Analyzing of influencing factors on dynamic response characteristics of double closed-loop control digital hydraulic cylinder

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Abstract
A digital hydraulic cylinder that uses mechanical closed-loop feedback regulation to realize double closed-loop control of displacement and velocity is designed in this study. Based on the working principle, the transfer function model and AMESim simulation model of digital hydraulic cylinder were established. The influence of pre-opening distance of the four-way reversing valve, ball-screw lead, inner diameter, and load on the displacement error, adjustment time, overshoot was analyzed. The standard regression analysis about the results was carried out as well. The results show the pre-opening distance of the four-way reversing valve has the greatest influence on the dynamic response characteristics of double closed-loop control digital hydraulic cylinder, and three Pearson coefficients are all greater than 0.9. The non-inertial load has a certain influence on the displacement error of the digital hydraulic cylinder. The optimal value of the ball-screw lead and inner diameter which have less effect on the dynamic characteristics of the system are 10mm and 90mm, respectively. Furthermore, the performance tests of the digital hydraulic cylinder sample were carried out.

Keywords: Double closed-loop control digital hydraulic cylinder, Dynamic response characteristic, Transfer function, AMESim, Standard regression analysis

1. Introduction

Due to the high power density, high reliability and diversified control methods, hydraulic systems are widely applied in industrial hydraulics and engineering machinery (Nakkarat and Kuntanapreeda, 2009). Compared with servo-hydraulic systems, digital hydraulic systems with mechanical closed-loop feedback regulation turn complex displacement and velocity control into a single pulse control. It not only has the characteristics of compact structure, simple control, high frequency response and precision, but has great application potential in fields with high environmental adaptability and cost control requirements, such as engineering machinery (Guo et al., 2013). The double closed-loop control digital hydraulic cylinder combines the valve, hydraulic cylinder, and feedback and drive devices to achieve digital control of hydraulic cylinder movement. Since the structural parameters, load and other nonlinear factors of the hydraulic components are closely related to the dynamic response characteristics of the system, it is of great significance for the design, manufacture and application to quantitatively analyze the main factors affecting the dynamic response characteristics of double closed-loop control digital hydraulic cylinder.

As the main actuators of hydraulic system, hydraulic cylinder with high positioning accuracy and high-quality dynamic characteristics has always been a concern of researchers, and thus emerges servo control hydraulic system and advanced control algorithms, such as robust controllers (Kim and Choi, 2014; Loukianov et al., 2009; Choi, 2013), quantitative feedback control theory (Ren et al., 2016; Ahn and Dinh, 2009), fuzzy controllers (Mendonça et al., 2012;
Ji and Li, 2009), adaptive controllers (Yao et al., 2014; He et al., 2017), extended disturbance observer (Kim et al., 2013; Guo et al., 2015) and some compound control methods (Sun et al., 2013; Guo et al., 2017). These measures not only greatly increase the complexity and operating costs of the system, but decrease the average energy efficiency of hydraulic system to about 21% owing to a large amount of throttling and overflow in the pipeline (Pham and Ito, 2016; Yang and Pan, 2015). Compared with servo control hydraulic systems, the systems with digital hydraulic components have advantages in reducing system noise, heat production, and energy consumption (Qi et al., 2014). Digital hydraulic components mainly consist of digital hydraulic valves and digital hydraulic cylinders. Yang et al. (Yang et al., 2016) summarized the development history, research status and application fields of digital hydraulic valves based on incremental and high-speed switching. At present, it is the emphasis to improve the frequency response characteristics under high pressure and large flow rate. Based on the transfer function model of the digital hydraulic cylinder, it was concluded that the load and asymmetric structure were the main reasons for the non-consistent behavior in different direction of the digital hydraulic cylinder (Chen et al., 2014; Bai et al., 2009). The state space method was used to analyze the influence of nonlinear factors such as time delay and dead zone on the tracking characteristics and stability of hydraulic cylinders (Magyar et al., 2010; Hua et al., 2008). Based on the nonlinear state space model of the digital hydraulic cylinder, the frictional and sealing characteristics were analyzed in MATLAB/Simulink. It was found that the negative damping frictional characteristic of the seals when the hydraulic cylinder at low speed is a primary reason for its creep (Xiao et al., 2008; Peng et al., 2011). The simulation model of digital hydraulic cylinders was established by the electromechanical and hydraulic integration simulation software AMESim (Wang et al., 2007; Pan et al., 2012). The results showed that improving the input signal frequency, reducing the reduction ratio, adopting conical valve port and increasing load viscosity are beneficial to reduce the reversing impact.

The above researches have done a lot on the design, improvement and application of double closed-loop control digital hydraulic cylinder. At present, there are relatively few studies on the influence of basic structural parameters and loads on the dynamic response characteristics of digital hydraulic cylinders. In this paper, the mathematical model and AMESim simulation model are established based on the working principle of digital hydraulic cylinder. The main parameters affecting the dynamic response characteristics are determined and quantitatively analyzed. The structural parameters of the digital hydraulic cylinder are optimized. Finally, the results are verified by performance tests.

