

# Zero Point Compensation Control of the Proportional Valve with Negative Overlap

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<https://doi.org/10.5755/j02.mech.29715>

## 1. Introduction

In order to improve the accuracy of load flow compensation control in the electro-hydraulic servo system, this paper analyses the laminar flow and turbulent flow state of the servo valve, and studies the load flow equation in the laminar flow state with small opening or small pressure difference of the valve orifice. Based on the expected linear flow equation and the laminar flow equation, a flow compensator is designed to compensate the flow affected by pressure difference, which makes the flow characteristics linear. It can be used as an open-loop controller to resist the nonlinearity of valve and form a composite control strategy with closed-loop control strategies to improve the accuracy of the system.

The electro-hydraulic system is widely used in industry, military, ocean and other fields because it has many advantages, such as compact structure, small workspace, large driving capacity, fast response speed and good positioning function. The system is affected by friction, the overlap, oil-hydraulic contraction, internal leakage and other factors. In [1, 2], a nonlinear sliding mode control based on an extended disturbance observer is proposed to track the desired position trajectory for electro-hydraulic single-rod actuators. In [3, 4], the model of the electro-hydraulic servo system was linearized, and the resulting linear model is used to synthesize an output-feedback  $H_\infty$  controller to improve the performance of the system. In [5], a double fuzzy control method with compensating factor was proposed to solve the asymmetry problem caused by the asymmetric cylinder, and the fuzzy PID controller is used to adapt the mutative load.

The control method studied above is mainly aimed at the servo valve control hydraulic cylinder system. These nonlinear control methods have complex algorithms and are difficult to be applied in industrial applications. In order to reduce costs and maintenance difficulties, more proportional valves are used in the industry. The manufacturing accuracy of the proportional valve is relatively low, especially in the middle of the valve, which has a certain overlap between the valve core and the valve sleeve. The overlap will cause the zero point of the system to not be at the zero position of the proportional valve, thereby affect the control accuracy of the system. According to the size of the valve core shoulder and the groove width of the valve sleeve, the

overlap of the valve can be divided into a positive overlap valve and a negative overlap valve.

For the positive overlap valve, the proportional valve has a certain flow dead zone. In [6], it proposed that asymmetric cylinder system controlled by the positive overlap valve is prone to overpressure, and appropriately reducing the overlap of the proportional valve can eliminate the overpressure phenomenon and improve the control performance of the system. Deng W. X. [7] established a valve-controlled symmetrical cylinder system flow model with a dead zone, which has a certain reference for studying the establishment of a valve-controlled cylinder system flow model with overlap. Peng X. W. et al. [8] proposed a fuzzy dead-band compensation algorithm to adjust the dead zone compensation based on error and error rate of change according to the variable dead zone characteristic of the system. T. Zhijun et al. [9] designed asymmetrical dead zone compensation for proportional valves to improve the position control accuracy of the pre-bending machine. Coelho [10] uses cascaded adaptive control based on genetic algorithm to compensate the dead zone of the electro-hydraulic system. Meng, D. et al. [11] employs the least square type indirect parameter estimation algorithm with on-line condition monitoring to estimate the dead-zone parameters and some other important model parameters. B. In [12], the proposed controller has a parallel structure comprising an inverse generalized Prandtl-Ishlinskii (P-I) model-based feedforward controller, with both hydraulic dead-zone and flow saturation limits, for compensating asymmetric hysteretic behavior. Deng, W. X. [14] proposed a robust adaptive controller with uncertain dead zone compensation to achieve high performance position tracking of hydraulic cylinders. Peng, X. W. [15] designed a fuzzy dead-zone compensation algorithm that can adjust the amount of dead zone compensation online based on error and error rate of change, which effectively reduced the impact of nonlinearity and time-varying of the system. Wang, L. [16] designed output feedback controller based on model-free linear active disturbance rejection theory for electro-hydraulic proportional system with unknown dead-zone. In [17], dither compensation technology is presented to compensate the nonlinear dead zone of electro-hydraulic servo valve. The experimental results shows that the linearity of the load flow is improved obviously.

