Optimization of Low Frequency Oscillation of Variable Piston Chamber in High-Speed Aviation Pump

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Abstract. High-speed aviation pump often used as a constant pressure source of hydraulic servo system, which is essential to the maneuverability and stability of the whole system, and requires its regulating mechanism have good stability and quick response. In actual operation process, the variable piston chamber of the high-speed pump often appears low frequency oscillation phenomenon, which causes the swash plate to produce corresponding low frequency vibration, and then causes the pressure fluctuation of the outlet, affecting the stability of the control system. In this paper, the influence factors of constant pressure variable control stability of high-speed pump are analyzed by means of theoretical analysis, AMESim simulation analysis and actual test. The results show that the stability margin of the system can be precisely controlled by changing the piston clearance and then the total leakage coefficient of the pump, and the stability of the control system can be effectively improved.

Keywords: Fluid transmission and control · Control stability · Pressure pulsation · AMEsim · High-speed aero pump

1 Introduction

Aerospace equipment represents the highest level of industrial development of a country, plays a vital role in maintaining national security and promoting the progress of scientific and technological. High-speed steering gear system is the core drive system of the missile, rocket, and aircraft attitude control to aerospace equipment. As the constant voltage source of the steering gear hydraulic servo system, high speed piston pump is the key of control performance of servo system, regulating its mechanism has a good stability and fast response. But in the process of actual operation, piston pump often appears outlet pressure of the low frequency oscillation phenomenon at constant pressure stage. In order to improve the control stability of piston pump, many scholars have carried out relevant research.

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Many scholars focus on stability judgment and control parameter optimization of piston pump control system. Gwak, et al. [1] analysed swashplate structure stability of the ship hydraulic piston pump hydraulic system by finite element method. Kim, et al. [2] established load sensitive nonlinear mathematical model of hydraulic system considering the dynamic characteristics, and the stability analysis was carried out by using Routh-Hurwitz stability criterion based on linearization of the model. Kim, et al. [3] analysed the influence of parameters sensitivity analysis system on the parameters of the piston pump control system. And the influence degree of each parameter and the rationality of reducing the order were discussed in detail based on simulation results. Zeiger, et al. [4] studied the optimal control of the axial piston pump, established a mathematical model of the axial piston pump, and proposed a method to calculate the average torque of the pump under a given geometry and any given operating conditions.

There are also some scholars mainly concentrated in other three aspects, structural optimization, influence factors analysis and fault diagnosis research based on the pulsating characteristics. Yang, et al. [5] find that increasing damping hole in variable displacement mechanism can effectively improve the performance of variable displacement control scheme, reduce the swash plate angle oscillation frequency, to reduce the system pulse impact. Many scholars [6–10] explored the influence of various parameters in the control system on the pressure pulsation characteristics of the piston pump by AMESim simulation and experiment, the results show that the spring stiffness of the pressure cut-off valve, the spring preload force, motor speed, leakage, and control oil pressure, both them have an impact on the pressure pulsation characteristics. Zhao, et al. [11] analyzed the mapping relationship between flow pulsation and pressure pulsation, and provided new theoretical basis and method support for fault diagnosis and health assessment of hydraulic pump motor and key components by analyzing the pulsation variation characteristics under various working conditions.

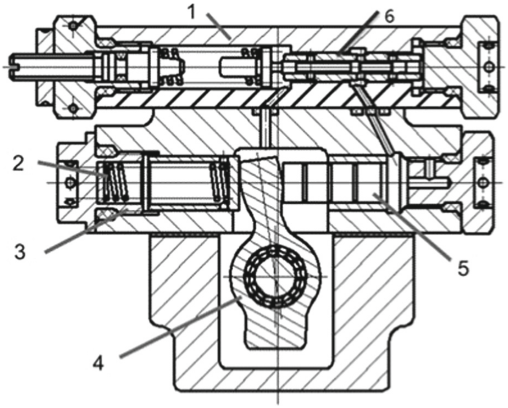
At present, there are many researches on the control stability of piston pump, and these researches are mature, but most of the research objects are ordinary industrial pumps, few researches are relatively on high-speed aviation pumps. This paper intends to adopt the way of combining the theoretically analysis, AMESim simulation analysis and actual test, analyze the influencing factors of constant pressure variable control stability of high-speed aerial pump, and study the stability optimization method and experimental verification.