2. Nomenclature

\[
\begin{align*}
A_l &= \text{effective area of the rod-less chamber} \\
A_r &= \text{effective area of the rod chamber} \\
B_p &= \text{viscous damping coefficient of load} \\
C_d &= \text{flow coefficient of valve port} \\
C_e &= \text{external leakage coefficient} \\
C_i &= \text{internal leakage coefficient} \\
C_{na} &= \text{coefficient of additional leakage} \\
C_{ne} &= \text{coefficient of equivalent leakage} \\
F_L &= \text{non-inertial load acting on the piston} \\
K &= \text{spring stiffness of load} \\
K_C &= \text{flow-pressure coefficient} \\
K_{ip} &= \text{a coefficient which only relates to } n \\
K_{ic} &= \text{total flow-pressure coefficient} \\
K_{op} &= \text{open-loop amplification factor} \\
K_q &= \text{flow gain coefficient of the valve} \\
m_l &= \text{total mass of load including piston} \\
n &= \text{ratio of } A_r \text{ to } A_l \\
p_0 &= \text{pressure of tank} \\
p_1 &= \text{pressure of the rod-less chamber} \\
p_2 &= \text{pressure of the rod chamber} \\
p_s &= \text{pressure of oil source} \\
p_L &= \text{load pressure} \\
q_i &= \text{supplied flow to rod-less chamber} \\
q_2 &= \text{return flow from rod chamber} \\
q_L &= \text{load flow} \\
S_1 &= \text{lead of screw nut pair} \\
S_2 &= \text{lead of ball screw} \\
V_1 &= \text{equivalent volume of rod-less chamber} \\
V_2 &= \text{equivalent volume of rod chamber} \\
V_t &= \text{total control volume of hydraulic cylinder} \\
x &= \text{input displacement} \\
x_s &= \text{spool displacement} \\
x_p &= \text{piston rod displacement} \\
\beta_e &= \text{effective bulk modulus} \\
\theta &= \text{step angle of stepping motor} \\
\theta_m &= \text{feedback angle}
\end{align*}
\]
3. Mathematical model

Figure 1 shows the schematic presentation of self-designed double closed-loop control digital hydraulic cylinder, which is mainly composed of a stepping motor, three-position four-way reversing valve, cylinder block and feedback ball screw. The working principle is as follows: the stepping motor, which drives the spool to rotate synchronously, is driven to rotate by the subdivision driver under the action of the modulated pulse signal. Due to the action of the screw nut pair at the other end of the spool, the spool generates axial displacement while rotating, and the throttling grooves are opened. It is the rotation direction and angular velocity of the stepping motor that control the flow direction and flow rate of the hydraulic oil in the circuit, thereby the direction and speed of the movement of hydraulic cylinder. The piston rod moves linearly under the pressure of one chamber of cylinder. Profiling nut integrated with the piston drives the ball screw to rotate in an opposite direction. Finally, through the action of the coupling and screw nut pair, the spool moves reversely to achieve negative feedback of position until the throttling grooves are closed. The digital hydraulic cylinder which integrates electricity-mechanical conversion components, control components, actuators and feedback mechanisms achieves double closed-loop control of displacement and velocity. It has few links, high frequency response, compact structure, simple control, and unparalleled advantages in anti-pollution capability and multi-cylinder coordinated motion control.

On the basis of rationally simplifying the digital hydraulic cylinder system, the basic state equations describing the valve-controlled asymmetric hydraulic cylinder can be represented as follows (Han et al., 2012; Ylinen et al., 2014):

\[
q_L = K_q x_v - K_p p_L
\]  
(1)

\[
q_L = \frac{q_1 + nq_2}{1+n^2} = A_1 \frac{dx_p}{dt} - C_{ua} p_s + C_{uc} p_L + K \frac{V_q}{\beta_c} \frac{dp_L}{dt}
\]  
(2)

\[
A_1 p_L = A_1 (p_1 - np_2) = A_1 p_1 - A_2 p_2 = m \frac{d^2 x_p}{dt^2} + B_p \frac{dx_p}{dt} + K x_p + F_L
\]  
(3)

Where \(q_L\) denotes load flow, \(p_L\) denotes load pressure, \(K_q\) denotes the flow gain coefficient of the valve, \(x_v\) denotes the spool displacement, \(K_p\) denotes flow-pressure coefficient, \(n = A_2 / A_1\), \(A_1\) and \(A_2\) denote the effective area of the rod-less chamber and the rod chamber respectively, \(q_1\) and \(q_2\) denote the supplied flow to rod-less chamber and return flow from rod chamber respectively, \(x_p\) denotes piston displacement, \(p_s\) denotes pressure of oil source, \(C_{ua}\) denotes the coefficient of additional leakage, \(C_{uc}\) denotes the coefficient of equivalent leakage, \(K_i\) denotes a coefficient which only relates to \(n\), \(V_q\) denotes total control volume of hydraulic cylinder,
denotes effective bulk modulus, \( p_1 \) and \( p_2 \) denote the pressure of two chambers, \( m_\ell \) denotes the total mass of load, \( B_p \) denotes the viscous damping coefficient of piston and load, \( K \) denotes the spring stiffness of load, \( F_L \) denotes the non-inertial acting on the piston.