The current researches show that there are more studies on the control of proportional valves with positive

overlap. There is no flow output in a large area near the center of the valve, which also be called dead zone. However, there are few researches on the proportional valve which have a certain negative overlap and the flow in the center of the valve is more complicated. This article mainly focuses on the zero point control of the proportional valve with symmetric and unequal negative overlap.

## 2. Mathematical model of zero point

The diagram of the system is shown in Fig. 1, which the proportional valve has the symmetric and unequal negative overlap  $\Delta_1$  and  $\Delta_2$ ,  $x_v$  is the displacement of the spool the proportional valve,  $L$  is the maximum displacement of the spool the proportional valve,  $p_s$  is the system pressure from the pump,  $p_R$  is the return pressure to the tank,  $p_1$  is the pressure of the rodless cavity,  $p_2$  is the pressure of the rod cavity,  $q_1$  is the total flow of the rodless cavity,  $q_2$  is the total flow of the rod cavity,  $y$  is the displacement of the rod, and the  $F$  is the external load.

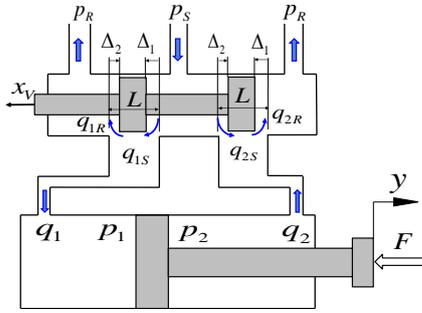


Fig. 1 The diagram of the proportional valve controlled cylinder system with negative overlap

The zero point of the proportional valve system with negative overlap changes with external load. In order to derive the mathematical model of the theoretical zero point, the flow of the proportional valve near the zero position needs to be analysed.

According to the classic orifice equation, the flow into the rodless cavity:

$$q_{1S} = \begin{cases} 0 & x_v \leq -\Delta_1 \\ a(p_s - p_1)^{1/2} (\Delta_1 + x_v) & -\Delta_{1S} \leq x_v \leq L - \Delta_1 \\ a(p_s - p_1)^{1/2} L & L - \Delta_1 \leq x_v \end{cases} \quad (1)$$

(cf., Ref. [18]), which,  $a = C_d \omega \sqrt{\frac{2}{\rho}}$ ,  $\omega$  is the area gradient of the valve,  $C_d$  is generally a constant, about 0.65.

The flow out of rodless cavity:

$$q_{1R} = \begin{cases} 0 & -x_v \leq -\Delta_2 \\ a(p_1 - p_R)^{1/2} (\Delta_2 - x_v) & -\Delta_2 \leq -x_v \leq L - \Delta_2 \\ a(p_1 - p_R)^{1/2} L & L - \Delta_2 \leq -x_v \end{cases} \quad (2)$$

According to the Eq. (1) and Eq. (2), when the valve is in the range of the overlap, the total flow of rodless

cavity can be expressed as follows:

$$q_1 = a(p_s - p_1)^{1/2} (x_v + \Delta_1) - a(p_1 - p_R)^{1/2} (\Delta_2 - x_v). \quad (3)$$

Similarly, the total flow model of rod cavity can be expressed as:

$$q_2 = a(p_s - p_2)^{1/2} (\Delta_2 - x_v) - a(p_2 - p_R)^{1/2} (x_v + \Delta_1). \quad (4)$$

When the speed of the system is zero, the output flow is also zero. The Eq. (3) and the Eq. (4) satisfy the following formula:

$$\begin{cases} a(p_s - p_1)^{1/2} (x_v + \Delta_1) = a(p_1 - p_R)^{1/2} (\Delta_2 - x_v) \\ a(p_s - p_2)^{1/2} (\Delta_2 - x_v) = a(p_2 - p_R)^{1/2} (x_v + \Delta_1) \end{cases} \quad (5)$$