2 Analysis of Constant Pressure Variable Control Characteristics

2.1 Principle of Constant Pressure Variable Control

The constant pressure variable control components of high-speed pump including constant pressure valve and swash plate components. First computing the differential value of the pump outlet pressure and the pressure setting value of variable mechanism, then adjusting the output flow based on this differential value, and keep the outlet pressure of pump for a constant value.

As shown in Fig. 1, the structure of pump's variable mechanism mainly includes constant pressure valve, variable piston and return piston. And the constant pressure variable control principle of hydraulic pump is shown in Fig. 2. Initially, the pressure



1- High pressure spring 2- Swash plate return spring 3- Return piston 4- Swash plate 5- Variable piston 6- Constant pressure control valve

Fig. 1. Structure of variable mechanism of hydraulic pump

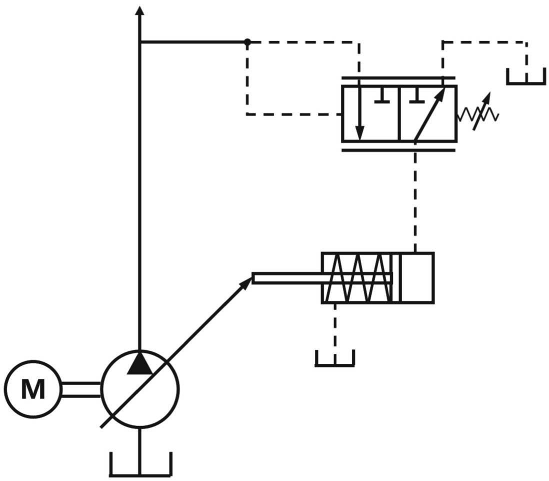


Fig. 2. Constant pressure variable control principle of hydraulic pump

control valve operates in the left position, the hydraulic pump returns to its maximum displacement under the action of a return piston spring, at this point, oil of the variable piston chamber flow into the pump housing cavity through the control valve. As the load pressure rises to zero flow pressure, control valve spool opens, the pressure control valve works in the right position. Oil of the outlet pressure enters the variable piston chamber and let the variable piston pushes the swash plate, make the pump work at zero displacement (small displacement), thus the constant pressure control of hydraulic pump is realized.

2.2 Analysis of Influencing Factors of Constant Pressure Variable Control Stability

The constant pressure variable control system of the high-speed pump mainly includes four parts: the constant pressure valve, subassembly of the swash plate, the flow and pressure of the outlet, respectively. Firstly, the transfer function of each part in the control system of the high-speed pump is analyzed.

The balance equation of the spool movement of the constant pressure valve is

$$F_V p_b = m_V \frac{d^2 x_V}{dt^2} + B_V \frac{dx_V}{dt} + K_V (x_0 + x_V) \tag{1}$$

where F_V is the area of the valve element; P_b is the setting value of the pressure; x_V is the displacement of the spring; m_V is the sum of valve element mass and 1/3 spring mass; B_V is the damping coefficient of spool motion; K_V is the spring stiffness; x_0 is the amount of spring precompression.

The transfer function of the control valve can be obtained after Laplace transform as

$$W_1(S) = \frac{x_V(S)}{F_V p_s(S) - K_V x_0(S)} = \frac{1/K_V}{\frac{S^2}{\omega_V^2} + \frac{2\xi_V}{\omega_V} S + 1} \tag{2}$$

where $\omega_V = \sqrt{\frac{K_V}{m_V}}$ is the natural frequency of the control valve; $\xi_V = \frac{B_V}{2\sqrt{K_V m_V}}$ is the damping coefficient of the valve element.

