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Adaptive Sliding Mode Control of a Nonlinear Electro-Hydraulic Servo System for Position Tracking

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1. Introduction

The electro-hydraulic servo system (EHSS) has been widely employed in the field of engineering due to prominent advantages such as fast dynamic response, small size, large force/torque output and high reliability [1, 2]. However, it is typically non-linear with various nonlinear nature of hydraulic dynamics due to the nonlinear pressure-flow characteristics of servo valve, nonlinear friction and nonlinear dynamics of pressure. Apart from dynamical characteristics of essential nonlinear, the EHSS also has other drawbacks such as parametric uncertainties and external disturbances that caused by surrounding environment [3]. The above-mentioned shortcomings limit the development of high performance controller for conventional valve-controlled hydraulic system. Hence, how to weaken the influences of uncertain nonlinearities and ensure the high tracking accuracy of the EHSS in practical implementations has always been the focus of attention.

Due to their simplicity, the PID controller and the input/output linearizing controller [4-6] are commonly used for position tracking of the EHSS in the situations of low control response and precision. However, linear controllers used in the nonlinear uncertain system may degrade the system performance and robustness [7]. Therefore, in order to obtain higher performance, selecting the appropriate controllers for nonlinear uncertain systems is an important aspect in the field of EHSS. Compared with the traditional linear control methods, non-linear control methods and intelligent approaches demonstrate better sensitivity to both the non-linearity and uncertainty of electro-hydraulic system. Sliding mode control (SMC) [8-10], fuzzy control [11, 12], adaptive control [4, 13-16] and neural network control [17, 18] have been successfully utilized in EHSS to meet practical and different control demands, but these methods still have their own limitations. Adaptive control is an effective method to deal with control problems coming from uncertain nonlinear system, especially the system uncertainty caused by uncertain parameters [19, 20]. However, due to the weak robustness caused by external disturbance, adaptive control is difficult to meet the high stability requirements. With the merits of robustness to parameter variations and insensitivity to disturbances, sliding mode control brings new ideas for nonlinear uncertain systems. Nevertheless, its inherent chattering phenomenon will take place when the system operates near the sliding surface [13]. Although both of the fuzzy control strategy and the neural network can approximate any nonlinear function to a desired accuracy, they may not have enough reliability on account that the former depends seriously on expert experiences and the latter may resulting the “explosion of complexity” problem [21].

From previous findings and discussion, it is reasonable to combine adaptive control techniques and the SMC to treat system parameter uncertainties and external disturbances for nonlinear uncertain systems. Obviously, adaptive sliding mode control (ASMC) is a preferred option with simple control structure, strong adaptability and robustness to parameter variations and external disturbances. The purpose in this paper is to provide a developed adaptive sliding mode controller to realize the high-performance tracking control of EHSS with model uncertainty. In particular, this scheme combines the sliding model control based on modified exponent reaching law with adaptive algorithm. Both the parametric adaptive estimation law and discontinuous projection algorithm are designed to estimate unknown parameters of the subject, which can effectively overcome the influence caused by the parameter uncertainty. The sliding surface with variable gain coefficient is presented for the switching control part of the sliding mode, which combines the idea of saturation function instead of the sign function to eliminate the chattering phenomenon. At last, the comparative experiment is performed to illustrate the control performance of the proposed ASMC strategy.

In general, this paper is organized as follows. Section 2 gives the detailed dynamic models and problem formulation of the EHSS. The controller combines the sliding model control based on modified exponent reaching law with adaptive algorithm is designed and the related stability analysis is given in Section 3. Then the comparative experimental results are presented in Section 4. Finally, the conclusion is drawn in Section 5.

2. Dynamic models of hydraulic servo system

As depicted in Fig. 1, the electro-hydraulic servo drive cylinder mainly consists of a single-rod cylinder, a three-position four-way servo valve, and an end-effector with external disturbances. Parameters in the system can be defined as follows: A_1 and A_2 are the effective working area at both sides of the piston, p_1 and p_2 denote the pressures in the cylinder forward and return chamber, respectively, q_1 and q_2 are the liquid flow of the two actuator chambers, respectively. Moreover, m and x_p represent the mass and displacement of the end-effector, respectively.

