

Design and Analysis of Power Head Axial Servo Loading System

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Abstract

Power head is the main motive mechanism of rotary drilling rig. High axial pressure and high frequency is one of its typical working conditions. Axial loading test is a key point in power head performance tests. Overall design, system theoretical analysis and simulation of power head axial loading system were carried out. The bond graph model and state equations of power head axial servo loading system were built on the basis of bond graph theory. Hydraulic system simulation model was constructed and axial steady-state and dynamic loading simulation was made by use of hydraulic simulation software. The maximum axial steady-state load is up to 450kN and the dynamic frequency is 20Hz, both of which can meet the technical requirements. The simulation validates the feasibility of the scheme about power head axial loading system and it provides theory and practice guidance for perfecting the loading system.

Keywords: power head; axial loading system; bond graph; hydraulic simulation

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1. Introduction

Rotary drilling rig is an ideal construction drilling machine in the foundation building engineering with large output torque and high construction efficiency. Rotary drilling rig is a kind of soil pile hole construction machinery. It can drill pipe by cutting soil to the outer ground driven by rotary bucket and cyclic operation [1]. Power head is the most important part of the rig and its structure is shown in Figure 1. Power head pressure plate directly contacts with components providing the axial pressure in the working process of rotary drilling rig. Axial steady-state load on the power head is constructed through the cylinder. Power head also bears axial impact load additionally because the feeding hole construction is in a harsh working environment. The working stability of power head of rotary drilling rig directly determines whether it can work normally. Therefore, the power head test is essential and both at home and abroad such as Germany's Bauer and lux, China's Sany and Xugong. Among them, the axial load test is the main test to simulate the actual condition of the power head. This paper presents the overall scheme of the power head axial servo loading test system including the mechanical system and hydraulic system, according to the structural characteristics and the operating characteristics of the power head. Combined with theoretical analysis, steady-state and dynamic simulation verifies the rationality of the axial loading scheme.



Figure 1. Power head structure

2. System Scheme Design

2.1. Mechanical System

According to the working characteristics of the power head itself, axial loading system also adopts hydraulic cylinder loading way, by arranging the pressurized cylinder head in the upper part of the power head. It relies on the top of the hydraulic cylinder to provide vertical pressure with maximum pressure 450kN. Parameters of hydraulic cylinder can get reference from the parameters of rotary drilling rig original hydraulic system. Pressure transfers through the drill pipe to the spline transmission on the bottom of the drill pipe. Force sensor and servo oil cylinder are settled on the bottom of drill pipe. The mechanical structure of servo loading system is shown in Figure 2. By the combination of dynamic servo axial loading oil cylinder and the servo controller, signal output waveform from the controller is sent to the servo valve and then servo cylinder so as to control the corresponding action of oil cylinder in order to apply the dynamic axial pressure up to power head pressure plate. The pressure sensor connected in series between the servo oil cylinder and the power head forms a closed control loop with PC and controller. Elastic element shown in Figure 2 can absorb the instantaneous impact force played by axial loading oil cylinder, playing a role of buffering. In addition to mechanical part in Figure 2, axial servo loading system also includes a pressurized hydraulic cylinder and a hydraulic pump station, and the power head drill pipe and the dynamic head pressure plate. The upper hydraulic cylinder system can be designed according to the original matching hydraulic cylinder on top of power head. Technical index of the whole axial loading system is shown in Table 1.

