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Design of Position Control Method for Pump-Controlled Hydraulic Presses via Adaptive Integral Robust Control

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Abstract: This paper takes the position control performance of pump-controlled hydraulic presses as the research object. The control methods are designed respectively for the two motion stages of rapid descent and slow descent of hydraulic presses in order to improve the control performance of the system. First of all, the accuracy model of the pump-controlled hydraulic presses position servo system (the pump-controlled hydraulic presses position servo system, which is called PCHPS) and its MATLAB/Simulink simulation platform are established. Based on the theoretical analysis and experimental data, the interference factors affecting the tracking accuracy and positioning accuracy of the PCHPS are analyzed. Then, an adaptive integral robust control (the adaptive integral robust control, which is called AIRC) for PCHPS is designed to reduce the influence of nonlinear factors on the system, and the effectiveness of the controller is verified by simulation. Finally, the position control experiment of PCHPS is designed, and the experimental results show that the AIRC can effectively reduce nonlinear factors such as unknown interference in the slow-down stage of the system. The positioning accuracy is raised to within 0.008 mm, which improves the process level of the hydraulic presses.

Keywords: pump-controlled hydraulic presses; electro-hydraulic servo position control system; adaptive integral robust control; controller design

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1. Introduction

In the hydraulic transmission equipment used in the production of forgings, hydraulic presses [1–4] have been widely used in automobile manufacturing, aerospace, warships, and other industries. However, the lack of advanced control methods not only leads to the decline of hydraulic presses process level and energy utilization but also brings great energy consumption loss [5–7]. It can be seen that improving the process level of hydraulic presses and optimizing the control performance of the pump-controlled hydraulic presses position servo system (the pump-controlled hydraulic presses position servo system, which is called PCHPS) are of great practical significance to promote the growth of production capacity in their application fields and improve economic efficiency.

In the past, many researchers have paid attention to the control problem of the electro-hydraulic system. Due to its simplicity and ease of implementation, the proportional–integral–derivative (PID) control is still widely applied in industry today. However, in order to deal with the nonlinear phenomenon in the system, fuzzy control is an intelligent control method that imitates the human logic thinking and is independent of the accurate mathematical model of the controlled plant [8]. Macchelli A. illustrate the design and the simulative validation of an adaptive controller developed for an hydraulic presses. By using singular perturbation arguments, they show robustness to slowly increasing leakage gains, which is a typical situation in a real-world application [9]. As far as the hydraulic control system [10], the traditional control strategy is often designed and optimized based on the linear framework, usually by linearization processing near the balance point of the

system mathematical model, and by analyzing the system transfer function and configuring the proper and reasonable feedback gain or design correction compensation link to deal with various interferences and parameter perturbation. However, the nonlinear factors prevalent in hydraulic systems cannot be ignored; for example, the non-linearity of differential formula structure in mathematical mechanism modeling, the parts of the model that cannot be modeled accurately, such as load interference, leakage, or the uncertainty of model parameters, changes in load quality, hydraulic elastic modulus due to wear and temperature, etc. [11,12]. The limitations of the traditional linear control strategy cannot cover and deal with these nonlinear factors. In recent years, various nonlinear control methods have been applied in the motion/speed control problem of the electro-hydraulic system. Among them, the adaptive control and the sliding mode control are thought as two effective and alternative methodologies for the nonlinear controller design of the electro-hydraulic system. Xuanjudan proposes an incomplete differential-based improved adaptive backstepping integral sliding mode control to realize the position control for the hydraulic servo system based on the friction compensation. The backstepping-based control integrated with the integral sliding mode surface-based sliding mode control with the friction compensation is used to solve the problem of non-structural uncertainty of the hydraulic system. The incomplete differential is introduced to the adaptive update law by which the low-pass filtering behavior in the incomplete differential is capable of effectively suppressing the interference caused by the pure differential mutation signal [13]. Chong Chee Soon deals with an evaluation on the designed controller named as sliding mode control in which the sliding surface of the controller has been integrated with a proportional–integral–derivative controller. The control scheme is established from the derived dynamic equation, whose stability is proven through Lyapunov theorem [14]. However, the significant drawback of adaptive control is that the transient performance is commonly unknown, and the ability to resist disturbance is normally poor. Although some modifications may be employed to enhance the system robustness, tracking accuracy can no longer be guaranteed. Unlike adaptive control, the SMC can achieve a guaranteed transient performance and final tracking accuracy through robust feedback in the presence of both parametric uncertainties and uncertain nonlinearities. However, the biggest disadvantage of sliding mode control is that it usually involves high-frequency switching in control signals, which may result in control chattering. Although some smoothing techniques may be adopted to reduce the control chattering, the tracking performance is degraded. Therefore, it is of great research value to deal with the nonlinear factors in the pump-controlled system and design the nonlinear control method to improve the control performance of the system.