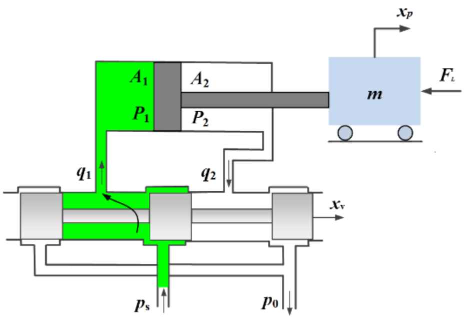

Fig. 1 Configurations of the electro-hydraulic servo drive cylinder

In order to make the end-effector as closely as possible to track the specified motion trajectory, a mathematical model that represents the hydraulic system dynamics is presented as follows.

The dynamics equation of the force balance can be given by:

$$A_1 p_1 - A_2 p_2 = F_L + kx_p + B_t \dot{x}_p + m\ddot{x}_p, \tag{1}$$

where: F_L is the external disturbance force, k denotes elastic damping in cylinder and B_t represents the viscous damping coefficient caused by hysteretic force-velocity characteristics of the shock absorber.

Define function as $\mathrm{sg}(*) = \begin{cases} 1, & \text{if } * > 0 \\ -1, & \text{if } * < 0 \end{cases}$, and q_1 and q_2 can be modeled by:

$$\begin{cases} q_1 = C_d x_v w\sqrt{\left[p_s + \mathrm{sg}(x_v)(p_s - 2p_1)\right]/\rho} \\ q_2 = C_d x_v w\sqrt{\left[p_s + \mathrm{sg}(x_v)(2p_2 - p_s)\right]/\rho} \end{cases}, \tag{2}$$

where: C_d is the discharge coefficient, x_v is the servo valve spool displacement, w is the spool valve area gradient and ρ is the fluid density.

In fact, the natural frequency of the servo valve in the experiment is much higher than that of the hydraulic cylinder. On the premise of not significantly reduce the model accuracy, the dynamics of the servo valve can be simplicity approximation by [16]:

$$x_v = K_v u, \tag{3}$$

where: K_v is a positive constant and u is the control input voltage. Hence, Eq. (2) can be transformed to:

$$\begin{cases} q_1 = C_d K_v uw\sqrt{\left[p_s + \mathrm{sg}(u)(p_s - 2p_1)\right]/\rho} \\ q_2 = C_d K_v uw\sqrt{\left[p_s + \mathrm{sg}(u)(2p_2 - p_s)\right]/\rho} \end{cases}. \tag{4}$$

Neglecting the influence of the fluid temperature and the bulk modulus of elasticity in two chambers, the dynamic equilibrium equations of influent and effluent flow can be expressed as:

$$\begin{cases} q_1 - C_i(p_1 - p_2) - C_e p_1 = A_1\dot{x}_p + \dfrac{(V_{10} + A_1 x_p)\dot{p}_1}{\beta_e} \\ C_i(p_1 - p_2) - C_e p_2 - q_2 = A_2\dot{x}_p + \dfrac{(V_{20} - A_2 x_p)\dot{p}_2}{\beta_e} \end{cases}, \tag{5}$$

where: V_{i0} (i=1, 2) is the initial control volume of the fluid between the valve and the piston; β_e is the bulk modulus of the fluid; C_i and C_e are the coefficients of the internal leakage and the external leakage of the cylinder, respectively.

Since the leakage flow is as insignificant as the fluid compression caused by β_e, from Eq. (5) we use the following approximation:

$$q_2 / q_1 \approx A_2 / A_1 = \varepsilon, \tag{6}$$

where: ε denotes the piston area ratio.