Under continuous input impulse, the displacement of the spool controlled by stepping motor can be represented as follows:

\[
x_s = \frac{S_1}{2\pi} \frac{2\pi \times (x - x_p)}{S_2} = \frac{(x - x_p) S_1}{S_2}
\]

(4)

Where \( x \) denotes the input displacement, \( S_1 \) and \( S_2 \) denotes the lead of screw nut pair and ball screw, respectively.

Therefore, the transfer function model of digital hydraulic cylinder can be represented as follows:

\[
x_p = \frac{K_a S_1}{A_2 S_2} x + \frac{C_{\alpha}}{A_1} p_s - \frac{K_v}{A_1^2} \left( 1 + \frac{K V_p}{\beta v K_v} \right) F_L
\]

\[
\frac{m V K_v}{\beta v A_1^2} s^3 + \left( \frac{m K_v}{\beta v A_1} + \frac{B_p V K_v}{A_1^2} \right) s^2 + \left( 1 + \frac{B_p K_v}{\beta v A_1^2} + \frac{K V_p}{A_1^2} \right) s + \frac{K_v K_v}{A_1} + \frac{K_v S_1}{A_1 S_2}
\]

(5)

Where \( K_{v_h} \) denotes the total flow-pressure coefficient, \( K_v = K_v + C_v \).

The detailed derivations of the above equations are given in Appendix. Ignoring the interference caused by the load and the leakage of the oil source, the open-loop transfer function of the feedback loop in the digital hydraulic cylinder can be represented as follows after the simplification:

\[
G_v(s) = G(s) H(s) = \frac{\omega_h^2 K_v}{s^2 + 2\beta_h s + \omega_h^2}
\]

(6)

Where \( K_v \) denotes open-loop amplification factor, \( K_v = \frac{K_a S_1}{A_2 S_2} \), \( \omega_h \) denotes hydraulic natural frequency, \( \beta_h \) denotes hydraulic damping ratio.

Known by the principle of automatic control, the feedback loop of digital hydraulic cylinder is a type I system. Under the ramp signal, there is a velocity error, which is inversely proportional to the open-loop amplification factor \( K_v \). Compared with the conventional valve-controlled symmetrical hydraulic cylinder model, the system output gain is related to the lead of ball screw, the effective area of rod-less chamber, and the direction of movement \( K_q \).

4. Establishment of simulation model of digital hydraulic cylinder

According to the physical structure and operational principle of digital hydraulic cylinder, the simplified simulation model was established by AMESim software (AMESim13.0), which can not only take into account the non-linear factors such as the compressibility of the oil, Coulomb friction, but also the modeling and simulation of mechanical-electrical-hydraulic integration. The model mainly includes hydraulic power unit, stepping motor, four-way reversing valve, hydraulic cylinder and feedback mechanisms etc., as shown in Fig. 2.

In the simulation model, the pressure of oil source is set by the relief valve of the hydraulic power unit. Through the angle conversion coefficient, the input displacement is converted into the rotation angle of stepping motor, which is also the rotation angle of the spool of four-way reversing valve. In order to facilitate the setting of mass, stroke, structural characteristics of throttling grooves and other parameters, the body of four-way reversing valve is established by the model unit in the Hydraulic Component Design. The hydraulic cylinder is also established through Hydraulic Component Design. The end of the piston rod of the hydraulic cylinder is connected to a load unit which can
accomplish the set in amplitude and form. The feedback mechanisms mainly include screw nut pair and ball screw, both of which transmit motion and power information through a threaded connection. The screw nut mechanism model is used to simulate the feedback elements of the digital hydraulic cylinder, and the lead, friction coefficient are set.

According to the operating requirements of double closed-loop control digital hydraulic cylinders, the main structural parameters of the system are preliminarily designed. Some of the parameter values are shown in Table 1.