Ignored the return pressure  $p_R$ , the pressure of the two cavities can be calculated according to the Eq. (5):

$$\begin{cases} p_1 = \frac{(\Delta_1 + x_v)^2}{(\Delta_1 + x_v)^2 + (\Delta_2 - x_v)^2} p_s \\ p_2 = \frac{(\Delta_2 - x_v)^2}{(\Delta_2 - x_v)^2 + (\Delta_1 + x_v)^2} p_s \end{cases} \quad (6)$$

Supposed the ratio of the overlap  $\chi$  is:

$$\chi = \frac{\Delta_1}{\Delta_2}. \quad (7)$$

Supposed the dimensionless displacement of the valve is as follows:

$$\bar{x}_v = x_v / \Delta_2. \quad (8)$$

When the speed of the system is zero, the displacement of the valve  $x_v$  is within the range of  $(-\Delta_1, \Delta_2)$ . According to the Eq. (7) and the Eq. (8), the dimensionless displacement of the valve satisfies:

$$-\chi < \bar{x}_v < 1. \quad (9)$$

Supposed the dimensionless pressure of the rodless cavity is  $\bar{p}_1 = p_1 / p_s$ , and the dimensionless pressure of the rod cavity is  $\bar{p}_2 = p_2 / p_s$ . According to the Eq. (6), the dimensionless pressure of the two cavities can be obtained as follows:

$$\begin{cases} \bar{p}_1 = \frac{(\chi + \bar{x}_v)^2}{(\chi + \bar{x}_v)^2 + (1 - \bar{x}_v)^2} \\ \bar{p}_2 = \frac{(1 - \bar{x}_v)^2}{(1 - \bar{x}_v)^2 + (\chi + \bar{x}_v)^2} \end{cases} \quad (10)$$

Supposed the dimensionless load pressure of system is as follows:

$$\bar{p}_L = \bar{p}_1 - \eta \bar{p}_2. \quad (11)$$

Which,  $\eta$  is the area ratio of rod cavity and rodless cavity.

Substituted the Eq. (11) into the Eq. (10), the relationship of the dimensionless load pressure and the dimensionless displacement of valve can be obtained:

$$\bar{p}_L = \frac{(\chi + \bar{x}_v)^2 - \eta(1 - \bar{x}_v)^2}{(\chi + \bar{x}_v)^2 + (1 - \bar{x}_v)^2}. \quad (12)$$

The dimensionless load pressure can be calculated by the pressure sensors of the system. So when the speed of the system is zero, the dimensionless displacement of the valve can be obtained according to the Eq. (12).

$$\bar{x}_v = \frac{\sqrt{\eta + \bar{p}_L} - \chi \sqrt{1 - \bar{p}_L}}{\sqrt{\eta + \bar{p}_L} + \sqrt{1 - \bar{p}_L}}. \quad (13)$$

Supposed the displacement of the valve is  $x_{v_0}$  when the speed of the system is zero, which is also called the theoretical zero point of the system. According to the Eq. (8) and the Eq. (13), the theoretical zero point  $x_{v_0}$  can be calculated as following.

$$x_{v_0} = \frac{\sqrt{\eta + \bar{p}_L} - \chi \sqrt{1 - \bar{p}_L}}{\sqrt{\eta + \bar{p}_L} + \sqrt{1 - \bar{p}_L}} \Delta_2. \quad (14)$$

The theoretical zero point of the system is related to the asymmetry of the overlap  $\chi$ , the asymmetry area ratio of the hydraulic cylinder  $\eta$ , and the dimensionless load pressure  $\bar{p}_L$ . When the system is ideally symmetrical, that is, the hydraulic cylinder is a symmetrical cylinder, the negative overlap of the spool valve is symmetrical and equal, and the load pressure is 0MPa, the theoretical zero point of the system is at zero position of the valve. In the actual system, the interference of the external load, the application of the asymmetrical cylinder, and the asymmetric overlap of the proportional valve will all cause the zero point of the system to not be at the zero position of the proportional valve. The offset of the zero point of the system will cause a large steady-state error when the system adopted the conventional control.