In addition, the load pressure p_L and the load flow q_L of the system are defined as:

$$\begin{cases} p_L = p_1 - \varepsilon p_2 \\ q_L = \dfrac{q_1 + \varepsilon q_2}{1 + \varepsilon^2} = q_1 \end{cases}. \tag{7}$$

When the piston moves slightly at the origin of the displacement, the displacement of the end-effector is close to zero. Taking the Eqs. (5-7) into account, the load pressure q_L is rewritten as:

$$q_L = A_1\dot{x}_p + \lambda_1\dot{p}_L + \lambda_2 p_L + \lambda_3 p_s, \tag{8}$$

where:

$$\begin{cases} \lambda_1 = \dfrac{V_{10}V_{20}}{\beta_e\left(\varepsilon^2 V_{10} + V_{20}\right)} \\ \lambda_2 = \dfrac{\left(\varepsilon^3 V_{10} + V_{20}\right)(C_i + C_e) + \varepsilon\left(V_{10} + \varepsilon V_{20}\right)C_i}{\left(1+\varepsilon^3\right)\left(\varepsilon^2 V_{10} + V_{20}\right)} \\ \lambda_3 = \dfrac{\varepsilon\left(V_{20} - V_{10}\right)(C_i + C_e) - \left(V_{20} - \varepsilon^2 V_{10}\right)C_i}{\left(1+\varepsilon^3\right)\left(\varepsilon^2 V_{10} + V_{20}\right)}\varepsilon^{1+\mathrm{sg}(u)} \end{cases}. \tag{9}$$

Taking the starting point of piston rod as its coordinate origin, and from Eq. (2) and Eq. (3), the load pressure q_L can be linearized as:

$$q_L = \frac{\partial q_L}{\partial x_v}x_v + \frac{\partial q_L}{\partial p_L}p_L = K_q K_v u - K_c p_L, \tag{10}$$

where: K_q and K_c are the flow gain coefficient and flow-pressure coefficient of the servo valve, respectively. Eq. (10) can be described by:

$$\begin{cases} K_q = C_d w\sqrt{\dfrac{(1+\varepsilon)p_s + \mathrm{sg}(u)\left((1-\varepsilon)p_s - 2p_L\right)}{\rho(1+\varepsilon^3)}}, \\ K_c = \dfrac{C_d K_v wu}{\sqrt{\rho(1+\varepsilon^3)\left((1+\varepsilon)p_s + \mathrm{sg}(u)\left((1-\varepsilon)p_s - 2p_L\right)\right)}}. \end{cases}$$

Finally, define the system state variables as $[x_1, x_2, x_3]^T=[x_p, \dot{x}_p, \ddot{x}_p]^T$. Combing Eq. (1) with Eqs. (6) and (7), a mathematical model of the entire system can be formulated by:

$$\begin{cases} \dot{x}_1 = x_2 \\ \dot{x}_2 = x_3 \\ \dot{x}_3 = (-\varphi_1 x_1 - \varphi_2 x_2 - \varphi_3 x_3 + u - d) / \beta \end{cases}, \tag{11}$$

where: u is the input of the system, and:

$$\begin{cases} \varphi_1 = \dfrac{k(\lambda_2 + K_c)}{A_1 K_q K_v} \\ \varphi_2 = \dfrac{A_1^2 + B_t(\lambda_2 + K_c) + k\lambda_1}{A_1 K_q K_v} \\ \varphi_3 = \dfrac{\lambda_1 B_t + (\lambda_2 + K_c)m}{A_1 K_q K_v} \\ \beta = \dfrac{\lambda_1 m}{A_1 K_q K_v} \\ d = \dfrac{\lambda_3}{K_q K_v} p_s + \dfrac{(\lambda_2 + K_c)F_L + \lambda_1 \dot{F}_L}{A_1 K_q K_v} \end{cases}. \tag{12}$$

From Eqs. (11) and (12), it is shown that the hydraulic system can be described as a linear system. However, some internal parameters C_i, Ce, β_e and C_d are uncertain, due to they are difficult to accurate identify and vary with oil temperature and pressure. Besides, the variety of working conditions and different of forces always cause the external disturbance d cannot be precisely acquired. In respect of the dynamic system shown in the above equations, the following assumption will be made.

Assumption 1. The coefficient β is a positive value, and the desired trajectory x_d, and its first, second and third-order derivatives with respect to time are all bounded. In addition, the external disturbance d is limited by $|d|$£D. p_1 and p_2 are bounded by p_s and p_0, where $0 = p_0 \le p_1, p_2 \le p_s$.