In this paper, the mathematical mechanism modeling and analysis of PCHPS are studied, and a reasonable simplified mathematical model is applied in the subsequent motion controller design. Aiming at the nonlinear factors such as unknown disturbance and the parameter uncertainty of the hydraulic presses model, an adaptive robust controller is designed based on the simplified mathematical model. Considering that the slow-down and fast-down motion stages have a great impact on the hydraulic presses position control system, a PCHPS position control based on an adaptive integral robust control algorithm is proposed, and the effectiveness of the controller is verified by simulation. Finally, the position control experiment of PCHPS is designed, and the control effect of the designed adaptive integral robust controller is verified and theoretically analyzed. The experimental results show that the adaptive integral robust control algorithm can effectively reduce the influence of nonlinear factors such as unknown interference in the slow-down stage of the system. The influence of nonlinear factors such as unknown interference and parameter uncertainty on the system is reduced, and its position tracking accuracy is improved.

The rest of this paper is organized as follows. The second section describes the problem formulation and dynamic models. The third section is the design and simulation of adaptive robust motion controller. The fourth part is the setting and results of the experiment. Conclusions are drawn in the last section.

2. The Mathematical Model and Analysis of PCHPS

2.1. The Principle of PCHPS

The hydraulic principle of PCHPS is shown in Figure 1. In Figure 1, element 1 is the coarse oil filter, element 2 is the servo motor, element 3 is the hydraulic pump, element 4 is the servo driver, element 5 is the rotary encoder, element 6 is the check valve, element 7 is the fine filter, element 8 is the overflow valve, element 9 is the hydraulic controller, element 10 is the magnetic exchange valve, element 11 is the balance valve, element 12 is the charge valve, element 13 is the pressure transducer, element 14 is the hydraulic cylinder, and element 15 is the displacement transducer.

Figure 1. The principle of PCHPS.

The PCHPS is shown in Figure 1, where the action of the hydraulic cylinder can be divided into the following working stages:

- (1) Fast-next stage: The electromagnetic directional valve 1DT, 3DT, and 4DT are powered, and the hydraulic oil enters the small cross-sectional area of the hydraulic cylinder. The hydraulic cylinder is driven to complete rapid movement with small flow, and it finally returns to the oil tank.
- (2) Slow-down stage: The electromagnetic directional valve 1DT is powered, the hydraulic oil enters the rodless cavity of the hydraulic cylinder to push the hydraulic cylinder to complete the slow motion with a large flow.
- (3) Position-holding stage: The electromagnetic directional valve 1DT is powered, the motor speed will be adjusted in real time according to the position parameters detected by the sensor, so as to realize the position closed-loop control and ensure that the hydraulic cylinder position is in the required position.
- (4) Fast-up stage: The electromagnetic directional valve 2DT and 3DT are powered, and the hydraulic oil pushes the hydraulic cylinder to reset.

2.2. The Mathematical Model of PCHPS

2.2.1. The Mathematical Model of AC Servo Motor

The physical model of the surface-mounted AC servo motor [15] is shown in Figure 2.

Figure 2. AC servo motor physical model of SPMSM in a plane coordinate system.

In order to simplify the mathematical model, the static stator three-phase winding of the AC servo motor can be equivalent to the shaft two-phase winding, which rotates synchronously with the rotor and is relatively static. At the same time, ignore the influence of the shaft voltage component of coordinate axis q , and the simplified voltage formula is shown as follows:

$$\begin{cases} u_d = Ri_d + L_d \frac{di_d}{dt} - \omega_e L_q i_q \\ u_q = Ri_q + L_q \frac{di_q}{dt} + \omega_e \psi_f \end{cases} \quad (1)$$

where, u_d is the d -axis component of stator voltage; u_q is the q -axis component of stator voltage; i_d is d -axis components of stator current, L_d is the stator d -axis equivalent inductance; L_q is the stator q -axis equivalent inductance, i_q is q -axis components of the stator current, ω_e is the electrical angular velocity, and R is the equivalent resistance of stator winding.