By combining constant pressure valve port flow Eq. (1), variable piston chamber flow Eq. (2) and swash plate force balance Eq. (3), the motion equations of swash plate subassembly can be obtained as follows

$$\begin{cases} (1) Q_V = C_V W x_V \sqrt{\frac{2}{\rho}(p_b - p_1)} \\ (2) Q_V = F_t \frac{dx_t}{dt} + \frac{V_t}{E_y} \frac{dp_1}{dt} + C_t p_1 \\ (3) F_t p_1 = m_t \frac{d^2 x_t}{dt^2} \end{cases} \tag{3}$$

where Q_V is the flow of the constant pressure valve; C_V is the leakage coefficient of control valve; ρ is the oil density; p_1 is the left end pressure of the variable piston chamber; F_t is the variable piston end area; x_t is the variable piston displacement; V_t is the volume sum of the cavity at pressure p_1; C_t is the leakage coefficient of variable piston; W is the width of constant pressure valve hole; E_y is the bulk elastic modulus of oil; m_t is the mass of swash plate subassembly.

After solving the equations and applying Laplace transform, the transfer function of the swash plate subassembly can be obtained as

$$W_2(S) = \frac{x_t(S)}{x_V(S)} = \frac{K_q/F_t}{S\left(\frac{S^2}{\omega_h^2} + \frac{2\xi_h}{\omega_h} S + 1\right)} \tag{4}$$

where $\omega_h = \sqrt{\frac{E_y F_t^3}{V_t m_t}}$ is the natural frequency of swash plate subassembly;

$\xi_V = \frac{(K_e+C_t)}{2F_t}\sqrt{\frac{E_y m_t}{V_t}}$ is the relative damping coefficient of swash plate subassembly; $K_q = C_V W\sqrt{\frac{2}{\rho}(p_b - p_1)}$ is the flow gain of control valve.

The output theoretical flow rate of pump is

$$Q_{lb} = \frac{\pi}{4} d_Z^2 D_t Z \tan \gamma n_b = -K_Q n_b x_t \tag{5}$$

where d_Z is the piston outer diameter; D_t is the diameter of piston distribution circle; Z is the number of piston; γ is the inclination angle of the swash plate; K_Q is the pump displacement coefficient; n_b is the pump speed;

The transfer function of pump output flow after Laplace transform can be obtained as

$$W_3(S) = \frac{-x_t(S)}{Q_{lb}(S)} = K_Q n_b \tag{6}$$

The flow continuity equation of pump output pressure is

$$Q_{lb} - Q_f = C_s p_p + \frac{V_s}{E_y}\frac{dp_b}{dt} \tag{7}$$

where Q_f is the load flow required by the servo system; V_s is the sum of the volume of the piston pressure chamber and the drainage pipe; C_s is the total leakage coefficient of pump;

The transfer function of pump output pressure by Laplace transform can be obtained as

$$W_4(S) = \frac{p_b(S)}{Q_{lb}(S) - Q_t(S)} = \frac{1/C_s}{1 + \frac{S}{\omega_s}} \tag{8}$$

where $\omega_s = \frac{E_y C_s}{V_s}$ is the volume lag turning frequency.

According to Eqs. (2), (4), (6) and (8), the open-loop transfer function of the system can be obtained as

$$W(S) = \frac{\frac{K_q K_Q n_b F_V}{K_V F_t C_S}}{S\left(1 + \frac{S}{\omega_s}\right)\left(\frac{S^2}{\omega_V^2} + \frac{2\xi_V}{\omega_V}S + 1\right)\left(\frac{S^2}{\omega_h^2} + \frac{2\xi_h}{\omega_h}S + 1\right)} \tag{9}$$

In a control system, it is required that the system can not only work stably, but also need to have enough stability margin, considering the influence of internal parameters and external environment changes on system stability. Stability margin is usually expressed by phase margin and gain margin. In control engineering, it is generally required that the phase margin of the system is within the range of 30°–60°, gain margin ⩾6 dB. When the system is intermittently unstable, it is generally considered that the stability margin of the system is insufficient. To obtain sufficient stability margin, which is reflected in the system transfer function, it is to reduce the open-loop gain K of the system.