3. Adaptive sliding model controller design

3.1. Sliding mode controller

Define the tracking error as $e=x_p-x_d$, and for convenience of writing, e, $\dot{e}$ and $\ddot{e}$ are generally substituted by e_1, e_2 and e_3, respectively. Thus, in the error state space, a sliding surface s can be chosen as:

$$s(t) = c_1 e + c_2 \dot{e} + \ddot{e} = c_1 e_1 + c_2 e_2 + e_3, \tag{13}$$

where: c_1 and c_2 are sliding surface parameters that can be calculated by using the pole collocation method and depended on the desired control performance. Moreover, both of c_1 and c_2 are positive constants, and they determine the dynamic performance of the system when the state of the system is located on the sliding surface.

Combine the Eqs. (11) to (13), the time derivative of s can be obtained by:

$$\dot{s}(t) = c_1 e_2 + c_2 e_3 + \frac{1}{\beta}\left(-\varphi_1 x_1 - \varphi_2 x_2 - \varphi_3 x_3 + u - d\right) - \dddot{x}_d. \tag{14}$$

In view of the inherent chattering problem in conventional sliding mode control, an exponential audience law is designed for the switch control part. In this way, the time derivative of s is present as:

$$\dot{s}(t) = -k_s s - \eta sgn(s), \tag{15}$$

where: k_s and η are the gain coefficients, and both of them are positive constants.

Furthermore, the switch gain η of sliding model control is given a moderate condition by:

$$\eta - D \ge 0, \tag{16}$$

where: Eq. (16) ensures the system can suppress modeling uncertainty and external disturbance, which have special significance to the performance of the controller. At this point, by combining the sliding model control based on exponent reaching law with adaptive algorithm, we can get:

$$u = u_{eq} + u_{smc}, \tag{17}$$

where: u_{eq} and u_{smc} represent equivalent control term and the switching term of system input, respectively. In particular, the equivalent control term determines the dynamics of the system on the sliding surface since it carries the system state vector over the reference trajectory. And they are defined by:

$$\begin{cases} u_{eq} = \varphi_1 x_1 + \varphi_2 x_2 + \varphi_3 x_3 + \beta\left(\dddot{x}_d - c_1 e_2 - c_2 e_3\right) \\ u_{smc} = -k_s s - \eta sgn\left(s\right) \end{cases}. \tag{18}$$

A Lyapunov stability theory is constructed to confirm the system stability with the SMC scheme.

Proof 1. Define a Lyapunov candidate function as:

$$V_1 = \frac{1}{2}\beta s^T s. \tag{19}$$

Inserting Eqs. (16) and (18) into Eq. (19), and differentiating V_1 with respect to t. We obtain:

$$\dot{V}_1 = \beta s\left(c_1 e_2 + c_2 e_3 - \dddot{x}_d\right) - s\left(\varphi_1 x_1 + \varphi_2 x_2 + \varphi_3 x_3 - u + d\right) \le - \\ \le -k_s s^2 - \left(\eta - D\right)|s| \le -k_s s^2. \tag{20}$$

Hence, according to equations (19) and (20), we can get that V_1 is positive definite and $\dot{V}_1$ is negative definite. Therefore, tracking errors of the end-effecter can converge asymptotically to an adequately small value as the time tends to infinity according to the established SMC law.

3.2. Parameter adaptive estimation algorithm

For a practical system of EHSS, the following assumption is first made in this section.

Assumption 2. The unknown term d is determined by other uncertain parameters and the external disturbance d_L, and assuming that d_{min} and d_{max} are the known lower and upper bounds of d. So the bounded d can be designed within a known limit as $d_{min} \le d \le d_{max}$. To simplify this discussion, an appropriate positive constant D is given to satisfy $-D \le d_{min}$ and $d_{max} \le D$. At this point, we can get $|d| \le D$.

In the real implementations of the proposed control strategy in this paper, the SMC controller is expanded with an improved adaptive algorithm to estimate the unknown parameters of the system. Thus, assuming that the uncertainty values β and φ_i (i=1, 2, 3) are estimated by $\hat{\beta}$ and $\hat{\varphi}_i$, so:

$$\begin{cases} \tilde{\varphi}_i = \varphi_i - \hat{\varphi}_i \\ \tilde{\beta} = \beta - \hat{\beta} \end{cases}, \tag{21}$$

where: $\tilde{\beta}$ and $\tilde{\varphi}_i$ are the ideal estimation errors of β and φ_i. Furthermore, choosing that the adjustable parameters can be automatically updated by the following adaptation laws as:

