Total mass of hydraulic cylinder kg	200
The displacement of variable displacement pump L/min	30
Motor speed r/min	1500

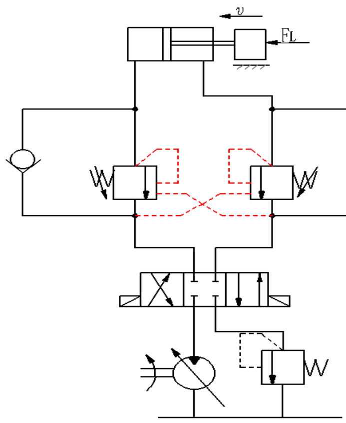


Fig. 11. Balance circuit and the relationship between negative load speed and force.

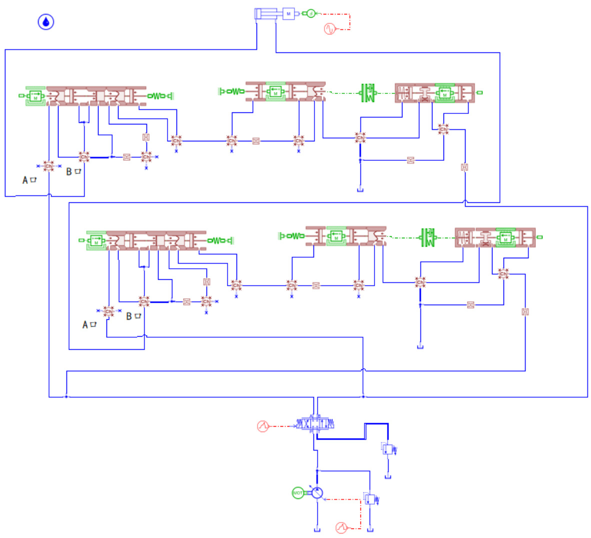


Fig. 12. Simplified balance system of hydraulic system for ship lock hydraulic opening and closing machine.

The load force acting on the hydraulic cylinder under negative load is opposite to the direction of movement of the hydraulic cylinder. When subjected to forward load, the load force on the hydraulic cylinder is in the same direction as the speed. The schematic diagram of positive load is shown in Figure 11.

The stability of the opening and closing operation of the ship lock's herringbone gate mainly relies on the bidirectional load balancing valve type balancing circuit in the hydraulic system of the hydraulic opening and closing machine. Based on the principle of the balancing circuit and the structure of the new plug-in balancing valve, a simulation model of the bidirectional load balancing valve balancing circuit was built using the hydraulic application library, mechanical library, signal library, and HCD library in AMESim, as shown in Figure 12.

To analyze the impact of sudden load changes on hydraulic cylinder speed, the external load is set as shown in Figure 13, with a simulation time of 20 seconds and a time interval of 0.1 seconds. The external load is a positive load, which is a tensile force for the hydraulic cylinder.

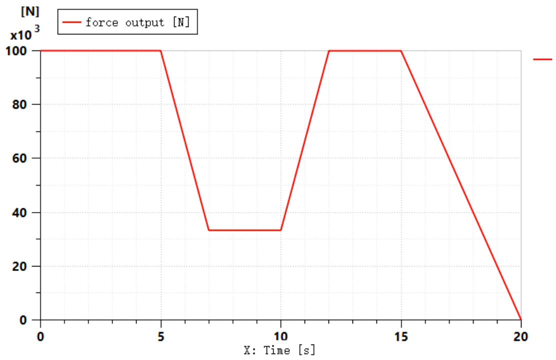


Fig. 13. Changes in external load.

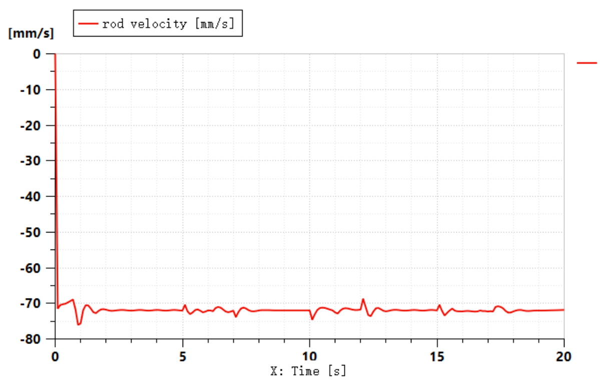


Fig. 14. Hydraulic cylinder speed variation.