![Fig. 2 Simulation model of digital hydraulic cylinder](image)

| Table 1 Partial parameters values of digital hydraulic cylinder |
|---------------------------------------------------------------|
| **Name** | **Parameters** | **Value** |
|-----------|----------------|----------|
| Oil source | Density / (kg/m³) | 850 |
| | Modulus of elasticity / MPa | 1700 |
| | Pressure / MPa | 4 |
| Four-way reversing valve | Diameter / mm | 16 |
| | Area of throttling groove / mm² | 4*4 |
| | Number of circumferential troughs | 4 |
| | Lead / mm | 2 |
| Digital hydraulic cylinder | Piston mass / kg | 10 |
| | Piston diameter / mm | 90 |
| | Piston rod diameter / mm | 40 |
| | Stroke / mm | 390 |
| | Viscous friction coefficient / (N·s/m) | 60 |
| Ball screw | Lead / mm | 10 |
| | Coulomb friction coefficient | 0.01 |

5. Result and discussions

As can be seen from the Eq. (6) in section 3, the dynamic response characteristics of digital hydraulic cylinder are affected by many factors. Combined with the actual working conditions, the influence of the pre-opening distance of four-way valve, ball-screw lead, inner diameter and load on the displacement error, adjusting time, overshoot of double closed-loop control digital hydraulic cylinder is analyzed through the simulation model.
5.1 Influence of the pre-opening distance of four-way reversing valve

The four-way reversing valve is an important component for controlling the operation of the digital hydraulic cylinder, whose pre-opening type and distance are closely related to the dynamic characteristics of the system. Five four-way reversing valve models are established, for which the pre-opening distances are 0.5mm, 0.25mm, 0mm, -0.25mm, -0.5mm, respectively. Negative distance indicates a positive lap spool. The input displacement of digital hydraulic cylinder is set as to be 0m from 0s to 0.5s, and linearly increase to 0.2m from 0.5s to 4.5s. The displacement and velocity curves of the digital hydraulic cylinder under the ramp signal (rated running velocity) are shown in Fig. 3.

![Graphs showing displacement, velocity, adjusting time, and overshoot for different pre-opening distances](image)

As can be seen from Fig. 3, the pre-opening type and distance of four-way reversing valve have an important influence on the dynamic response characteristics of the hydraulic cylinder. Graph (a) and (b) in Fig. 3 show that the hydraulic cylinder with a negative lap spool has a slight negative displacement at the initial moment, which is about 0.1m. It is the result of the pressure of the oil source acting on both side of the hydraulic cylinder at the same time. And the effective areas of the two chambers are different, so that the piston rod has a slight displacement along the axial direction. After a quick and short adjustment, the initial steady state of the system is established, and there is also a certain displacement error. For the valve with a positive lap spool, the stepping motor is required to drive the spool to rotate a certain angle to cross the dead zone. The displacement error of piston rod at the steady state increases with the decrease of the pre-opening distance. When the pre-opening distance is -0.5mm, the displacement error is the largest, which is 4.32mm. While the pre-opening distance is zero or positive, the displacement error at steady state is quite small, which satisfies precision requirement of 2mm.

According to the velocity response curves in Fig. 3 (c) and (d), with the decrease of pre-opening distance, the adjusting time and overshoot of the digital hydraulic cylinder increase significantly. The overshoot and adjusting time of the spool with a 0.5mm pre-opening distance is 5.44% and 0.1s, respectively. However, in the case of valve with a...
positive lap spool, the overshoot is 117.2% and the adjustment time is over 1.5s. It also results in some fluctuations in the displacement error curves. The four-way reversing valve with a negative lap spool can not only improve the response speed, but also increase the damping and stability of the system. Therefore, the target value of velocity can be quickly reached and stabilized.

The velocity of digital hydraulic cylinder with a negative lap spool has a wide range fluctuation for a long time, which reduces the security of the system. When the pre-opening distance is positive, the throttling grooves are always open, causing a large amount of power loss caused by leakage. Therefore, considering the accuracy and energy efficiency requirements of the system, a 0.5 mm pre-opening distance is approximately optimal.

5.2 Influence of ball-screw lead

As an important component of feedback mechanisms, the lead of ball screw is critical to positioning accuracy and velocity characteristics of digital hydraulic cylinder. Three ball screw models are established, for which the leads are 8 mm, 10 mm and 12 mm respectively. The input displacement of digital hydraulic cylinder is the same as that in section 5.1. The displacement and velocity curves of the digital hydraulic cylinder under the ramp signal are shown in Fig. 4.

Graph (a) and (b) in Fig. 4 show that the displacement error of digital hydraulic cylinder increases with the increase of ball-screw lead. The displacement error is largest, which is 1.01mm, when the lead is 12 mm. However, the change is small. The displacement curves of the hydraulic cylinder have a higher degree of coincidence. The displacement error is increased by 0.33mm when the ball-screw lead increases from 8mm to 12mm. When the lead of ball screw becomes smaller, the ratio of S1 to S2 increases, the open-loop amplification factor of the digital hydraulic cylinder system increases, which can improve the response speed and decrease displacement error under the ramp signal. The theoretical displacement error of the ball screw with a 12mm lead is 1.5 times than that of 8mm, which can be calculated from Eq. (6). The result obtained from simulation is 1.494, showing the accuracy of the simulation results.
However, the increase in open-loop amplification factor also makes the system stability slightly worse. As can be seen from Fig. 4 (c) and (d), when the lead is 8mm, the speed fluctuation is the largest. The overshoot and adjustment time is 12.56% and 0.146s, respectively. When the lead is 12 mm, the overshoot is 0.6% and the adjustment time is shortened to 0.111s. In general, the lead of ball screw has less effect on the dynamic response of the digital hydraulic cylinder.