The AC servo motor selected in this paper is a surface-mounted three-phase synchronous motor, so the equivalent values of the inductance of the stator winding are equal in the $d - q$ axis. So, the electromagnetic torque formula is shown as follows:

$$T_e = \frac{3}{2} p_n i_q \psi_f \quad (2)$$

where T_e is the electromagnetic torque and the p_n is the number of magnetic poles of motor.

Laplace transform of Formulas (1) and (2) is shown below.

$$\begin{cases} \omega(s) = \frac{1}{Js + B_r} [T_e(s) - T_L(s)] \\ T_e(s) = \frac{3}{2} p_n \psi_f I_q(s) \\ I_q(s) = \frac{U_q(s) - \psi_f \omega(s)}{L_q s + R} \end{cases} \quad (3)$$

Let $G_n(s) = k_{np} + \frac{k_{ni}}{s}$ and $G_i(s) = k_{ip} + \frac{k_{ii}}{s}$ denote the PI controller of the motor speed loop and current loop.

The transmission control block diagram of the AC servo motor is obtained, as shown in Figure 3.

Figure 3. Diagram of AC servo motor.

2.2.2. The Mathematical Model of the Pump-Controlled Cylinder System

The hydraulic pump studied in this paper is an inclined plate axial piston pump [16–18]. Assuming that at a certain time, the number of pistons at the hydraulic position is n , then the instantaneous flow of the piston pump is shown as follows:

$$\begin{aligned} Q &= Q_1 + Q_2 + \dots + Q_n = \sum_{i=1}^n Q_i \\ &= AR\omega \tan \gamma \sum_{i=1}^n \sin \psi_i \end{aligned} \quad (4)$$

where A is the cross-sectional area of the plunger, R is the single piston radius, and w is the single plunger angular velocity.

It can be seen from Formula (4) that the axial velocity of the piston of the swashplate piston pump changes according to the sine law, and the flow discharged by multiple pistons will superimpose a relatively uniform flow. Therefore, the displacement formula of the piston pump can be shown as follows:

$$V = \frac{\pi d^2}{2} z R \tan \gamma. \quad (5)$$

The output flow of the variable pump is shown as follows:

$$q_p = V\omega_p - C_p(p_1 - p_2) \quad (6)$$

where V is the displacement of the piston pump, d is the piston diameter, z is the number of plungers, R is the distribution circle radius of each plunger, γ is the swash plate angle, and C_p is the Leakage coefficient of piston pump.

The flow continuity formulas of the hydraulic cylinder are established below. The continuous flow formula of a rodless cavity is shown as follows:

$$q_1 = A_1 \frac{dx_p}{dt} + C_{ic}(p_1 - p_2) + C_{ec}p_1 + \frac{V_{01}}{\beta_e} \frac{dp_1}{dt}. \quad (7)$$

The continuous formula of rod cavity flow is shown as follows:

$$q_2 = -A_2 \frac{dx_p}{dt} + C_{ic}(p_1 - p_2) - C_{ec}p_1 - \frac{V_{02}}{\beta_e} \frac{dp_2}{dt}. \quad (8)$$

It is assumed that the initial volumes of the two cavities are equal, that is $V_t = V_{01} = V_{02}$, the average flow is taken as the load flow.

$$q_L = \frac{(q_1 + q_2)}{2} \quad (9)$$

$$q_L = \frac{(A_1 + A_2)}{2} \frac{dx_p}{dt} + \frac{V_0}{\beta_e} \left(\frac{dp_1}{dt} - \frac{dp_2}{dt} \right) + \left(\frac{C_{ec}}{2} + C_{ic} \right) p_L \quad (10)$$

where C_{ic} is the leakage coefficient in the hydraulic cylinder, C_{ec} is the leakage coefficient outside the hydraulic cylinder, A_1 is the cross-sectional area of the rodless cavity piston rod in the hydraulic cylinder, A_2 is the hydraulic cylinder rod cavity piston rod cross-sectional area, V_{01} is the initial volume of the rodless cavity of the hydraulic cylinder, V_{02} is the initial volume of the hydraulic cylinder with the rod cavity, q_L is the load flow, q_1 is the rodless cavity flow, q_2 is the rod cavity flow, and β_e is the effective bulk modulus.