The hydraulic cylinder speed variation is shown in Figure 14. The balance valve reaches a steady state after adjusting the hydraulic cylinder speed for about 1 second. The initial overshoot of the hydraulic cylinder speed is 5.6%. The operating speed of the hydraulic cylinder is 72mm/s. From this, it can be seen that when the external load suddenly changes (load inflection point), the balance valve will adjust for a certain period of time to keep the hydraulic cylinder at a constant speed of 72mm/s. For the gate under negative load opening conditions, the hydraulic cylinder of the hydraulic opening and closing machine can control the speed without any hydraulic cylinder drift phenomenon.

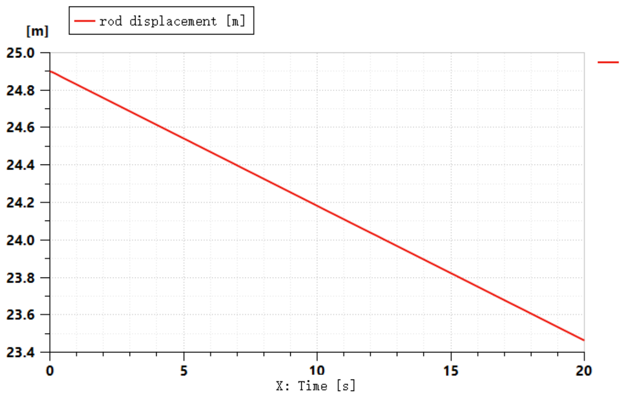


Fig. 15. Hydraulic cylinder displacement

From Figure 15, it can be seen that the displacement of the hydraulic cylinder is a linear function with linear properties, which also indicates that slight fluctuations in velocity are acceptable for the system. From the displacement of the hydraulic cylinder, there was no sign of hydraulic cylinder crawling.

5 Adaptability Analysis of Balance Valve to Different Reverse Head Operating Conditions of Miter Gate

5.1 Submerged Water Depth of 20m and Reverse Head Condition of 0.4m for the Miter Gate

In order to verify the adaptability of the balance valve during the opening of the trapdoor at different submergence depths and reverse water heads. The reverse water head generated when the downstream water level of the initial gate is higher than the upstream water level causes the gate to bear a reverse load, resulting in a negative dynamic water resistance torque. As the gate opens, it gradually becomes a positive dynamic water resistance torque. The hydrodynamic resistance moment shows a saddle shaped curve with two large ends and a small middle as the gate opening time changes, and the maximum value appears in the early stage of opening. In the middle and later stages of opening the door, the operation of the zigzag gate is relatively stable, and the dynamic water resistance torque value is basically not affected by the height of the reverse water head, stabilizing within a lower range. The load variation curve is shown in Figure 16.

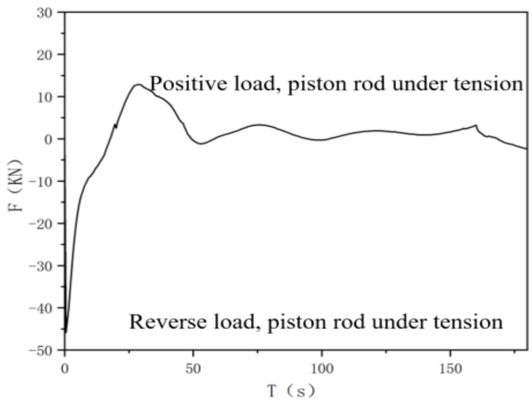


Fig. 16. External load.