### 3. Verification of zero point model

The AMESIM simulation model of the valve controlled asymmetric cylinder system with negative overlap is shown in Fig. 2. The overlap value  $\Delta_1$  and  $\Delta_2$  are set 1% and 4% of the total stroke of the proportional valve, and the other parameters of the simulation model are shown in Table 1. The position of the hydraulic cylinder was controlled by the proportional control (P), and the proportional parameter is set to 0.12. In [6], the limit load of valve controlled cylinder system was analysed. Through the AMESIM simulation model, the theoretical zero-point value and the error of the zero point was analysed, when the system was is

loaded with the maximum load, respectively 4000 N and 2000 N.

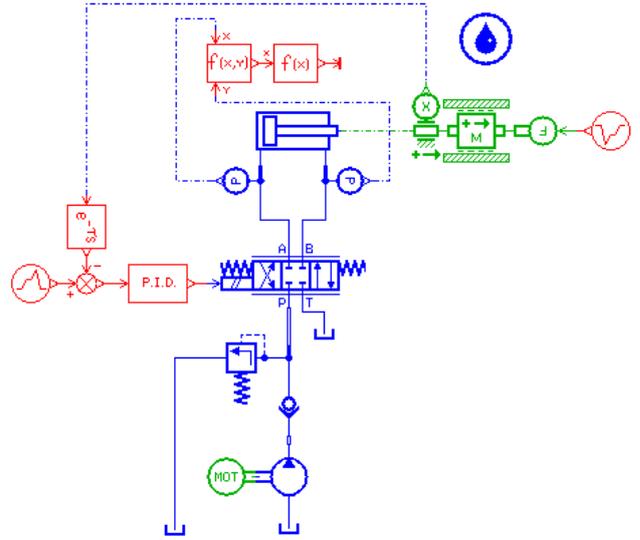


Fig. 2 The AMESIM simulation model of the valve-controlled asymmetric cylinder system with negative overlap

Table 1

The basic system parameters

Parameter, measurement unit	Value
Pressure of the system, MPa	5
Speed of the pump, r/min)	960
Displacement of the pump, ml/r	16
$\rho$ , kg/m <sup>3</sup>	850
Flow under the differential pressure of 1.5 MPa; L/min	9
Frequency of the valve, Hz	100
Radius of the hydraulic cylinder, mm	50
Rod diameter of the hydraulic cylinder, mm	28

Under the load force of 4000KN, the displacement, response of the system is shown in Fig. 3a. Due to the influence of the negative overlap, the zero point of the system is not at the zero position of the proportional valve, and the zero-point offset results a steady-state error of the displacement response, which is about 0.187 mm.

The simulated pressure response of the two chambers of the hydraulic cylinder is shown in Fig. 3b. Due to the influence of the inertia force and damping force of the hydraulic cylinder during the starting process, the load pressure has certain fluctuations. When the hydraulic cylinder is close to zero speed, the load pressure of the system is about 2.0 MPa. According to the pressure of the system and the pressure of the two chambers, the dimensionless load pressure is obtained according to Eq. (11), and the theoretical zero point of the system is calculated according to Eq. (14), which is shown in Fig. 3, c. In the simulation model, when the hydraulic cylinder is in the zero-speed state, the input signal of the valve is about 1.97%, and the value of theoretical zero point is about 1.88%. There is a certain error of 4.5% with the simulation zero point and the theoretical zero point.

Under the load force of -2000 KN, the displacement response is shown in Fig. 4, a, and the steady-state error of the displacement is about 0.08 mm. The theoretical

zero point of the system can be calculated through the pressure shown in Fig. 4, b, and the theoretical zero point is compared with the simulated signal of the valve, as is shown in Fig. 4, c. When the system is close to the zero-speed state, the actual zero point is about 0.80%, while the theoretical zero point is about 0.94%. The error of the theoretical zero point is shown in Fig. 4, d. When the system is stable, the error is about 17.5%.