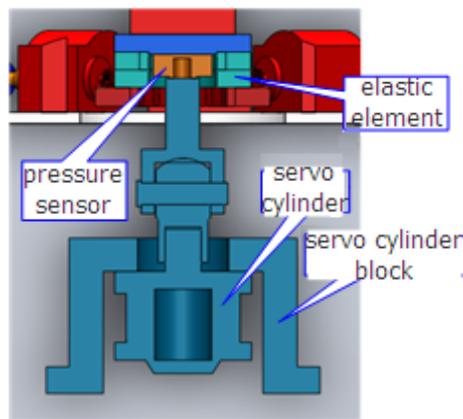


Figure 2. Mechanical system of axial loading system

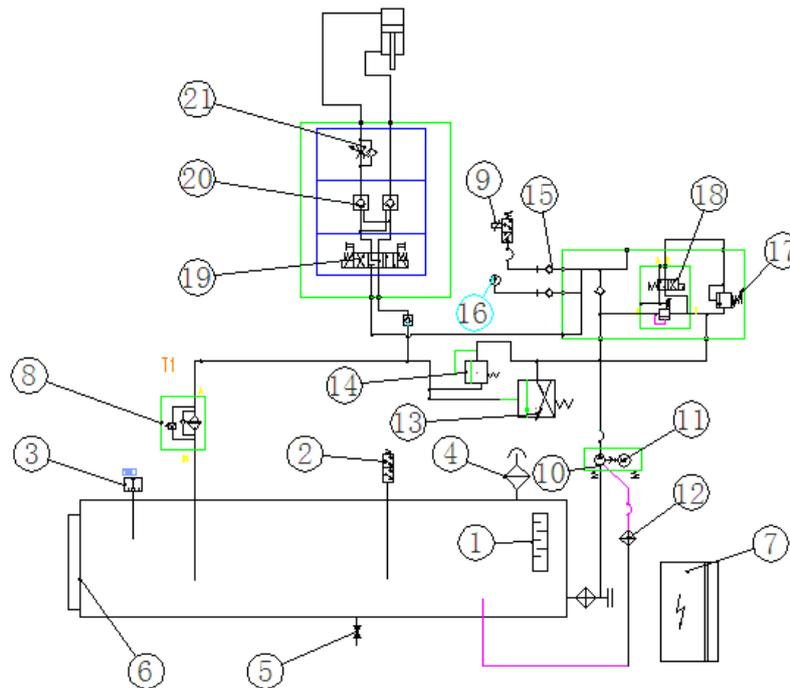
Table 1. Technical parameters of axial loading test

Serial number	Parameters	unit	value
1	The maximum axial loading pressre	kN	450
2	The axial force loading frequency	Hz	20

2.2. Hydraulic System

Steady-state hydraulic axial loading system diagram is shown in Figure 3. The oil liquid level relay 2 controls the oil capacity in the tank. Pressure transmitter 9 shows at all times and feedbacks the remote regulating pressure value 17. Superposition-type liquid control one-way valve realizes the self-locking function which can control the motion of oil cylinder piston rod. The relief valve 14 plays a role of adjusting system pressure. The unloading valve 13 works when a fault or a need for timely realization of oil return tank through the unloading valve in order to reduce the energy consumption of the system. The overflow valve 14 plays a role of security protection and adjusting system maximum working pressure. Electromagnetic reversing valve 19 has reversing, unloading and pressure maintaining function so as to control cylinder

piston rod moving direction and displacement. The electromagnetic valve 18 can adjust the pressure of the cylinder according to the loading force. The pressure sensor in Figure 2 feeds back the signal and makes a closed loop control to realize axial servo loading. Coupled with the mechanical system, the hydraulic circuit realizes the axial steady-state load and servo load of power head. The main technical parameters of components are shown in table 2. The whole hydraulic loading system is constructed on the basis of hydraulic fault diagnosis methods [2] and some hydraulic control methods [3], and the relationship among the hydraulic elements is also considered.