According to Newton's second law, the formula of balancing forces for the hydraulic cylinder and load is shown as follows:

$$F_c = A_p(p_1 - p_2) = m_t \frac{d^2 x_p}{dt^2} + B_p \frac{dx_p}{dt} + kx_p + F_L \quad (11)$$

where m_t is the total mass of the piston and load converted to piston, B_p is the viscous damping coefficient, k is the load spring stiffness, F_L is the load force, C_{tc} is the total leakage coefficient, and A_p is the effective area of hydraulic cylinder at work.

Laplace transform of the above formula has the following transfer function:

$$\begin{cases} Q_p(s) = D_p \omega_p(s) - C_{tp} P_1(s) \\ Q_p(s) = C_{tc} P_1(s) + A_p X_p(s) + \frac{V_0}{\beta_e} s P_1(s) \\ F_g(s) = A_p P_1(s) = m_t s^2 X_p(s) + B_p s X_p(s) + k X_p(s) + F_L(s) \end{cases} \quad (12)$$

According to the above formulas, the pump-controlled cylinder transfer block diagram is shown in Figure 4.

Figure 4. Pump-controlled cylinder transfer block diagram.

In summary, the PCHPS transfer block diagram is shown in Figure 5.

Figure 5. Electro-hydraulic servo position control system of PCHPS.

3. Adaptive Integration Robust Control Algorithm

3.1. Adaptive Integration Robust Controller

3.1.1. System State Space Expression

In order to make the designed controller simple and stable, the transfer function of the AC servo motor is simplified as a proportional link, and its dynamic characteristics and

unmodeled parts can be compensated by robust control. According to the mathematical model of the PCHPS, the status variables are selected as follows:

$$x = [x_1, x_2, x_3]^T \triangleq [y, \dot{y}, A_1 P_1 / m_t - A_2 P_2 / m_t]^T. \quad (13)$$

The state-space expression of the PCHPS is shown as follows:

$$\begin{aligned} \dot{x}_1 &= x_2 \\ \dot{x}_2 &= x_3 - b x_2 + f_0 + d(t, x_1, x_2) \\ \dot{x}_3 &= \theta_1 f_1 u - \theta_2 f_2 - \theta_3 f_3 \end{aligned} \quad (14)$$

where $\theta_1 = D_{pe} \beta_e$, $\theta_2 = \beta_e$, and $\theta_3 = C_i \beta_e$, which makes $f_0 = \frac{f}{m_t}(\sigma_0 z + \sigma_1 \dot{z} + \sigma_2 v)$, $f_1 = \frac{1}{m_t} \left(\frac{A_1}{V_1} + \frac{A_2}{V_2} \right)$, $f_2 = \frac{1}{m_t} \left(\frac{A_1^2}{V_1} + \frac{A_2^2}{V_2} \right) x_2$, $f_3 = \frac{P_L}{m_t} \left(\frac{A_1}{V_1} + \frac{A_2}{V_2} \right)$, and $b = \frac{B}{m_t}$, in which C_i is the leakage coefficient of the hydraulic cylinder, $d(t, x_1, x_2)$ is the centralized nominal function of system unmodeled dynamics, and D_{pe} is the equivalent displacement.

For the unknown parameter θ , assume that the parameter is bounded and satisfied

$$\theta \in \Omega_\theta = \{\theta : \theta_{\min} \leq \theta \leq \theta_{\max}\} \quad (15)$$

where $\theta_{\min} = [\theta_{1\min}, \theta_{2\min}, \theta_{3\min}]$, $\theta_{\max} = [\theta_{1\max}, \theta_{2\max}, \theta_{3\max}]$, and $\theta_{\min}$ and $\theta_{\max}$ are known. $d(t, x_1, x_2)$ is considered smooth enough to be satisfying

$$\left| \dot{d} \right| \leq \delta_1 \quad \left| \ddot{d} \right| \leq \delta_2.$$