As the load changes, the zero point of the system will shift, and the calculation error of the theoretical zero point is relatively large in some certain pressure states. In order to compensate the steady-state error caused by the zero point, a compensation controller needs to be designed.

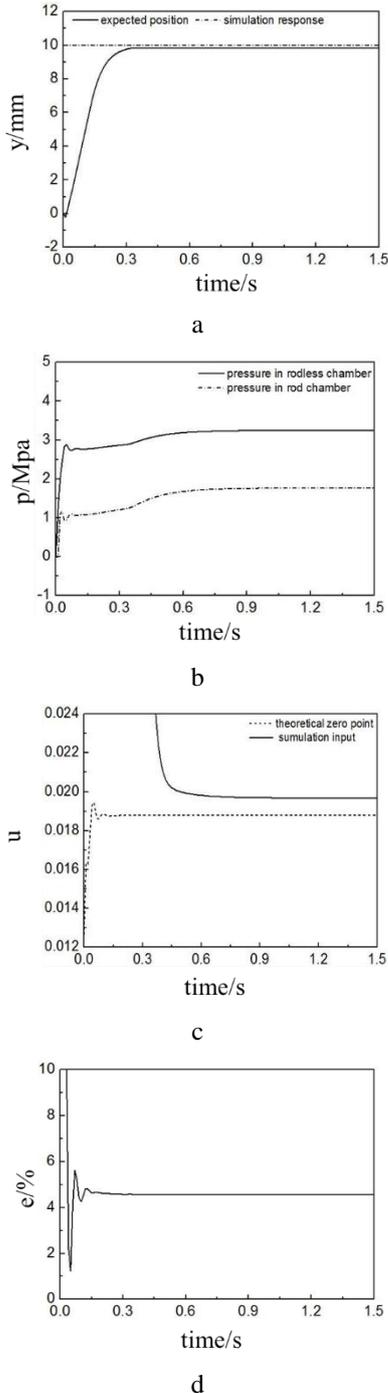


Fig. 3 The simulation results of P control under 4000 KN. a) Displacement response; b) Pressure; c) Input signal of the proportional valve and the theoretical zero point; d) Error of the zero point

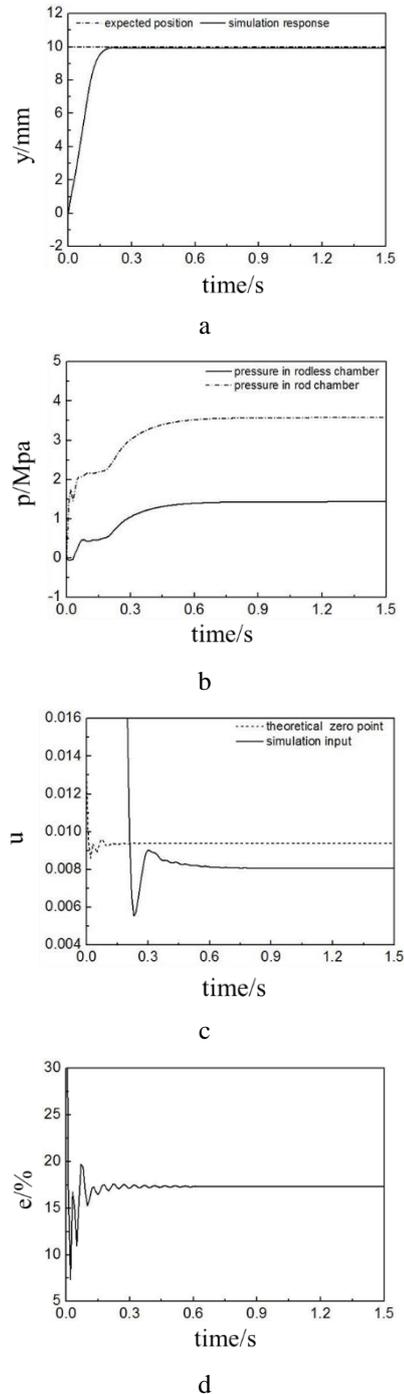


Fig. 4 The simulation results of P control under -2000 KN. a) Displacement response; b) Pressure; c) Signal; d) Error of zero point