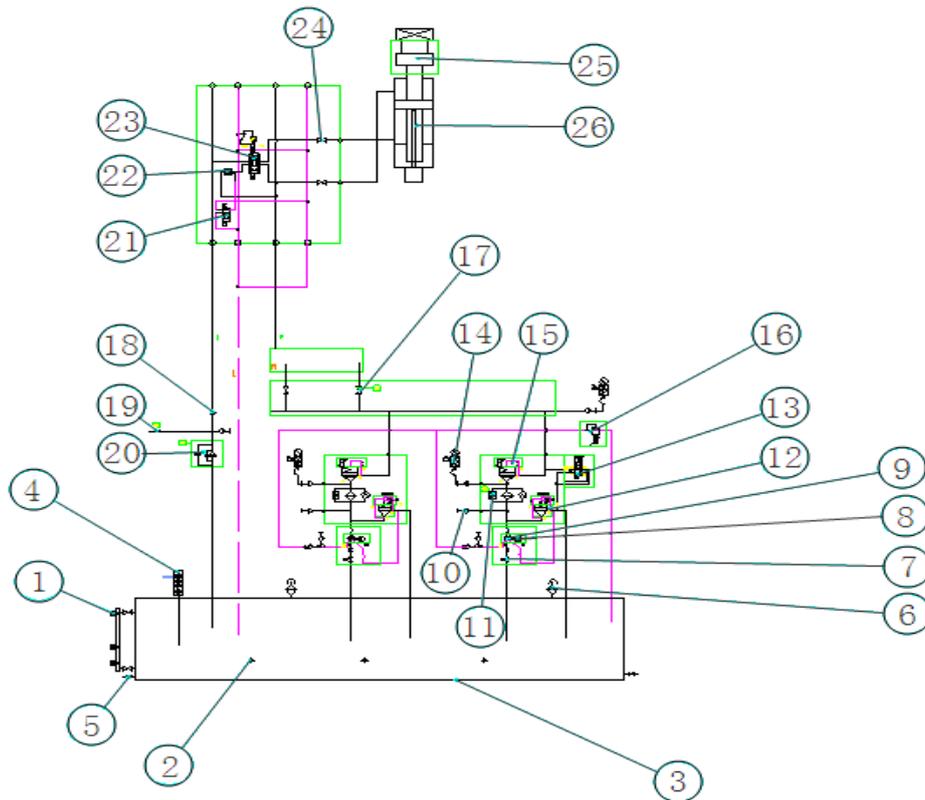
1- Level 2- level relay 3- thermal resistance 4- air filter 5- ball valve 6- tank 7- terminal box 8- oil return filter 9- pressure transmitter 10- plunger pump 11- motor 12- cooler 13- unloading valve 14- overflow valve 15- one-way pressure valve 16- pressure gauge 17- remote pressure regulating valve 18- proportional relief valve 19- reversing valve 20- superimposed type hydraulic control one-way valve 21- stacked unidirectional throttle valve

Figure 3. Hydraulic principle of axial loading

Table 2. Technical parameters of axial loading system

Serial number	Parameter	unit	value
1	the discharge of pump	ml/r	25
2	the related pressure of pump	MPa	32
3	number of pump		1
4	motor power	kW	7.5
5	motor speed	rpm	1460
6	cylinder bore	mm	200
7	piston rod diameter	mm	110
8	thrust	kN	450
9	system pressure	MPa	32

As shown in Figure 2, the servo cylinder fitted on the power head rod lower applies axial dynamic load to the drill pipe. According to the test requirements, the maximum dynamic loading force should reach 450kN and the loading frequency should be up to 25Hz. The force sensor can measure the loading force and build a closed-loop control system to feedback the force signal to the control system. The principle diagram of hydraulic servo loading system is shown in Figure 4.



1- level relay 2- heater 3- oil tank body 4- temperature measuring instrument 5- ball valve 6- air cleaner 7- butterfly valve 8- motor 9- plunger pump 10- pressure adapter 11- high pressure filter 12- electromagnetic relief valve 13- pressure relief valve 14- pressure transmitter 15- plug-in unit 16- proportional relief valve 17- high-pressure ball valve 18- check valve 19- ball valve 20- oil return filter 21- electromagnetic valve 22- liquid control one-way valve 23- servo valve 24- plate valve 25- force sensor 26- servo cylinder

Figure 4. Hydraulic principle of servo loading system

3. Theoretical Analysis

3.1. Mathematical Modeling

Figure 5 shows the simplified hydraulic principle diagram of the axial pressure loading system. Driving system consists of motor and hydraulic pump. The output hydraulic oil is divided into two ways by the three position four way reversing valve. One hydraulic circuit is constructed with the loading cylinder rodless cavity and another circuit is that the hydraulic oil goes back to tank through the cylinder rod chamber and reversing valve. Relief valve plays a role of setting pressure parameters of pressure regulation system.