3.1.2. Controller Design

Firstly, based on the definition of the projective function, the parameter adaptive law is designed; let $\hat{\theta}$ denote the estimate of unknown parameters θ , $\hat{\theta}$ representing the estimation error of unknown parameter vectors θ , that is, $\theta = \hat{\theta} - \theta$, which is shown as follows:

$$\dot{\hat{\theta}} = Proj_{\hat{\theta}}(\Gamma \tau)$$

where Γ is the parameter adaptive rate diagonal matrix, and $Proj_{\hat{\theta}}$ is a discontinuous insinuation function, which can be shown as follows:

$$Proj_{\hat{\theta}_i}(\tau_i) = \begin{cases} 0, & \text{if } \hat{\theta}_i = \theta_{i\max} \text{ and } \tau_i > 0 \\ 0, & \text{if } \hat{\theta}_i = \theta_{i\min} \text{ and } \tau_i < 0 \\ \tau_i, & \text{otherwise} \end{cases}. \quad (16)$$

For arbitrary τ , the projective function satisfies the following two properties

$$\begin{aligned} 1. & \hat{\theta} \in \Omega_{\hat{\theta}} \triangleq \{\hat{\theta} : \theta_{\min} \leq \hat{\theta} \leq \theta_{\max}\} \\ 2. & \tilde{\theta}^T [\Gamma^{-1} Proj_{\hat{\theta}}(\Gamma \tau) - \tau] \leq 0, \forall \tau \end{aligned}$$

Since the system is not modeling dynamics and external interference exists, the controller is designed by an anti-stepping method. According to Formula (14), the first formula does not contain any uncertainty. Set x_d to the input track, which defines the following error variables.

$$\begin{aligned} e_1 &= x_1 - x_d \\ e_2 &= \dot{e}_1 + k_1 e_1 = x_2 - x_{2eq}, \quad x_{2eq} \triangleq \dot{e}_{1d} - k_1 e_1 \end{aligned} \quad (17)$$

where e_1 is the tracking error, k_1 is any positive feedback gain, the $G_{e1}(s) = \frac{e_1(s)}{e_2(s)} = \frac{1}{s+k_1}$ is the controller representing the error ring, wherein, when e_1 becomes small enough or converges to zero, e_2 also becomes small enough or converges to zero. So, the following

design goal is to make e_2 as small as possible while ensuring stable transient performance. Additional controller design freedom is obtained by designing auxiliary error signals r .

$$\begin{aligned} r &= x_3 - \dot{x}_{2eq} + k_2 e_2 - b x_2 + f_0 + d(t, x_1, x_2) \\ &= e_3 + \alpha_2 - \dot{x}_{2eq} + k_2 e_2 - b x_2 + f_0 + d(t, x_1, x_2) \end{aligned} \quad (18)$$

where α_2 is the virtual control law of the state variable x_3 , and e_3 is the deviation between the two.

For virtual control law α_2 , its structure and expression are described as follows

$$\begin{aligned} \alpha_2 &= \alpha_{2a} + \alpha_{2s} \\ \alpha_{2a} &= \dot{x}_{2eq} - k_2 e_2 + b x_2 - \hat{f}_0 \\ \alpha_{2s} &= \alpha_{2s1} + \alpha_{2s2}, \quad \alpha_{2s1} = -k_r z_2 \end{aligned} \quad (19)$$

where α_{2a} is model-based feedback control law, and α_{2s} is the robust control part. The α_{2s1} is designed as a linear robust feedback control law, which is used to stabilize the nominal model of the hydraulic system.

The above formulas can be obtained:

$$r = e_3 - k_r e_2 + \alpha_{2s2} + d(t, x_1, x_2). \quad (20)$$

In order to overcome the effect of modeling uncertainty and unknown external interference on the system, the error symbol integral robust control item is introduced here. Therefore, α_{2s2} can be expressed as follows:

$$\alpha_{2s2} = - \int_0^t k_r k_2 e_2 + \beta \text{sign}(e_2) dv \quad (21)$$

where β is the integral robust feedback gain; $\text{sign}(e_2)$ is a standard symbol function about errors e_2 .