Increasing the ball-screw lead can enhance the stability of the hydraulic cylinder, but the positioning accuracy is lower. In practice, it is not easy to replace once the ball-screw lead is determined. Therefore, the ball screw with a larger lead can be preferentially selected on the premise of satisfying the requirements of accuracy and response speed.

### 5.3 Influence of the inner diameter of digital hydraulic cylinder

Due to machining error and wear, the dynamic characteristics of hydraulic cylinder will gradually change. When multiple hydraulic cylinders work in coordination, the effect of the inner diameter change on the performance will be more prominent. The input displacement of digital hydraulic cylinder is the same as that in section 5.1. The dynamic response curves of the digital hydraulic cylinder with the inner diameter of 86mm, 88mm, 90mm, 92mm and 94mm are shown in Fig. 5.

![Response curves of digital hydraulic cylinder under different inner diameters](image)

**Fig. 5 Response curves of digital hydraulic cylinder under different inner diameters**

It can be inferred from Eq. (6) that with the increase of inner diameter of hydraulic cylinder, the effective area of the rod-less chamber increases and the open-loop amplification factor decreases. Since the digital hydraulic cylinder is a type I system, the steady-state displacement error under ramp signal increases. The maximum displacement error is 0.948mm. However, the change of inner diameter has little effect on the displacement error. The displacement error increases 0.204mm when the inner diameter increase from 86mm to 94mm, as shown in Fig. 5 (b).

The inner diameter is smaller with a larger open-loop amplification factor. The response speed and the positioning accuracy are higher, whereas the stability is poorer. Graph (c) and (d) in Fig. 5 shows there is an overall downtrend in overshoot with the increase of inner diameter of the hydraulic cylinder. But the amplitudes are all below 7%. The adjusting time is about 0.1s. Compared with the effect of pre-opening distance of four-way reversing valve and the lead...
of ball screw, the effect of inner diameter on dynamic characteristics of digital hydraulic cylinder can be neglected.

In the mechanical design process, the inner diameter is selected in standard values according to the load capacity. For safety considerations, the variation of the inner diameter in simulation exceeds 4% of the design value, which is unlikely to occur in actual situations. When the load requirements are met, the hydraulic cylinder with a smaller inner diameter can be preferentially selected to meet the economic requirements. The dynamic response of digital hydraulic cylinder with an inner diameter of 90mm meets the design requirements.

### 5.4 Influence of load

Ensuring the accuracy and stability of digital hydraulic cylinders under different loads, especially at rated loads, is essential to improve their suitability. Four constant non-inertial load models with the magnitude of 0kN, 5kN, 10kN and 15kN are established. The input displacement of digital hydraulic cylinder is the same as that in section 5.1. Fig. 6 shows the dynamic response curves of the digital hydraulic cylinder with different non-inertial loads.

As shown in Fig. 6 (a) and (b), both at the initial moment and steady state, the displacement error increases with the increase of non-inertial load. The system is almost at the rated pressure when the non-inertial load is 15kN, and the steady-state displacement error is 1.80mm, which is 2.12 times of the displacement error without non-inertial load. It shows the obvious effect of non-inertial load on the positioning accuracy. The load increases the pressure difference between the two chambers of the hydraulic cylinder and suppresses the output of the system. The difference corresponds to the third term of numerator in Eq. (5). At the same time, non-inertial loads increase the damping of the system. There is little change in the velocity response curves. The adjusting time maintains at about 0.1s, and there is no obvious fluctuation in velocity. In general, the change of response characteristics caused by non-inertial load can meet the design requirements. However, appropriate compensation or other improvement measures are needed in high precision and ultra-high precision situations.
Considering the fact that the introduction of external load tends to increase the inertial mass of the system, the influence of inertial mass on the response characteristics is analyzed by setting the quality of the piston rod. The inertial masses are set to 200kg, 600kg, 1000kg, 1400kg, and 1800kg respectively (the gravity of the 1800kg inertial mass is approximately equal to the maximum thrust of digital hydraulic cylinder). It is found that the increase of inertial mass hardly has effect on the displacement error of the digital hydraulic cylinder, which can be explained by the influencing factors of the open-loop amplification factor in Eq. (6). However, the natural frequency of the system increases when the inertial mass increases, which reduces the response speed of the system, resulting in a slight increase in overshoot and adjustment time. When the inertia mass increases from 200kg to 1800kg, the overshoot is increased from 5.4% to 11.9%, and the adjustment time is increased to 0.138s.