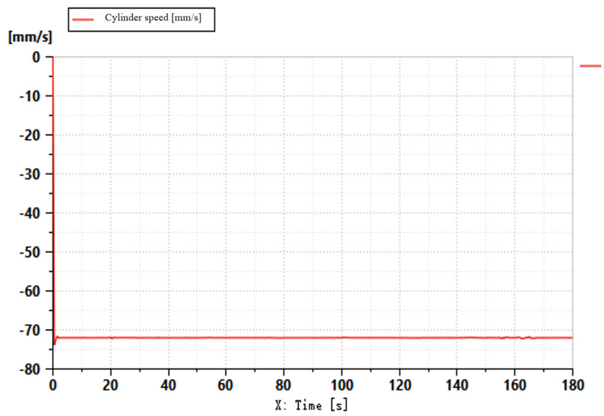


Fig. 17. Hydraulic cylinder speed.

From Figure 17, it can be seen that during the initial stage of reverse head external load loading, the hydraulic cylinder speed fluctuates to a certain extent, reaching the design speed of 72mm/s and maintaining stability.

From Figure 18, it can be seen that the pressure in the rodless chamber of the herringbone door fluctuates with the change of external load, while the pressure in the rodless chamber fluctuates slightly and remains basically constant, with a pressure of 5.5 MPa, indicating that there is no crawling phenomenon.

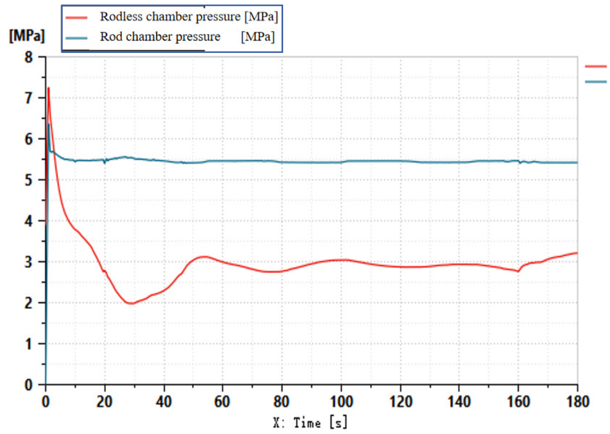


Fig. 18. Hydraulic cylinder two chamber pressure.

5.2 Submerged Water Depth of 35m and Reverse Head Condition of 0.3m for the Miter Gate

The external load force is shown in Figure 19.

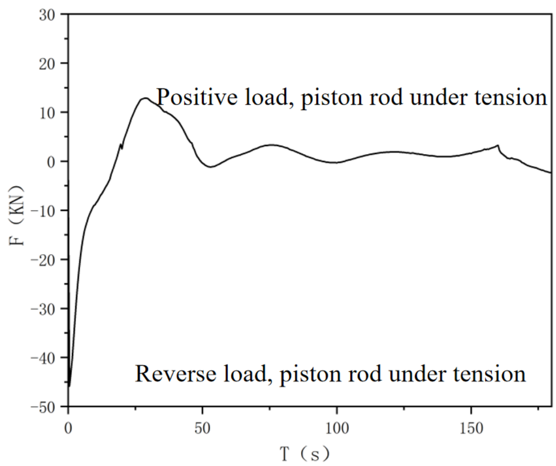


Fig. 19. External load force.

From Figure 20, it can be analyzed that when the load is switched, the hydraulic cylinder speed can still maintain the design speed of 72mm/s, which is basically constant.