#### 4. Design of compound controller

Due to the influence of the overlap, the system has a large steady-state error. The use of conventional proportional and integral (PI) controller to eliminate the steady-state error will also cause a large overshoot. In order to compensate the offset of the zero point, zero compensation control is added when the error of the system is gradually stable. In order to judge the error trend, the moving average of the absolute error is adopted, as is shown in Fig. 5. When the moving average  $MA_t < \gamma$ , the system starts the compensation of the zero point.

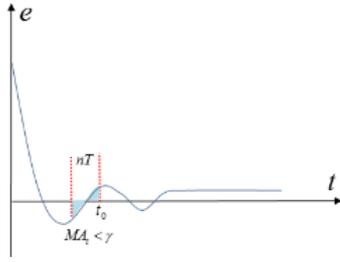


Fig. 5 The error line of the system

The error of the average line is shown in Eq. (15):

$$MA_t = \frac{|e_t| + |e_{t-T}| + |e_{t-2T}| + \dots + |e_{t-(n-1)T}|}{nT}, \quad (15)$$

which:  $e_t$  is the error in the current period;  $e_{t-T}$  is the previous period error;  $T$  is the sampling time;  $n$  is the number of samples.

Because there is a certain error between the theoretical zero point and the actual zero point, the zero point compensation controller integrates the feedback error on the basis of the theoretical zero point, so that the steady-state error is gradually reduced. The zero-point compensation function is shown in Eq. (16):

$$\Delta u(t) = (u_0 + k \int edt) \cdot (|MA_t| < \gamma), \quad (16)$$

which:  $\Delta u(t)$  is the compensated signal of zero point;  $u_0$  is the input signal corresponding the theoretical zero point  $x_{v_0}$ ;  $k$  is the integral constant.

According to the Eq. (16), the block diagram of the compound controller is shown in Fig. 6.

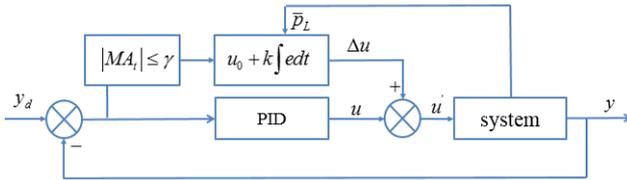


Fig. 6 Block diagram of compound controller for the zero-point compensation and the proportional control

### 5. Simulation of compound controller

#### 5.1. Simulation of controller parameter

The controller was built according to the simulation model in Fig. 2, and the external load is set to 0N, the number of samples was set to 20. To study the law of the value of  $\gamma$  on the response of the system, the displacement response was simulated when the value of  $k$  was set to 0.7 and the value of  $\gamma$  was respectively set to 1, 2.5, 5 and 10. The simulated response is as shown in Fig. 7.

According to the Fig. 7, b, when the value of  $\gamma$  is larger, the zero-point compensation start earlier and the system response is faster, but the larger value of  $\gamma$  will cause larger overshoot. When the value of  $\gamma$  is less than the moving average of the steady-state error, the zero-point compensation control cannot be started. So, the value of  $\gamma$  should be set in the optimal range to ensure the performance of the system.

To study the law of the value of  $k$  on system response, the displacement response was simulated when the value of  $\gamma$  was set to 2.5 and the value of  $k$  was respectively set to 0.3, 0.7, 1.4 and 2.1. The simulated response is as shown in Fig. 8.

According to the Fig. 8, b, when the value of  $k$  is more than 0.7, the overshoot and oscillation of the system increases accordingly. When the value of  $k$  is 0.5, there is a small overshoot about 2.5 % and the response time is about 1.0 s, which is less than when the value of  $k$  is 0.35.