In order to obtain the actual control input of the system, the following face (20) is guided

$$\begin{aligned} \dot{r} &= \dot{x}_3 - k_r \dot{z}_2 + \dot{\alpha}_{2s2} + \dot{d} \\ &= \dot{x}_3 - \dot{\alpha}_2 - k_r r - \beta \text{sign}(z_2) + \dot{d} \end{aligned} \quad (22)$$

The joint Formulas (20) and (22) are available

$$\dot{r} = \theta_1 f_1 u - \theta_2 f_2 - \theta_3 f_3 - \dot{\alpha}_2 - k_r r - \beta \text{sign}(z_2) + \dot{d}.$$

Due to the uncertainty of parameters, the actual input of the system consists of estimates of unknown parameters.

$$\begin{aligned} u &= u_a + u_s \\ u_a &= \frac{\hat{\theta}_2 f_2 + \hat{\theta}_3 f_3 + \dot{\alpha}_2}{\hat{\theta}_1 f_1} \\ u_s &= \frac{-k_3 z_3}{f_1} \end{aligned} \quad (23)$$

In the upper, k_3 is the positive feedback gain, u_a is an adjustable model compensation control by updating parameters online, and u_s is a robust control. So, Formulas (22) and (23) are available,

$$\begin{aligned} \dot{r} &= -\varphi^T \tilde{\theta} - k_3 \theta_1 e_3 - k_r r - \beta \text{sign}(e_2) + \dot{d} \\ \varphi &= [f_1 u_a, -f_2, -f_3]^T. \end{aligned} \quad (24)$$

φ is the regression vector of the parameter adaptation, and the e_3 's dynamic characteristics are shown as follows:

$$\dot{e}_3 = -\varphi^T \tilde{\theta} - k_3 \theta_1 e_3. \quad (25)$$

The performance of the controller is analyzed below.

Define the auxiliary function $L(t) = r[\dot{d} - \beta \text{sign}(e_2)]$. When the robust gain β satisfies the inequality $\beta \geq \delta_1 + \frac{1}{k_2} \delta_2$, the auxiliary function $P(t) = \beta|e_2(0)| - e_2(0)\dot{d}(0) - \int_0^t L(v)dv$ is constantly positive. The adaptive function selected by the controller is $\tau = \varphi(r + e_3)$.

Define the feedback gain matrix Λ , and the value of the feedback gain k_1, k_2, k_3, k_r is large enough to make the gain matrix Λ a positive matrix.

$$\Lambda = \begin{bmatrix} k_1 & -1/2 & 0 & 0 \\ -1/2 & k_2 & -1/2 & 0 \\ 0 & -1/2 & k_r & \theta_1 k_3/2 \\ 0 & 0 & \theta_1 k_3/2 & \theta_1 k_3 \end{bmatrix} \quad (26)$$

Select the Lyapunov function $V = \frac{1}{2}(e^T e + \tilde{\theta}^T \Gamma^{-1} \tilde{\theta}) + P(t)$; therefore,

$$\dot{V} = -e^T \Lambda e + \tilde{\theta}^T \Gamma^{-1} (\dot{\tilde{\theta}} - \Gamma \tau). \quad (27)$$

Depending on the nature of the innuendo function, it can be seen that $\dot{V} \leq -e^T \Lambda e$; since the gain matrix Λ is a positive matrix, there is

$$\dot{V} \leq -\lambda_{\min}(\Lambda)(e_1^2 + e_2^2 + r^2 + e_3^2) \triangleq -W \quad (28)$$

$-\lambda_{\min}(\Lambda)$ is the minimum characteristic value of the gain matrix.

Then, when the numerical selection of the relevant control parameters meets the above conditions, all the signals of the system are bounded under closed-loop control, and the control system also obtains the progressive output tracking performance of the signal, that is, when $t \rightarrow \infty$, $e_1 \rightarrow 0$.

3.2. The Dynamic Simulation of PCHPS

In this section, the adaptive integral robust control algorithm designed above is verified by MATLAB/Simulink simulation platform. At the same time, the dead zone and some nonlinear disturbances are added to the PCHPS to make the system simulation characteristics closer to the reality. The values of relevant control parameters of an adaptive integral robust controller are as follows.

$k_1 = 1200, k_2 = 200, k_3 = 50, k_r = 300$ — Controller gain;

$\hat{\theta}_1(0) = 26990, \hat{\theta}_2(0) = 1.24 \times 10^9, \hat{\theta}_3(0) = 0.005$ — The initial values of the adaptive parameters;

$\Gamma = \text{diag}\{6 \times 10^{-5}, 1.8 \times 10^9, 1 \times 10^{-10}\}$ — Parameter adaptive gain;

$\beta = 300$ — Robust control gain.

The track planning table for verifying the AIRC is shown in Table 1.

Table 1. Motion track planning table.