$$\begin{cases} \dot{\hat{\varphi}}_1 = -\gamma_1 s x_1 \\ \dot{\hat{\varphi}}_2 = -\gamma_2 s x_2 \\ \dot{\hat{\varphi}}_3 = -\gamma_3 s x_3 \\ \dot{\hat{\beta}} = -\gamma_4 s(\dddot{x}_d - c_1 e_2 - c_2 e_3) \end{cases}, \tag{22}$$

where: γ_i (i =1, 2, 3, 4) represent the strict positive constants related to the adaptation law. Then, combining Eq. (18) with Eq. (21), the expression for the input of the adaptive sliding controller can be shown as:

$$\hat{u} = \hat{u}_{eq} + u_{smc} =$$
$$= \hat{\varphi}_1 x_1 + \hat{\varphi}_2 x_2 + \hat{\varphi}_3 x_3 + \hat{\beta}\left(\dddot{x}_d - c_1 e_2 - c_2 e_3\right) - k_s s - \eta \operatorname{sgn}(s), \tag{23}$$

Finally, a Lyapunov stability theory is constructed to confirm the system stability with the ASMC scheme.

Proof 2. Let, the Lyapunov candidate function chosen as:

$$V_2 = \frac{1}{2}\beta s^2 + \sum_{i=1}^{3} \frac{1}{2\gamma_i}\tilde{\varphi}_i^2 + \frac{1}{2\gamma_4}\tilde{\beta}^2. \tag{24}$$

Then, the derivative of the V_2 with respect to time is:

$$\begin{aligned} \dot{V}_2 &= \beta s\dot{s} - \sum_{i=1}^{3}\frac{1}{\gamma_i}\tilde{\varphi}_i\dot{\hat{\varphi}}_i - \frac{1}{\gamma_4}\tilde{\beta}\dot{\hat{\beta}} = \beta s\left[c_1 e_2 + c_2 e_3 - \dddot{x}_d - \beta^{-1}\left(\sum_{i=1}^{3}\varphi_i x_i - u + d\right)\right] - \sum_{i=1}^{3}\frac{1}{\gamma_i}\tilde{\varphi}_i\dot{\hat{\varphi}}_i - \frac{1}{\gamma_4}\tilde{\beta}\dot{\hat{\beta}} = \\ &= s\left[(\beta - \hat{\beta})(c_1 e_2 + c_2 e_3 - \dddot{x}_d) - \sum_{i=1}^{3}(\varphi_i - \hat{\varphi}_i)x_i - k_s s - \eta \operatorname{sgn}(s) - d\right] - \sum_{i=1}^{3}\frac{1}{\gamma_i}\tilde{\varphi}_i\dot{\hat{\varphi}}_i - \frac{1}{\gamma_4}\tilde{\beta}\dot{\hat{\beta}} = \\ &= s\left[\tilde{\beta}\left(c_1 e_2 + c_2 e_3 - \dddot{x}_d\right) - \sum_{i=1}^{3}\tilde{\varphi}_i x_i - k_s s - \eta \operatorname{sgn}(s) - d\right] - \sum_{i=1}^{3}\frac{1}{\gamma_i}\tilde{\varphi}_i\dot{\hat{\varphi}}_i - \frac{1}{\gamma_4}\tilde{\beta}\dot{\hat{\beta}} = \tilde{\beta}\left[(c_1 e_2 + c_2 e_3 - \dddot{x}_d)s - \right. \\ &\left. - \frac{1}{\gamma_4}\dot{\hat{\beta}}\right] - k_s s^2 - \eta|s| - ds - \sum_{i=1}^{3}\tilde{\varphi}_i\left(s x_i + \frac{1}{\gamma_i}\dot{\hat{\varphi}}_i\right). \end{aligned} \tag{25}$$

When Eq. (22) is introduced into Eq. (25), Eq. (25) will be transformed into:

$$\dot{V}_2 = -k_s s^2 - \eta|s| - ds \le -k_s s^2 - (\eta - D)|s| \le -k_s s^2 \le 0, \tag{26}$$

which implies $\dot{V}_2 = 0$ as long as s=0, ie. when $\dot{V} = 0$ and then s=0. According to LaSalle invariance principle, the closed-loop system is asymptotically stable that $s \to 0$ as $t \to \infty$. Thus, from the adaptive law in Eq. (22) and the control input in Eq. (23), it is concluded that the EHSS will converge to zero along the sliding mode switching plane in a limited time.