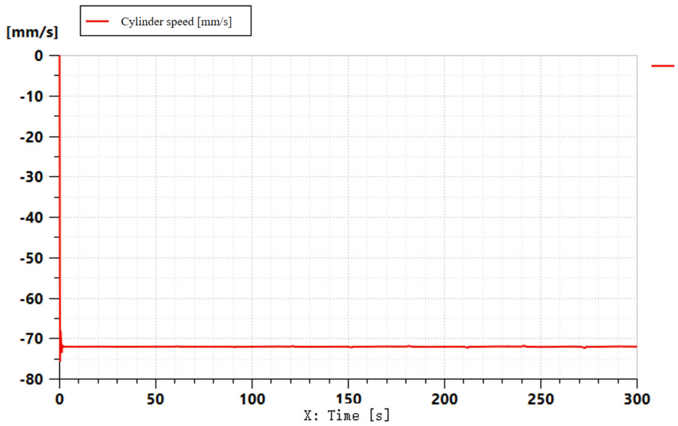


Fig. 20. Hydraulic cylinder speed

As shown in Figure 21, when the external load fluctuates between positive and reverse load, the pressure in the rodless chamber also fluctuates with the external load, while the pressure in the rodless chamber remains basically unchanged, further indicating that the gate will not experience crawling phenomenon.

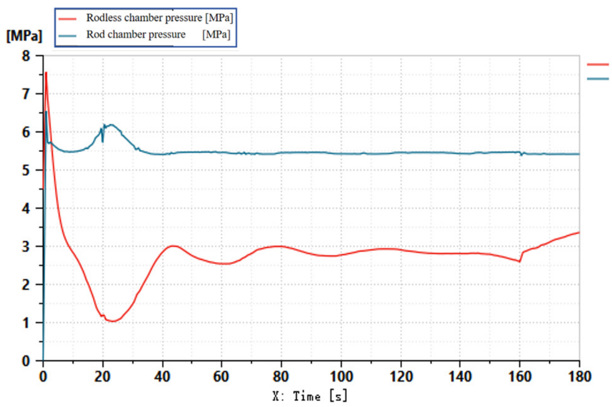


Fig. 21. Hydraulic cylinder two chamber pressure.

5.3 Submerged water depth of 35m and reverse head condition of 0.4m for the miter gate

The external load force changes under reverse head conditions with a submerged depth of 35m and 0.4m are shown in Figure 22.

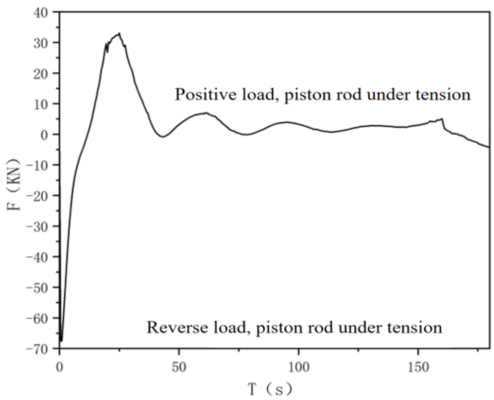


Fig. 22. External load force.

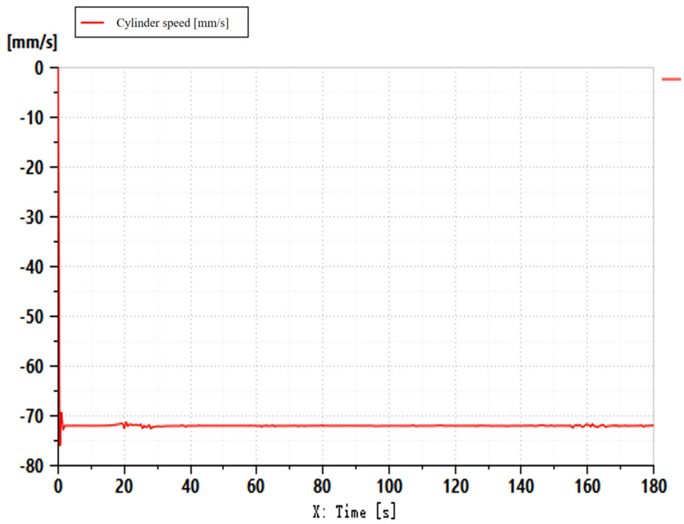


Fig. 23. Hydraulic cylinder speed.

From Figures 23, it can be analyzed that when the load switches from negative to positive, the hydraulic cylinder speed can still maintain the design speed of 72mm/s, which is basically constant. When the trapdoor model opens the door under reverse water head, there is no phenomenon of trapdoor crawling or drifting.