According to Eq. (9), the open-loop gain K of the system is

$$\boldsymbol{K} = \frac{\boldsymbol{K_Q K_q n_b F_V}}{\boldsymbol{K_V C_S F_t}} \tag{10}$$

The stability margin of the system can be increased by reducing the flow gain of the control valve $\boldsymbol{K_q}$, the area of the valve core $\boldsymbol{F_V}$, or by increasing the spring stiffness of the control valve $\boldsymbol{K_v}$, the total leakage coefficient of the pump $\boldsymbol{C_S}$, and the area of the variable piston end $\boldsymbol{F_t}$, except that the pump displacement coefficient $\boldsymbol{K_Q}$ and the speed $\boldsymbol{n_b}$ are fixed under the fixed working condition. Among the parameters that can affect the stability margin of the control system, the total leakage coefficient of the pump $\boldsymbol{C_S}$ is relatively easier to accurately control through a variety of ways. Therefore, this paper mainly controls the stability margin of the system by changing the total leakage coefficient of the pump $\boldsymbol{C_S}$, to improve the stability of the control system.

3 Simulation Analysis

Through the analysis of the influencing factors of constant pressure variable control stability in Sect. 2.2, it is determined that the stability margin of the system can be controlled by changing the total leakage coefficient of the pump $\boldsymbol{C_S}$, to improve the stability of the control system. This paper intends to changing the slot clearance of the variable piston to increasing its leakage amount, and then increase the total leakage coefficient of the pump $\boldsymbol{C_S}$. In order to verify the optimization performance of control stability, simulation analysis is carried out in AMESim firstly in this section.

3.1 Model Constructing

According to the constant pressure control mechanism and principle of high-speed aviation pump, the AMESim simulation model is built as shown in Fig. 3. In order to make the simulation model more accurate simulation of hydraulic pump control mechanism of the actual movement and force, constant pressure valve is built with the HCD hydraulic components library, and the swash plate, variable piston and the return piston are using PLM plane institutional repository. The main parameters of the model are set by referring to the design drawings and size measurement results of the relevant parts of the high-speed pump, which ensures the consistency of the simulation model and the research object.

3.2 Model Verification

In order to verify the accuracy of the established model, the parameters of the simulation model were set according to a faulty pump with low-frequency oscillation, and then the simulation curve of the faulty pump was compared with the actual test results. According to the above simulation model, the pressure change curve of the variable piston chamber of the pump was obtained, as shown in Fig. 4.

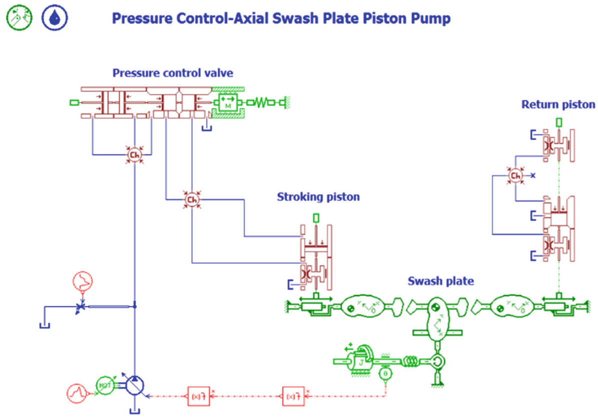
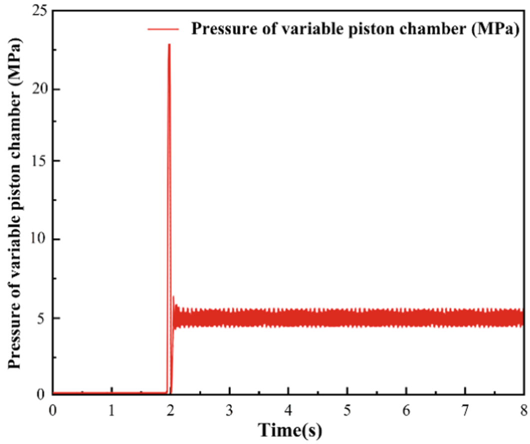
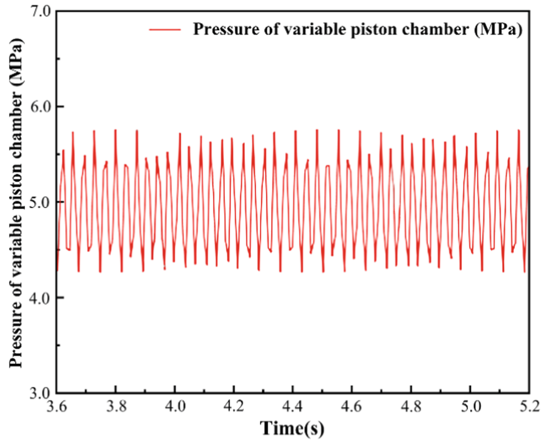