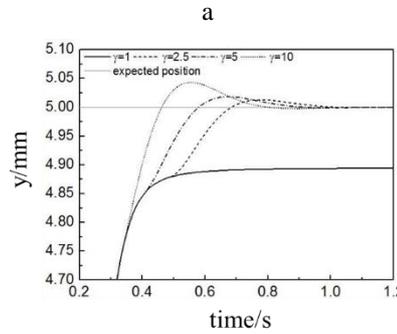
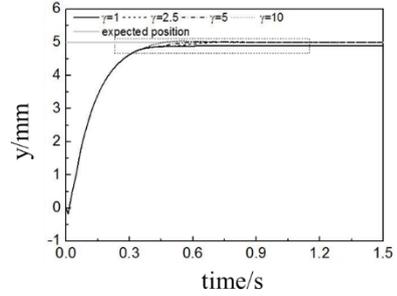


Fig. 7 The displacement response of zero point compensation control with different  $\lambda$ . a) Displacement response; b) Enlarged image of (a)

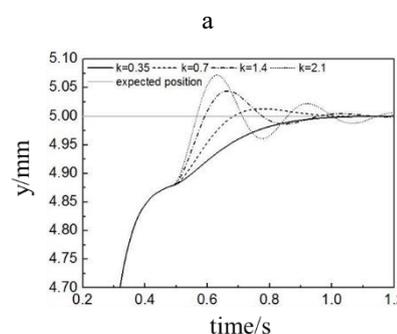
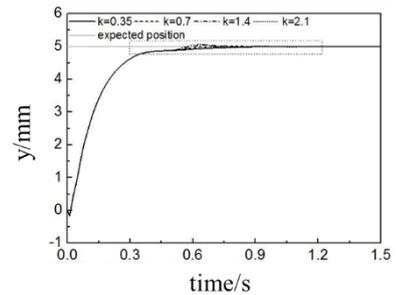


Fig. 8 The displacement response of zero point compensation control with different  $k$ . a) Displacement response; b) Enlarged image of (a)

## 5.2. Simulation of compound controller under load interference

In order to test the displacement response of the controller under load interference, transient load interference and sinusoidal load interference were set respectively.

The position of the hydraulic cylinder was controlled by the proportional control (P), and the proportional parameter is set to 0.08. In order to get the best response of the system, the value of  $\lambda$  was set to 2.5 and the value of  $k$  was set to 0.5 according to the simulation results in Section 5.1.

Fig. 9 showed the displacement response and the signal of the valve under transient load interference, which the load was added at 0 s, 1.5 s and 2.5 s, and the force was respectively 4000 N, 0 N and -2000 N. As is shown in Fig. 9a, when the system is under the load force of 4000 N, 0 N, -2000 N, there are steady-state errors of 0.25 mm, 0.15 mm, and 0.10 mm respectively with the P controller. However, through the compound controller with the zero-point compensation, the steady-state error of the system is within 0.01 mm under different load forces. As is shown in Fig. 9, b, the signal of the zero-point compensation coincides with the output signal line of the compound controller, indicating that the zero-point compensation controller can quickly find the zero point of the system.

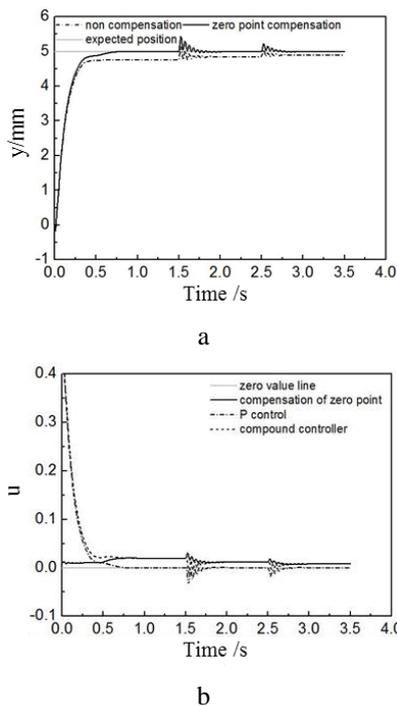


Fig. 9 The simulation results of zero point compensation control with  $k = 0.5$ ,  $\lambda = 2.5$ . a) Displacement response; b) Signal