From Figure 24, we can see that when the external load fluctuates between positive and negative loads, the pressure in the rodless chamber also fluctuates with the external load, while the pressure in the rodless chamber remains basically unchanged, further indicating that the gate will not experience crawling phenomenon.

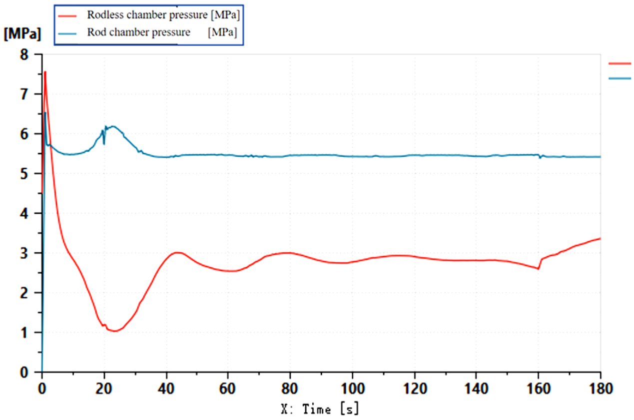


Fig. 24. Hydraulic cylinder two chamber pressure.

Comparing the adaptability of the balance valve to load changes under different submergence depths of 20m and 35m and the same reverse water head (0.4m) conditions, as well as under the same submergence depth (35m) and different reverse water heads (0.3m and 0.4m) conditions, the reverse water head and submergence depth only affect the pressure of the hydraulic cylinder rodless chamber. The new balance valve can maintain the pressure and velocity of the hydraulic cylinder rodless chamber basically constant, without the phenomenon of herringbone gate crawling and drifting.

53Experimental Platform Testing

To simulate the adaptability of the hydraulic system balance circuit of the herringbone gate opening and closing machine to the 0.4 m reverse water head condition, it is necessary to simulate the dynamic water resistance caused by loading the reverse water head and build an experimental platform for the balance circuit of the herringbone gate hydraulic opening and closing machine system. During testing, the external load is reduced by 1:50, which means the external load is reduced to 1/50 of its original size.

Based on the actual door opening speed, the speed gradually increases from 0mm/s to 13.5mm/s, with a hydraulic cylinder flow rate of 6.39L/min and a load cylinder flow rate of 18L/min. The effective length of the hydraulic cylinder on the experimental platform is 2m, and the action time is 8s. Observe and record the pressure at the inlet (A port), outlet (B port), and pilot port of the balance valve in the experiment. Record the hydraulic cylinder speed, rod chamber pressure, and rodless chamber pressure displayed on the Labview interface, as shown in Table 3.

Table 3. Experimental site record form of balance valve.

Experimental Conditions	0.4m Reverse Head
Balance valve inlet pressure MPa	11
Balance valve outlet pressure MPa	0
Balance valve pilot pressure MPa	5.2
Hydraulic cylinder speed mm/s	13.5

Hydraulic cylinder rodless chamber pressure MPa	11
Hydraulic cylinder has rod chamber pressure MPa	5.2

The load curve and speed curve fitted under actual testing conditions are shown in Figure 25 and Figure 26, respectively.

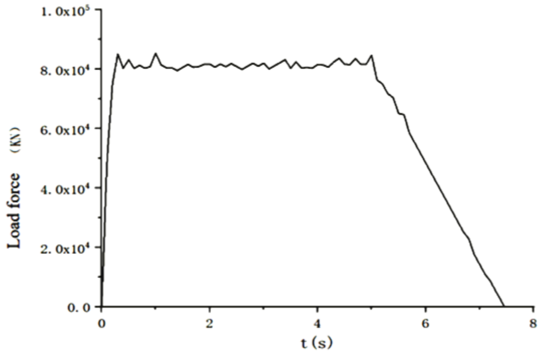


Fig. 25. Load variation curve.

By loading a 0.4m reverse head load as shown in Figure 25, maintaining it for 5 seconds and then decreasing to 0 after 3 seconds, the hydraulic cylinder speed curve is obtained as shown in Figure 26. It can be seen from the figure that the hydraulic cylinder speed increases linearly with the increase of load force during the loading period. When the load changes, the speed can basically stabilize.