The static friction and dynamic friction between the piston and the cylinder are the basic performance parameters of the hydraulic cylinder, which have a certain influence on the dynamic response characteristics. Referring to the design criteria of minimum starting pressure of hydraulic cylinder, the static friction of the digital hydraulic cylinder is set to 600N, 800N, 1000N and the dynamic friction is set to 200N, 400N, 600N to meet the principle that the dynamic friction is not greater than the static friction. Fig. 7 shows the dynamic response characteristics of digital hydraulic cylinders under different static and dynamic friction.

![Displacement error](image1)

![Overshoot](image2)

![Adjusting time](image3)

**Fig. 7 Response curves of digital hydraulic cylinder under different static and dynamic friction**

The digital hydraulic cylinder needs to overcome the maximum static friction and the dynamic friction respectively when it starts and moves. Therefore, the static friction has no effect on the steady-state displacement error of the system. Due to fact the dynamic friction can be regarded as a kind of non-inertial load, the steady-state displacement error increases from 0.817 mm to 0.839 mm with the increase of dynamic friction. It can be seen from Fig. 7 (b) and (c) that when the dynamic friction is the same, there is an uptrend in the overshoot and adjusting time as the static friction
increases. It is because that the frictional force is changed from static friction to dynamic friction when the hydraulic cylinder starts moving. The greater the difference between the two kinds of friction, the greater the impact load of the system, and the longer the adjustment process to reach the steady state. When the dynamic friction is 200N, the overshoot and adjustment time at 1000N static friction are 53.72% and 0.19s, respectively. And the overshoot and adjustment time at 600N static friction are reduced to 1.93% and 0.08s, respectively. When the static friction is the same, there is a downtrend in the overshoot and adjusting time as the dynamic friction increases, which is also related to the magnitude of the impact load caused by the change of state of hydraulic cylinder. In the manufacturing process of digital hydraulic cylinders, the strict control of sealing friction, contact surface roughness, assembly precision and other measures will help to further improve the response characteristics of digital hydraulic cylinders.

Figure 8 shows the relationship between steady-state displacement error and non-inertial load under different pre-opening distances. It can be found the load has less influence on the dynamic response characteristics of digital hydraulic cylinder than pre-opening distance. When the pre-opening distance is 0.5mm, the steady-state error under nominal pressure is 1.79mm, and for the distance of 0mm, the corresponding load is only 0.45kN. As the load increases, the displacement errors under different pre-opening distances increase, and the steady-state error increases by about 0.9mm under nominal pressure over that under no-load.

![Figure 8](image_url)

**Fig. 8 Relationship between steady-state displacement error and non-inertial load under different pre-opening distances**

### 5.5 Standard regression analysis

To improve the positioning accuracy and stability of digital hydraulic cylinder more effectively, the influence of pre-opening type and distance of four-way reversing valve, ball-screw lead, inner diameter and non-inertial load on the dynamic characteristics of digital hydraulic cylinder is quantitatively analyzed. The standardized coefficients are obtained by SPSS, as shown in Table 2.

|                          | Steady-state displacement error | Adjusting time | Overshoot |
|--------------------------|--------------------------------|----------------|-----------|
| Pre-opening distance of four-way valve | Beta: -0.983, Sig: <0.001 | Beta: -0.905, Sig: <0.001 | Beta: -0.945, Sig: <0.001 |
| Ball-screw lead          | 0.061                         | 0.009          | 0.046     |
| Inner diameter           | 0.042                         | 0.004          | 0.006     |
| Non-inertial load        | 0.301                         | -0.054         | 0.044     |

Note: Beta denotes Pearson coefficient; Sig. denotes significant coefficient.
For steady-state displacement error, the Pearson coefficient of pre-opening distance of four-way reversing valve is largest, being 0.983 (absolute value). It indicates the pre-opening distance of four-way reversing valve has the largest influence on the positioning accuracy of double closed-loop control digital hydraulic cylinder, followed by the non-inertial load. Both the significant coefficients are less than 0.05, indicating the correlation is significant. The lead of ball screw and the inner diameter of cylinder have little correlation with steady-state error and are not significant. For the dynamic characteristic parameters, the pre-opening distance of the four-way reversing valve is the most relevant, and the Pearson coefficient is more than 0.9. The correlation of other parameters is weak. It can be concluded that guaranteeing the machining and assembly accuracy of the four-way reversing valve is the key to improve the response characteristics of the digital hydraulic cylinder. Under high precision requirements, it is also necessary to compensate the displacement error caused by the non-inertial load.