Fig. 3. Simulation model of pump in AMEsim



(a) Pressure curve from 0s to 8s



(b) Enlarged pressure curve from 3.6s to 5.2s

Fig. 4. Simulation curve of pressure in variable piston chamber

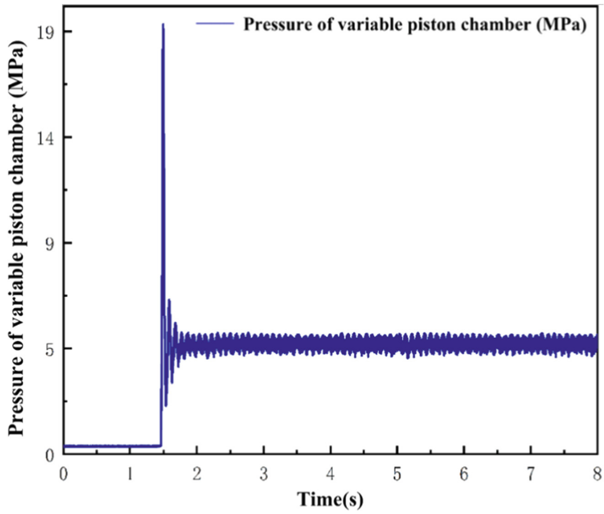


Fig. 5. Measured curve of pressure in variable piston chamber of fault pump

Figure 4(a) shows that the pressure curve of variable piston chamber obtained by simulation model appears low frequency oscillation. From the enlarged pressure curve in Fig. 4(b), it can be seen that the oscillation frequency is 28 Hz, oscillation amplitude is 1.5 MPa.

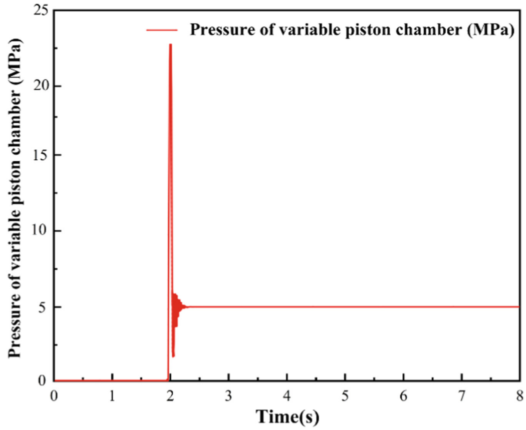
Figure 5 for the pressure measured curve of faulty pump variable piston chamber. Combined with Fig. 4. And Fig. 5., it's easy to get the conclusion that the amplitude and frequency of the simulated curve are basically consistent with that of the pressure curve collected by the actual fault pump, therefore, the simulation model can reflect the actual situation of the measured pump accurately.