Conditions	Acceleration (mm/s ³)	Max Acceleration (mm/s ²)	Max Speed (mm/s)	Travel (mm)
①	5	5	5	20
②	5	5	5	35
③	5	5	10	35
④	10	10	10	35

Figures 6–9 are the performance of the slow motion phase of the PCHPS from simulation operating conditions (1) to (4).

Figure 6. System displacement and deviation curve under working condition ①. (a) System displacement curve. (b) System displacement deviation curve.

Figure 7. System displacement and deviation curve under working condition ②. (a) System displacement curve. (b) System displacement deviation curve.

Figure 8. System displacement and deviation curve under working condition ③. (a) System displacement curve. (b) System displacement deviation curve.

Figure 9. System displacement and deviation curve under working condition ④. (a) System displacement curve. (b) System displacement deviation curve.

It can be seen from Figures 6–9 that after adding a dead zone and nonlinear interference to the simulation model, the tracking performance and positioning accuracy of the PCHPS under the PID algorithm cannot show an ideal control performance. When the position control algorithm of the system is changed to the AIRC algorithm, the motion trajectory of the system overcomes the original lag in a short time, and it starts to track the planned trajectory of the system stably, with the deviation kept within the range of 0 to 0.2 mm. When the system runs to the specified position, there is almost no over range, and the positioning accuracy is within 0.005 mm; thus, the control effect is very ideal. The simulation results show that the designed AIRC algorithm greatly improves the positioning accuracy and improves the position control performance of the PCHPS.

4. Experimental Verification of PCHPS

4.1. Experimental Test Platform of PCHPS

In order to verify the effectiveness of the friction compensation and control algorithm of the open PCHPS, Figure 10 is an industrial open PCHPS made of a Figure 1 schematic.

Figure 10. Open pump-controlled hydraulic electromechanical fluid servo position control system.

4.2. Experimental Test of PCHPS

The slow-down stage of the system is running at a low speed and is very susceptible to unknown interference from the system and the outside, so the motion stage uses an AIRC controller with the same controller parameter settings as the 3.2 simulation parameters; see Figures 11–14. The figure shows the position control performance of the system after adjustment with the conventional PID algorithm after adjusting the AIRC algorithm.

Figure 11. System displacement and deviation curve under working condition ①. (a) System displacement curve. (b) System displacement deviation curve.

Figure 12. System displacement and deviation curve under working condition ②. (a) System displacement curve. (b) System displacement deviation curve.

Figure 13. System displacement and deviation curve under working condition ③. (a) System displacement curve. (b) System displacement deviation curve.

Figure 14. System displacement and deviation curve under working condition ④. (a) System displacement curve. (b) System displacement deviation curve.

It can be seen that in the initial stage of slow-down motion of the hydraulic cylinder, the phenomenon in which the actual displacement is ahead of the expected displacement after a period of motion has been greatly improved. The actual displacement curve of the system does not change greatly after gradually approaching the trajectory curve of the motion planning, showing good position tracking performance and fully overcoming the positioning error in the fast-down stage. Through the displacement deviation curves under various working conditions, it can be found that the variation of system deviation under the AIRC algorithm is gentler than that under the PID algorithm, and the deviation values are significantly smaller than those under the PID algorithm. When the system moves slowly to the deceleration stage of the planned trajectory, the deviation of the system gradually approaches zero. This can fully ensure that the system avoids the occurrence of over-range phenomenon to the greatest extent, the final positioning accuracy is significantly improved, and the difference is finally stabilized at about ± 0.005 mm. It can be seen that the AIRC algorithm fully overcomes the negative impact of unknown disturbances and parameter changes in the slow stage of the system, and it gives the appropriate control compensation for the system position.

5. Discussion

5.1. The Research Background and Significance of Hydraulic Press

The pump-controlled hydraulic presses adopts the volume control mode. Compared with the valve-controlled hydraulic system, the heating of the pump-controlled hydraulic system is reduced and the energy utilization rate can be increased by more than 40%. In addition, the pump-controlled system has a compact structure and is convenient for installation and maintenance. The nonlinear factors of hydraulic system affect and restrict the control accuracy and quality of the system to a great extent. The traditional linearization control strategy cannot cover and deal with these nonlinear factors, and its limitations can be seen here. Therefore, it is of great research value to deal with the nonlinear factors in the pump-controlled system and design the nonlinear control method to improve the control performance of the system.