6. Experimental verification

6.1 Test device

According to the simulation results, the digital hydraulic cylinder with the optimal structural parameters was assembled, as shown in Table 3. And the performance tests were carried out on the self-designed test bench at the State Key Laboratory of High Performance Complex Manufacturing (Central South University, CSU), as shown in Fig. 9. The test bench mainly consists of a hydraulic power unit, digital hydraulic cylinder and data acquisition system. The maximum flow rate and highest pressure of the system is 40L/min and 25MPa, respectively. A computer-based signal input and data acquisition system (including displacement sensor (measuring range: 0-500mm, accuracy: 0.1%F.S.), Danfoss pressure sensor (measuring range: 0-60MPa, accuracy: 0.1%F.S.), NI cDAQ-9112, NI 9474 for pulse output, NI 9203 for current input and output, power adapter) was used to input displacement signal and record the output from the sensors.

| Parameter | Pre-opening distance of four-way valve /mm | Ball-screw lead /mm | Inner diameter /mm | Load /kN | Oil pressure /MPa |
|-----------|------------------------------------------|--------------------|-------------------|--------|------------------|
| Value     | 0.5                                      | 10                 | 90                | 0      | 4                |

Fig. 9 Double closed-loop control digital hydraulic cylinder performance test bench

6.2 Test results

The simulation and experimental comparison results of the dynamic characteristics of the digital hydraulic cylinder during reciprocating motion are shown in Fig. 10. It can be found that the variation trend of the displacement error of the digital hydraulic cylinder during the test is the same as the simulation result. The maximum displacement error is
less than 1.5mm. And the error between simulation and experimental result is about 20%. Besides, the average displacement error of the hydraulic cylinder is different in the two strokes of reciprocating motion. It is the result of the flow gain coefficient of the valve-controlled asymmetric hydraulic cylinder which is related to the moving direction, and the coefficient is smaller when the piston rod is retracted. Thus, the steady-state displacement error is larger which can be inferred from Eq. (6). The error of ratio of displacement error in two strokes between simulation and the test is only 4.2%. It can be seen from Fig. 10 (c) that the velocity characteristic of the hydraulic cylinder in simulation result is better. The velocity shows some fluctuations around the expected value in the test. This may be due to the fact that the simulation conditions are too ideal. And the digital hydraulic cylinder has some errors in the process of machining and assembling, which causes the system damping to be too small. It also results in the fluctuations of the displacement error.

Due to the backlashes in the screw nut pair and the ball screw, the digital hydraulic cylinder has dead zone and hysteresis loop in the reciprocating motion. During the test, the hysteresis error of the hydraulic cylinder is 1.95mm (less than 1%F.S), and the dead zone is 0.31mm (about 0.16%F.S), as shown in Fig. 10 (d). The error between simulation and test are 1.03% and 19.35%, respectively. The simulation results of the positioning accuracy and dynamic characteristics of the digital hydraulic cylinder are basically consistent with the test. Besides, it shows the same change regulations as the mathematical model, which verifies the correctness and accuracy of the simulation model. The results can provide a reference for the design and manufacture of related products.

7. Conclusions

In this study, a high-precision digital hydraulic cylinder that can realize the double closed-loop control of displacement and velocity was proposed. According to the transfer function model, it was determined that the pre-opening distance of four-way valve, ball-screw lead, inner diameter and load were the key parameters affecting the
dynamic response characteristics. The influence of the four factors on the dynamic response characteristics was quantitative analyzed through AMESim model, which was verified by performance tests. The following conclusions can be drawn:

1. As the pre-opening distance of the four-way reversing valve increases, the positioning accuracy and stability of the digital hydraulic cylinder are improved. Absolute values of three Pearson coefficients are all greater than 0.9, which indicates that it is the main factor affecting the dynamic response characteristics of digital hydraulic cylinders.

2. The non-inertial load has a great influence on the positioning accuracy of the digital hydraulic cylinder, and a certain compensation measure is required in the case of high precision requirements. At the same time, the load increases the damping of the system, and the changes of adjusting time and overshoot are small.

3. The ball-screw lead and inner diameter have little effect on the dynamic response characteristics of double closed-loop control digital hydraulic cylinder. The 10mm ball-screw lead and the 90mm inner diameter can meet the positioning accuracy, stability and economy requirements.

4. The errors of displacement error, dead zone and hysteresis loop between simulation and test results are 20.00%, 1.03%, and 19.35% respectively, which verifies correctness and accuracy of the simulation model.

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Appendix

In this appendix, the derivation process of Eq. (5) is described. In order to show the position of parameters more clearly, the digital hydraulic cylinder is represented by a schematic diagram of the valve-controlled asymmetric cylinder since it belongs to a special kind of valve-controlled asymmetric cylinder, as shown in Fig. A1.

![Fig. A1. Schematic diagram of valve-controlled asymmetric cylinder](image)

To meet the power matching requirements of valve-controlled asymmetric cylinder, the load pressure and load flow are redefined as follows (Han et al., 2012; Ylinen et al., 2014):

\[ p_L = p_1 - np_2 \]  \hspace{1cm} (A.1)
\[ q_L = \frac{q_1 + nq_2}{1 + n^2} \quad (A.2) \]

Taking \( \chi_v \geq 0 \) as an example, the Eq. (A.3) and Eq. (A.4) can be obtained according to orifice flow formula.

\[ q_1 = C_d \omega x_v \sqrt{2(p_i - p_1) / \rho} \quad (A.3) \]

\[ q_2 = C_d \omega x_v \sqrt{2(p_2 - p_0) / \rho} \quad (A.4) \]

Where \( C_d \) denotes the flow coefficient of valve port, \( \omega \) denotes the area gradient of valve port, \( p_0 \) denotes the pressure of tank which is regarded as 0MPa, \( \rho \) denotes the hydraulic oil density.