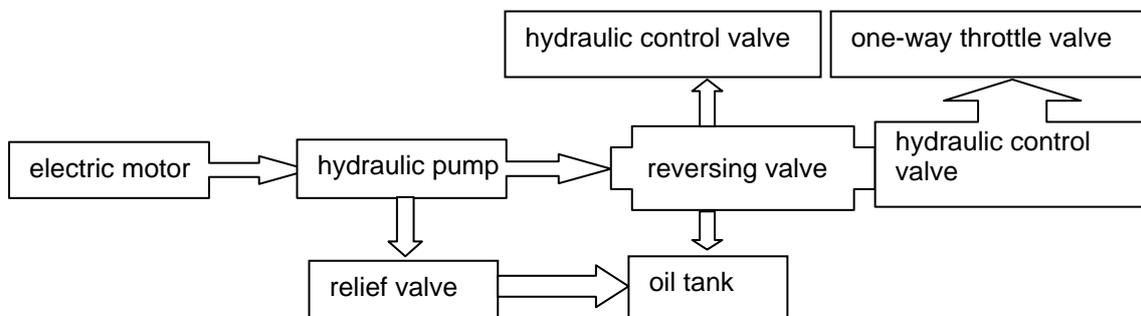


Figure 5. Simplified principle diagram of the axial loading system

Reversing valve can regulates the output flow direction and opening size between the hydraulic pump and oil cylinder, so as to realize the up and down reciprocating movement of the hydraulic cylinder rod. In the back and forth movement process of the cylinder rod, in order to avoid the impact of the piston caused by the weight of cylinder during downward movement, the combination of two liquid control one-way valves and the one-way throttle valve controls the flow and pressure of the cylinder rodless cavity and then the piston rod doesn't appear unstable impact in the falling process. This design improves the stability of the loading system. According to bond graph theory [4-6] and the comprehensive factors affecting the hydraulic system, bond graph model of the simplified axial loading system (which is shown in Figure 5) is built which is shown in Figure 6.

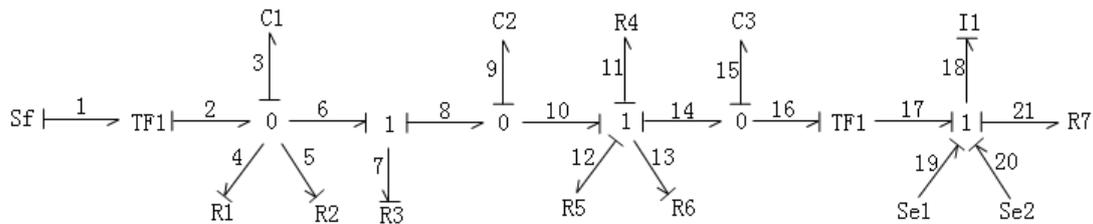


Figure 6. Bond graph model of the axial loading system

The capacitive elements:

where C1 is the liquid capacity [7-8] of the hydraulic pump. C2 is the liquid capacity of the long hose. C3 is the liquid capacity the rodless cavity.

The resistive elements:

where R1 is the liquid resistance [9] of the hydraulic pump leakage. R2 is the liquid resistance of the relief valve. R3 is the liquid resistance of the long hose. R4 is the liquid resistance of the reversing valve. R5 is the liquid resistance of the liquid control one-way valve. R6 is the liquid resistance of the throttle valve. R7 is the liquid resistance of the piston internal leakage.

The inertial elements:

where I1 is the equivalent inertia mass of cylinder piston and liquid.

The source elements:

where Se1 is the cylinder piston gravity. Se2 is the external load force.

3.2. Establishment of State Equation

On the basis of variable provisions concerning about the system bond graph, the flow variable integral of the capacitive element C and the potential variable integral of the inductive element I are used as the state variables of the system. In this system, liquid flow volume variables and pressure momentum variables are the state variables. System input variables as energy input factors are used to indicate the potential source map in the bond graph. System space state variables X and input variables U and total energy variables [10-11] as shown below:

$$X = \begin{bmatrix} q_p \\ q_p \\ q_{1p} \\ p_{1p} \end{bmatrix} \quad U = \begin{bmatrix} Sf \\ Se1 \\ Se2 \end{bmatrix} \tag{1a, 1b}$$

The total energy variables as follows:

$$e_p = \frac{q_p}{C_1} \tag{2}$$

$$e_9 = \frac{q_9}{C_2} \quad (3)$$

$$e_{15} = \frac{q_{15}}{C_3} \quad (4)$$

$$f_{10} = \frac{p_{10}}{I_1} \quad (5)$$

where e_3 , e_9 and e_{15} are on behalf of pressure variables of the capacitive components in the hydraulic system. f_{10} is the speed variable of the inductive components in hydraulic system.