3.3 Analysis of Influence Degree of Different Clearance

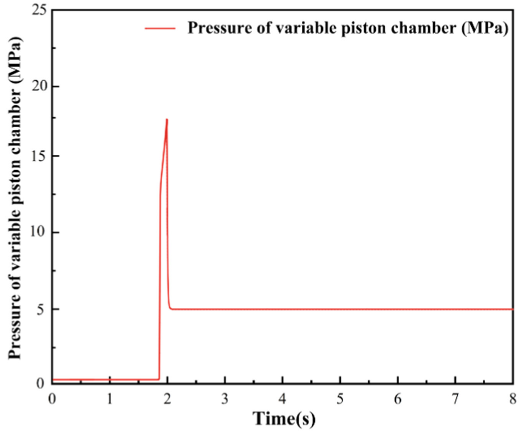
The gap of variable piston was changed to analyze the influence of different gap of variables piston on low-frequency oscillation of variable piston. The gap of variable piston of the faulty pump was 0.015 mm above.

First, the clearance of variable piston was set as 0.025 mm, an increase of 0.01 mm compared with that of the faulty pump. The simulation results are shown in Fig. 6(a), It can be seen that when the gap increases, the low-frequency oscillation of the variable piston chamber pressure disappears, indicating that increasing the variable piston clearance is conducive to eliminating the low-frequency oscillation of the variable piston chamber.

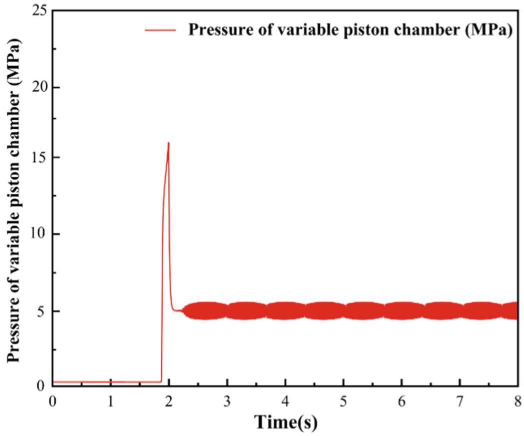
Continue increasing the variable piston gap to 0.07 mm and the pressure curve of the variable piston chamber is shown in Fig. 6(b), the pressure oscillation phenomenon of the variable piston chamber is still disappearing.



(a) piston gap 0.025mm



(b) piston gap 0.07mm



(c) piston gap 0.08mm

Fig. 6. Simulation curve of pressure in variable piston chamber of different piston gap

But as the variable piston gap increasing to 0.08 mm, the pressure oscillation phenomenon of the variable piston chamber is repeated as shown in Fig. 6(c), and there is a certain beating vibration phenomenon. Explain that the variable piston gap is not the bigger the better, but exist an optimal interval. For find the optimal interval, the simulation parameters have been adjusted for many times according to the above simulation model, and the optimal gap range of the variable piston is obtained from 0.025–0.07 mm.

4 Experimental Verification

The simulation results in Sect. 3 show that increasing variable piston gap in optimal interval is beneficial to eliminate low-frequency oscillation. In the original design, the gap range of the variable piston was required to be 0.013–0.031 mm, thus the variable piston of the failed pump was reworked, that is, the gap of 0.015 mm was increased to 0.025 mm, and the experimental verification was carried out. The optimized pressure curve of the variable piston chamber was shown in Fig. 7.

Figure 7 shows that when the gap of the variable piston of the faulty pump is increased to 0.025 mm, the low-frequency oscillation phenomenon of the variable piston chamber pressure disappears. After the piston chamber pressure is optimized, the pump outlet pressure is continued to be tested, and the measured curve is shown in Fig. 8.

Figure 8 shows that when the gap increases to 0.025 mm, the low-frequency oscillation phenomenon at the pump outlet also disappears. And when the gap continues to increase to 0.08 mm, the beating vibration phenomenon appears again, which is consistent with the above simulation results.