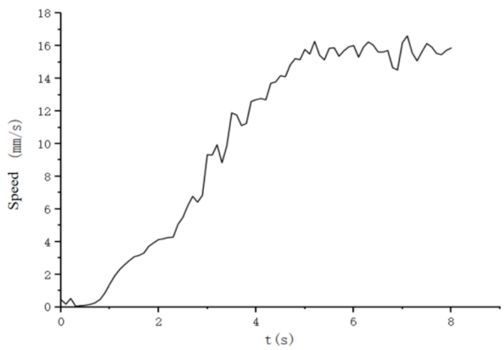


Fig. 26. Speed variation curve.

6 Conclusion

(1) A simulation study was conducted on the dynamic and static characteristics of a new type of pilot plug-in balance valve. The results showed that the pressure difference at the main valve port opened within the range of 0.1 MPa to 0.2 MPa, indicating that the one-way opening pressure of the balance valve was small, the energy consumption

was low, the throttling effect was good, and the pressure control range reached 1.55 MPa to 4.5 MPa. The pressure regulation range was large and the balance valve regulation performance was good. The new type of pilot plug-in balance valve has the characteristics of high flow rate and low pressure drop.

(2) The new type of pilot plug-in balance valve can operate stably on the 1:50 simulation model and experimental testing platform of the herringbone gate hydraulic opening and closing machine. Under different submergence depths and reverse water head conditions, the new balance valve exhibits excellent performance, and the hydraulic cylinder has a rod chamber pressure that can be maintained; During the loading period, the speed of the hydraulic cylinder tends to increase linearly with the increase of load force. When the load changes, the speed can basically remain stable and there will be no crawling or drifting of the gate. The new type of pilot plug-in balance valve has achieved insensitivity of hydraulic cylinder speed to load, basically realizing load and speed independence.

(3) During the experiment, it was found that there were certain differences in the performance of samples from the same batch due to different processing accuracies. Subsequent research can be conducted on the machining and assembly processes of the valve core, valve sleeve, and other related components of the balance valve to ensure that the machining accuracy of each valve port and the assembly clearance between the valve core and valve sleeve meet the requirements, in order to ensure that the actual performance of the balance valve is as close as possible to the design expectations.

References

1. YANG G l, MAO Y Y, LIU Y C, et al. Stability Analysis of Cartridge Balance Valve in Hydraulic System of Hydraulic Hoist of Yangtze River Shiplock Based on AMESim [J]. Chinese Hydraulics & Pneumatics, 2023,47(02): 100-108.
2. WEI W W. Design of miter gate hoist machine for TGP permanent shiplock [J]. YANGTZE RIVER, 1998, 29(12): 7-8.
3. WANG X D, CHEN K. An Elementary Approach to Drift of Miter Gate of the Three Gorges Permanent Shiplock [J]. Port &. Waterway Engineering, 2002, (04):37-40+46.
4. LI H Z. Reliability Analysis of Motor-Oil Pump Coupling of Three Gorges Ship Lock Based on LSEDA-BFO[J]. Water Resources and Power, 2022, (11):193-197.
5. CHEN B. Research on Dynamic and Static Characteristics of Pilot Large Flow Hydraulic Balance Valve [D]. Wuhan University of Science and Technology,2021.
6. CHEN H B. Research on the Dynamic Characteristics and the Optimization of Parameters for a Novel Piloted Load Control Valve [D]. Zhejiang University, 2020.
7. XIONG S J. The Application of Hydraulically-actuated Speed-limiting Counterbalance Valve in Hydraulic Gate Hoists of Three Gorges Project[J]. Machine Tool & Hydraulics, 2008,36(09):237-239.
8. YIN J, HE B, XIAN X P, et al. Dynamic Characteristic of Hydraulic Balance Valve Based on AMESim [J]. Machine Tool & Hydraulics, 2017,45(20):107-109.