Fig. 10 showed the displacement response and the signal of the valve under sinusoidal load interference, which the average load is 3000 N, the amplitude is 1000 N, and the frequency is 1 Hz. In order to make the controller find the zero point quickly under variable load, the value of  $k$  was set to 3.0, and the value of  $\lambda$  was also set to 2.5.

As is shown in Fig. 10, a, there is a steady-state error of 0.23 mm, and the error fluctuates in the range of 0.05 mm as the load fluctuates with the P controller. While

there is nearly no steady-state error and the error fluctuates in the range of 0.03 mm with the compound controller. As is shown in Fig. 10, b, the signal of the zero-point compensation follows with the output signal line of the compound controller, indicating that the zero-point compensation controller can find the zero point when the system is under the variable load.

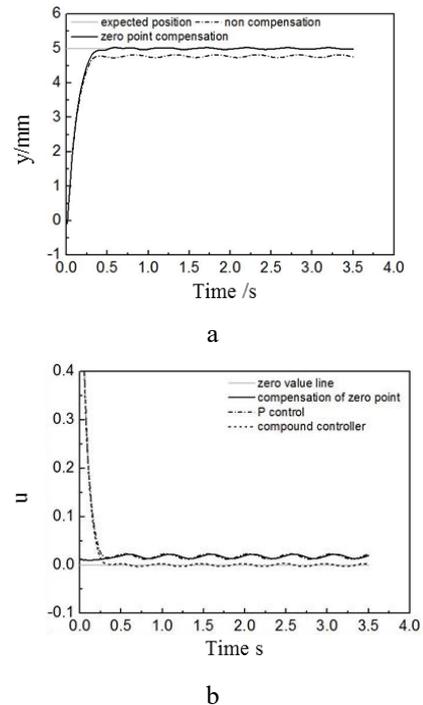


Fig. 10 The simulation results of zero point compensation control with  $k = 3$ ,  $\lambda = 2.5$ . a) Displacement response; b) Signal

## 6. Conclusions

In order to improve the control accuracy of the negative overlapped valve controlled asymmetric cylinder system, a theoretical zero-point model was established. And based on the theoretical mathematical model of the zero point, a compound controller with the proportional control and the zero point compensation was designed. Through the simulation results of the controller parameters, the setting rules of the control parameters are obtained. With the optimal control parameters, the simulation results show that the compound controller with the P controller and zero point compensation controller can find the zero point quickly and greatly improve the positioning accuracy of the system under transient load and sinusoidal load.

## Acknowledgments

The research is funded by the Scientific Research Project of the Education Department of Hunan Province (21C1557) and the Announcing and Leading Project in the key areas of the Science and Technology Board of Foshan City (2120001009772).

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#### ZERO POINT COMPENSATION CONTROL OF THE PROPORTIONAL VALVE WITH NEGATIVE OVERLAP

#### S u m m a r y

In order to improve the performance of the negative overlapped valve controlled asymmetrical cylinder system, a theoretical zero point model of the valve with symmetric and unequal negative overlap was established. The simulation results showed that the zero point of the system will shift as the load changes, and the error of the theoretical zero point is relatively larger in some certain pressure states. In order to find the zero point quickly and steadily and improve the positioning accuracy of the system, a zero point compensation controller based on theoretical zero point plus error integral control was designed. Compared the pure P controller with the compound control of P controller and zero point compensation controller, the control accuracy of the system has been greatly improved under transient load and sinusoidal load interference.

**Keywords:** electro-hydraulic servo system, negative overlapped valve, zero point compensation.

Received September 1, 2021

Accepted August 2, 2023



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