The space state equations derived from the bond graph and the relationship of the parameters as follows:

$$\dot{q}_3 = \frac{1}{C_2 R_3} q_9 - \frac{R_2 R_3 + R_1 R_2 + R_1 R_3}{C_1 R_1 R_2 R_3} q_3 - \frac{1}{k} S f \quad (6)$$

$$\dot{q}_9 = \frac{1}{C_1 R_3} q_3 - \frac{1}{C_2 [(R)_4 + R_5 + R_6]} q_{15} - \frac{R_4 + R_5 + R_6 - R_3}{C_2 [(R)_4 + R_5 + R_6]} q_9 - \frac{2}{k} S f \quad (7)$$

$$\dot{q}_{15} = \frac{1}{C_2 [(R)_4 + R_5 + R_6]} q_9 - \frac{1}{C_2 [(R)_4 + R_5 + R_6]} q_{15} - \frac{A_1}{A_2 I_1} p_{10} \quad (8)$$

$$\dot{p}_{10} = \frac{A_1}{A_2 C_3} q_{15} - \frac{R_7}{I_1} p_{10} - S e_1 - S e_2 \quad (9)$$

where p is on behalf of the momentum variables of the inductive components I . p_{10} is the mechanical translation momentum in cylinder movement system, and its derivative is mechanical displacement force corresponding to the system. Q is on behalf of the change position variables of the capacitive components. q_3 , q_9 and q_{15} are on behalf of the oil volume corresponding to the components of the hydraulic system, and their first derivatives indicate the corresponding flow. Sf is on behalf of the current source variables of the system and it says the input velocity of the hydraulic pump in detail. Se is on behalf of the potential source variables of the system. Se_1 is the oil cylinder gravity and Se_2 is the load reaction.

4. System Simulation

4.1. Simulation Model

AMESim [12-13] (English abbreviation: Advanced Modeling Environment for performing Simulation of Engineering Systems) is one kind software used for hydraulic / mechanical system modeling, simulation and dynamic analysis which is launched by the France Imagine company based on bond graph in 1995. AMESim provides a time domain simulation modeling environment for the user. It can use the existing models to build new model components to construct the optimization of practical prototype design for easy identification, using standard ISO icon and simple multiport diagram. It provides specific application examples which are convenient for users to build complex systems. It can modify the model and simulation parameters for steady-state and dynamic simulation curve and simulation results, building a convenient interface for users. With the help of software AMESim, simulation model can be constructed later.

The vertical force from the dynamic head pressure plate is applied on rotary drilling rig to drill down into the soil, and the vertical pressure of the power head is from the hydraulic cylinder on the mast of the rotary drilling rig. The test bench of power head is designed to simulate the dynamic load in a real work process by use of hydraulic system simulation software AMESim and the bond graph and the axial load simulation model is constructed which is shown in Figure 7. Steady-state and dynamic simulation of the axial loading system can be achieved and exchanged by adjusting the input signal. The axial load simulation model based on the

following principles: it focuses on simplifying the main hydraulic components and the analysis under ideal condition. Factors such as cooler etc which have almost no effect on the system don't be considered. The pressure plate of power head is simplified as a spring as an equivalent contact between two rigid bodies and the spring has its own rigidity.