According to the system characteristics of a pump-controlled hydraulic press, this paper designs a reasonable nonlinear control method and tries to compensate for the nonlinear friction phenomenon in the system, in order to obtain a higher precision system control performance and lay a good theoretical research foundation for improving the control performance of PCHPS.

5.2. The Selection of Control Strategy

There are many nonlinear links and links that cannot establish an accurate model in the PCHPS. In many studies, nonlinear control methods are introduced into the compensation control of nonlinear factors of an electro-hydraulic servo system in order to obtain a better control effect. Bin Yao of Purdue University proposed a general adaptive robust controller design method. The robust control part is used to reduce the influence of the unknown modeling part on system stability [19,20], and the parameter adaptation is used to reduce the uncertainty of model parameters. Bin Yao has carried out a detailed mathematical derivation and demonstration on the controller structure and proved that the controller can overcome nonlinear factors and be asymptotically stable.

It is proved by investigation and actual situation that the adaptive integral robust control method can obviously suppress the interference in the system. Especially for the unmodeled part of the mathematical model, there is inaccuracy in the parameters and uncertain external interference in the modeled part. The method used in this study can reduce the adverse impact of the above interference on the system.

6. Conclusions

In this paper, PCHPS is taken as the research object. Through simulation and experimental comparison, the position control performance of the system in the fast and slow motion stages of the conventional control algorithm is analyzed. The control algorithms are designed for the main factors affecting the position control accuracy in each motion stage, and the position compensation control of the pump-controlled hydraulic press is analyzed. The mathematical mechanism models of a permanent magnet synchronous AC servo motor and pump-controlled cylinder are established respectively, and the simulation model of a PCHPS machine is built by MATLAB/Simulink software. An AIRC algorithm for the PCHPS is proposed to compensate for the unknown interference, parameter variation, and model inaccuracy in the system. Simulation and experiments show that the AIRC algorithm can effectively overcome the nonlinear interference of the system, improve the position control effect of the system, and greatly improve the positioning accuracy of the hydraulic press.

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Abbreviations

The following abbreviations and symbols are used in this manuscript:

PCHPS	pump-controlled hydraulic position servo system
AIRC	adaptive integral robust control
u_d	d-axis component of stator voltage
u_q	q-axis component of stator voltage
i_d	d-axis components of stator current
L_d	stator d-axis equivalent inductance
L_q	the stator q-axis equivalent inductance
i_q	q-axis components of stator current
ω_e	electrical angular velocity
R	equivalent resistance of stator winding
T_e	electromagnetic torque
p_n	number of magnetic poles of motor
A	cross-sectional area of plunger
R	the single piston radius
w	single plunger angular velocity
V	displacement of piston pump
d	piston diameter
z	number of plunger
R	distribution circle radius of plunger
γ	swash plate angle
C_p	leakage coefficient of piston pump
C_{ic}	leakage coefficient in the hydraulic cylinder
C_{ec}	leakage coefficient outside the hydraulic cylinder
A_1	cross-sectional area of the rodless cavity piston rod in the hydraulic cylinder
A_2	hydraulic cylinder rod cavity piston rod cross-sectional area
V_{01}	Initial volume of rodless cavity of hydraulic cylinder
V_{02}	Initial volume of hydraulic cylinder with rod cavity
q_L	load flow
q_1	rodless cavity flow
q_2	rod cavity flow
β_e	effective bulk modulus
m_t	total mass of piston and load converted to piston
B_p	viscous damping coefficient
k	load spring stiffness
F_L	load force
C_{tc}	total leakage coefficient
A_p	effective area of hydraulic cylinder at work
C_i	leakage coefficient of hydraulic cylinder
$d(t, x_1, x_2)$	centralized nominal function of system unmodeled dynamics
D_{pe}	equivalent displacement
Γ	parameter adaptive rate diagonal matrix
$Proj_{\hat{\theta}}$	discontinuous insinuation function
e_1	tracking error
k_1	any positive feedback gain
α_2	virtual control law of the state variable x_3
e_3	deviation between the two
α_{2a}	model-based feedback control law
α_{2s}	robust control part
β	integral robust feedback gain
$\text{sign}(e_2)$	standard symbol function about errors e_2 .
k_3	positive feedback gain
u_a	adjustable model compensation control by updating parameters online
u_s	robust control
φ	regression vector of parameter adaptation

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