At the steady state, \( \dot{x}_p = \frac{q_1}{A_1} = \frac{q_2}{A_2} \), then,

\[ p_1 = \frac{n^3 p_i + p_L}{n^3 + 1} \quad (A.5) \]

\[ p_2 = \frac{n^2 (p_i - p_L)}{n^3 + 1} \quad (A.6) \]

The results are similar to Eq. (A.5) and Eq. (A.6) at \( \chi_v \leq 0 \), and are not described below. The flow linearization equation of orifice is as follows:

\[ q_L = K_q \chi_v - K_c p_L \quad (A.7) \]

Where \( K_q \) denotes the flow gain coefficient of the valve, \( K_c \) denotes the flow-pressure coefficient of the valve.

Ignoring the effects of pipeline pressure loss and the changes of temperature and bulk modulus, the flow continuity equations of hydraulic cylinder can be obtained.

\[ q_i + nq_2 = (1 + n^2) A_i \dot{x}_p + (1 + n) C_e (p_i - p_2) - n C_e p_2 + \frac{V_i}{\beta_e} \frac{dp_1}{dt} - n \frac{V_2}{\beta_e} \frac{dp_2}{dt} \quad (A.8) \]

Where \( C_e \) denotes the external leakage coefficient, \( C_i \) denotes the internal leakage coefficient, \( V_i \) denotes the equivalent volume of rod-less chamber, \( V_2 \) denotes the equivalent volume of rod chamber.

Assuming that the cylinder performs a slight displacement near the minimum stiffness position of the hydraulic spring, the load flow can be given as (Chen et al., 2014),

\[ q_L = A_1 \dot{x}_p + \frac{(1 + n) C_i (p_i - p_2)}{1 + n^2} - \frac{n C_e p_2}{1 + n^2} + \frac{V_i}{\beta_e} \frac{1}{1 + \sqrt{n}} \frac{dp_1}{dt} - \frac{n^2 \sqrt{n}}{1 + \sqrt{n}} \frac{dp_2}{dt} \quad (A.9) \]

\[ = A_1 \dot{x}_p - C_{i0} p_2 + C_{i0} p + K_i \frac{V_i}{\beta_e} \frac{dp_L}{dt} \]

Where \( C_{i0} = \frac{n^2 [(1 - n^2) C_i + n C_e]}{(1 + n^2)(1 + n^3)}, \quad C_{i1} = \frac{(1 + n^2)(1 + n) C_i + n^3 C_e}{(1 + n^2)(1 + n^3)}, \quad K_i = \frac{1 + n^4 \sqrt{n}}{(1 + \sqrt{n})(1 + n^2)(1 + n^3)} \)

The balance equation of output force and load force of hydraulic cylinder is as follows:
\[ A_1 p_L = A_1 (p_1 - np_2) = A_1 p_1 - A_2 p_2 = m_{r} \frac{d^2 x_p}{dt^2} + B_{p}\frac{dx_p}{dt} + K x_p + F_L \] (A.10)

Under continuous input impulse, the displacement of the spool controlled by stepping motor can be represented as follow:

\[ x_y = \frac{S_1}{2\pi} (\theta_t - \theta_m) = \frac{S_1}{2\pi} \left( \frac{360 \times 2\pi x_p}{360 \times \theta \times S_2} \times \theta - 2\pi x_p \right) = (x - x_p) \frac{S_1}{S_2} \] (A.11)

Where \( \theta_t \) denotes the step angle of stepping motor, \( \theta_m \) denotes the feedback angle, \( \theta_r \) denotes the input angle.

The Eq. (A.7) and Eq. (A.9) can be Laplace transformed to give,

\[ p_L = \frac{K_\gamma x_p - A_1 x_p s - C_m p_s}{K_{sc} + K_{f} \frac{V}{\beta_c} s} \] (A.12)

Combined the Eq. (A.11) and Eq. (A.12) with the Laplace transformed Eq. (A.10), and the Eq. (5) can be obtained.

\[ x_p = \frac{m_{r}V_t K_{sc}}{\beta_c A_1^2} s^3 + \left( \frac{m_{r}K_{sc}}{A_1^2} + \frac{B_{p}V_t K_{sc}}{A_1^2} + \frac{K V_t K_{sc}}{A_1^2} \right) s^2 + \left( 1 + \frac{K V_t}{\beta_c A_1^2} \right) s + \frac{K_{sc} K_{sc}}{A_1^2} + \frac{K_{sc} S_1}{A_1 S_2} \] (A.13)

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