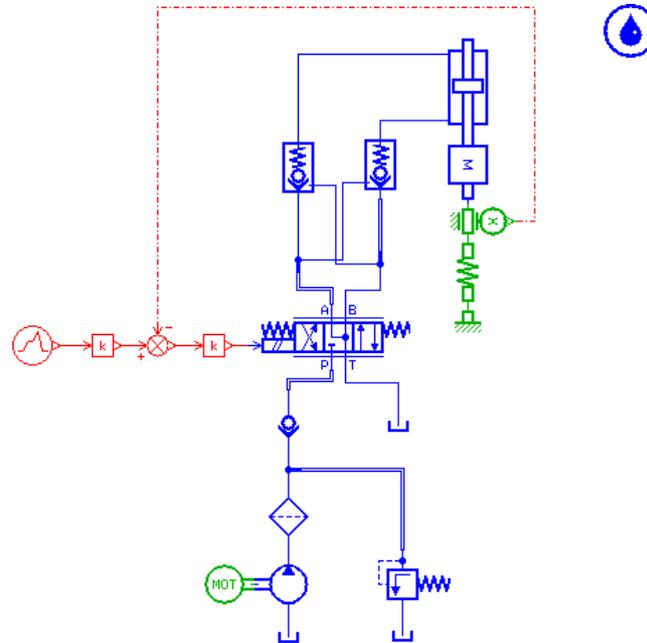


Figure 7. Simulation model of axial loading system

4.2. Results Analysis

Figure 8 shows the steady-state output pressure of axial hydraulic cylinder which says that the maximum steady-state axial loading pressure is 450kN and the corresponding time is very short. This curve indicates the axial loading system can quickly reach to a steady state. Figure 9 shows the changes curve of import and export flow of hydraulic cylinder and import.

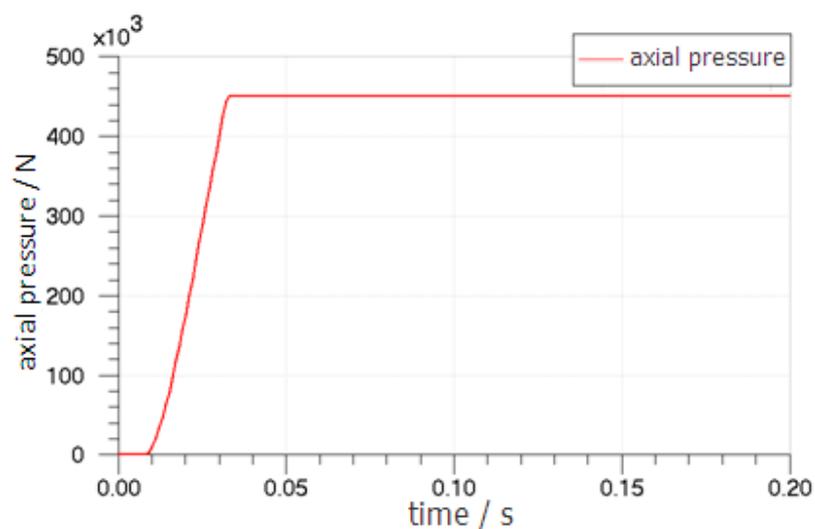


Figure 8. Axial static pressure

Export flow change trend has an agreement because it is a symmetric cavity with a rod. At initial time of reversing valve action, flow has a mutation, and then returns to a steady state. The change trends of Import and export flow of oil cylinder and axial pressure are consistent. When the cylinder moves to the lowest position and power head is applied with maximum axial load, oil cylinder is in the state of pressure keeping. Due to the leakage of cylinder, the import and export volume is not zero. Figure 10 and Figure 11 show the dynamic axial load simulation curves. In Figure 10, it is the dynamic axial pressure curve on the state of sine signal input state under the frequency of 20Hz and the maximum pressure of 450kN, which meets the technical requirements in table 1. Figure 11 shows the flow changes of port A and B of the reversing servo valve and it can be seen that the flow trend is consistent with Figure 10 according to servo input signal changes. From Figure 10 and Figure 11, it also can be seen that both peaks of the curves slightly increase in the initial stage and the dynamic changes of flow as well as output axial pressure are synchronous, which is in line with the actual situation.