The result of experimental verification shows that through exploring the influence factors of constant pressure variable control stability of the high-speed pump, and by AMESim simulation calculation and analysis, the improvement measures proposed can effectively improve the low-frequency oscillation of variable piston chamber, then improve the outlet pressure of the aviation pump.

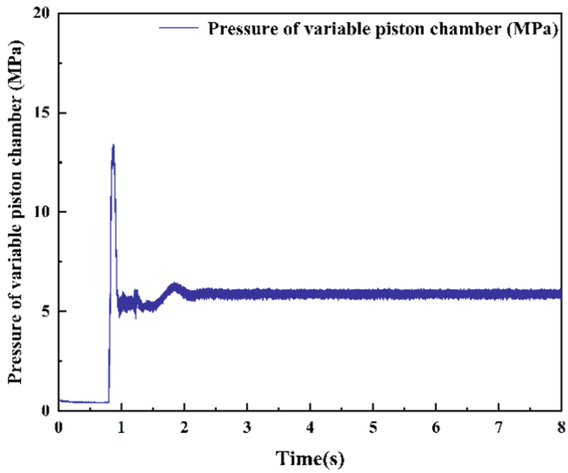


Fig. 7. Pressure curve of optimized variable piston chamber

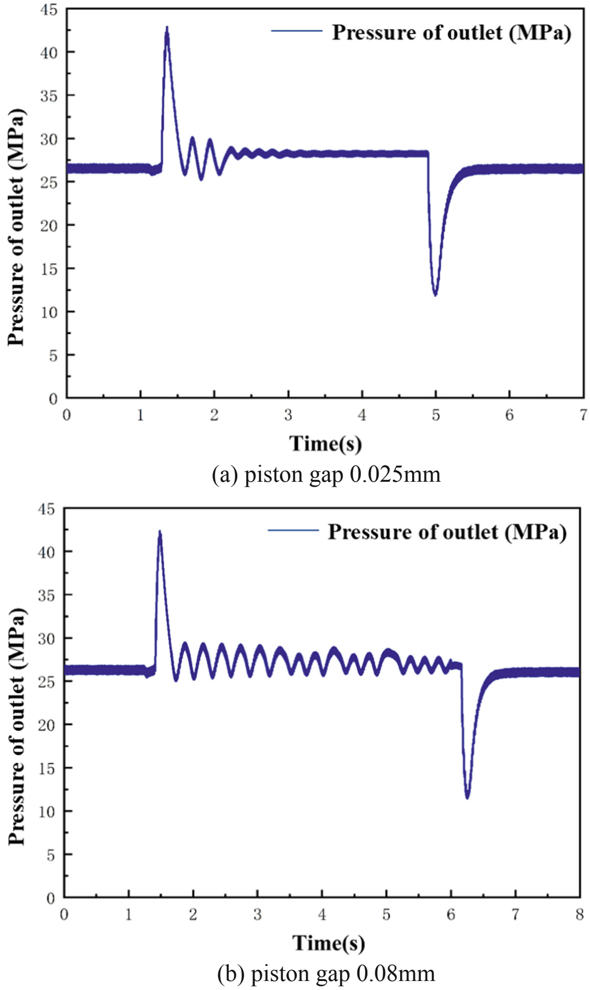


Fig. 8. Pressure curve of pump outlet

5 Conclusion

(1) By analyzing the influencing factors of constant pressure variable control stability of high speed pump, it is found that the stability margin of the system can be precisely controlled by changing the piston gap of variable and then changing the total leakage coefficient of the pump, thus effectively improving the stability of the control system.

(2) Although increasing the variable piston clearance can improve the stability of the system, the variable piston gap is not the bigger the better, but exist an optimal interval. In the optimal range (0.025 mm–0.07 mm), the control system of the pump tends to be stable, the variable piston chamber does not appear low frequency oscillation. But when the variable piston gap exceeds 0.08 mm, the variable piston

cavity of the hydraulic pump will appear low frequency oscillation phenomenon again.

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