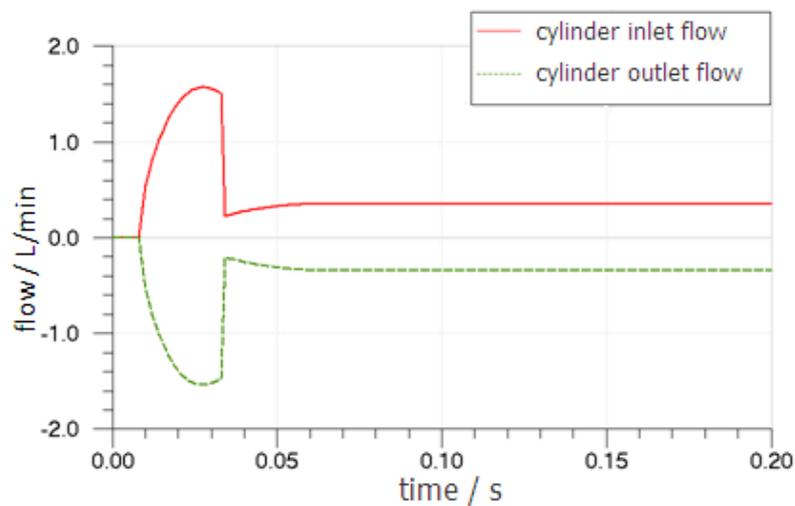


Figure 9. Steady-state flow of hydraulic cylinder

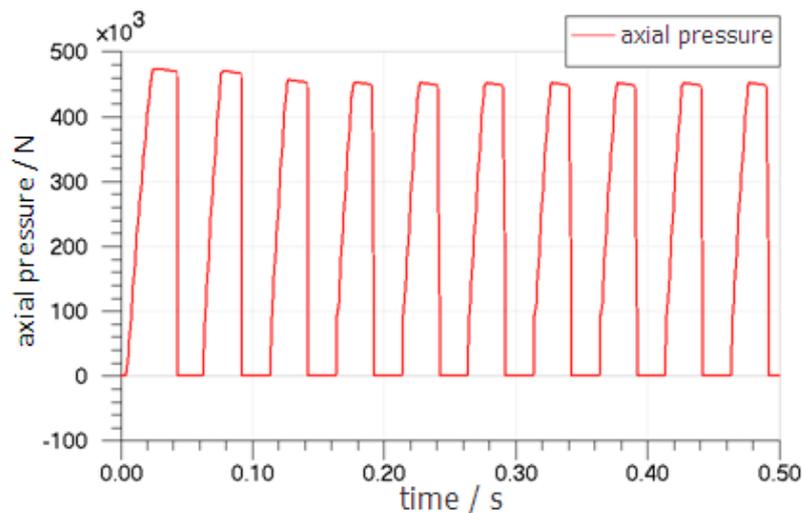


Figure 10. Axial dynamic pressure

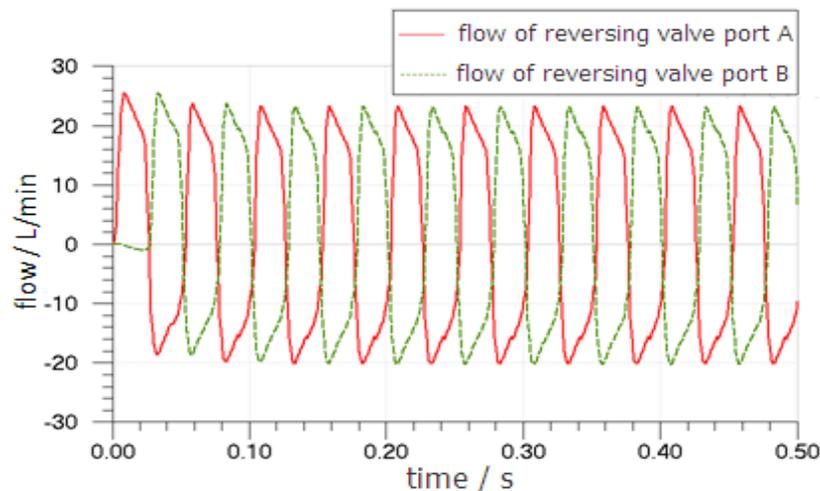


Figure 11. Flow of reversing valve port A and B

5. Conclusion

Axial loading test system of power head is constructed by use of hydraulic cylinder and some other hydraulic components. Based on bond graph theory, axial loading system is analyzed in theory. With the help of simulation software AMESim, the steady loading and servo dynamic loading tests are simulated and analyzed. The results show that the maximum axial load can reach to 450kN and servo loading frequency can reach to 20Hz. In addition, the corresponding speed is fast and the change process is stable. It verifies the feasibility of the mechanical system and hydraulic system principle, providing theoretical guidance for perfecting the axial loading system. The present study of rotary drilling rig power head is not much, most of which is just simulation analysis and simple theoretical calculation about the mechanical structure strength of power head. Combination of further theoretical analysis and mechanical and hydraulic hybrid simulation is better than that of only with mathematical calculations because it can more directly explain the principle, methods and results. There is a more real reflection of power head's working condition and it can bring improvements for the structural design of power head. Axial steady loading and servo loading system design scheme of power head is reasonable and it can be used in the future tests.

Acknowledgements

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