Remark 1. *In spite of the demonstrated properties of the controller, the system may exhibits obvious chattering phenomenon due to the inaccurate model and large disturbance. Based on this, $\hat{\beta}$ and $\hat{\varphi}_i$ (i=1, 2, 3) are inevitably affected by tracking error and its corresponding derivatives in the actual physical system of EHSS. Therefore, this problem will overestimate these parameters and result in large oscillation or instability of the system.*

In order to avoid excessive estimation of parameters from seriously affecting the stability of controller or even destroying the system, the widely used projection algorithm [14] can be adopted, and the learning law (6) should be modified to:

$$\dot{\hat{\varphi}}_i = \operatorname{Proj}(\gamma_i \chi_i) = \begin{cases} 0, \ if \ \hat{\varphi}_i = \hat{\varphi}_{i\max} \ and \ \gamma_i \chi_i > 0 \\ 0, \ if \ \hat{\varphi}_i = \hat{\varphi}_{i\min} \ and \ \gamma_i \chi_i < 0 \\ \gamma_i \chi_i, \ otherwise \end{cases}, \tag{27}$$

where: $\dot{\hat{\beta}}$ is replaced by $\dot{\hat{\varphi}}_4$ easy to express, $\chi_1 = -s x_1$, $\chi_2 = -s x_2$, $\chi_3 = -s x_3$ and $\chi_4 = -s(\dddot{x}_d - c_1 e_1 - c_2 e_2)$. Moreover, the inequality $\varphi_{imin} < \hat{\varphi}_i(t) < \varphi_{imax}$ is established when the initial values of $\hat{\varphi}_i$ are described as

$\varphi_{imin} < \hat{\varphi}_i(0) < \varphi_{imax}$. Therefore, the projection algorithm (24) can guarantee the boundedness of unknown parameter estimates.

Remark 2. According to Eq. (15), the gain coefficient k_s should be large enough to make the system error quick approach the sliding surface. In addition, k_s as the slope of sliding mode surface ought to be as small as possible to help gain coefficient η system states stay on the sliding surface all the time. Hence, we should adjust them reasonably to prevent unexpected chattering problem to the system. Though the method, adjusting k_s by using the fuzzy logic control strategy while replacing the sign function with a saturation function, can somewhat decrease the undesirable chattering problem, it may have restricted due to its disadvantages of computation spending and seriously depend on expert experiences.

Therefore, for the sake of weakening the chattering phenomenon and shortening the time for tracking errors to reach the sliding surface, this paper proposes the idea of adjusting the gain coefficient k_s while replacing the sign function with the saturation function. In particular, the switching term of system input u_{smc} can be given as:

$$u_{smc} = \begin{cases} \eta sgn(s) + k_s s \ , & |s| > \sigma \\ \eta \dfrac{s}{\sigma} \ , & |s| \leq \sigma \end{cases}, \tag{28}$$

where: σ is a positive constant representing the boundary layer thickness of the saturation function. In Eq. (25), the gain coefficients η and k_s will be forced award the sliding mode surface at once the error states of the system are away from the surface, and the gain coefficient η will keep the system near the sliding surface when the error states of the system are in the neighborhood of the switching surface.

Furthermore, by combining Eqs. (23) and (25), the modified ASMC law can be shown as:

$$\hat{u} = \begin{cases} \hat{u}_{eq} - k_s - \eta sgn(s), & |s| > \sigma \\ \hat{u}_{eq} - \eta \dfrac{s}{\sigma} \ , & |s| \leq \sigma \end{cases}. \tag{29}$$

According to Eq. (29), the Eq. (26) will be expressed as:

$$\dot{V}_2 = \begin{cases} -k_s s^2 - (\eta + d)|s| \ , & |s| > \sigma \\ -\eta \dfrac{s^2}{\sigma} - ds \ , & |s| \leq \sigma \end{cases}. \tag{30}$$

Here, according to Eq. (26) that $\dot{V}_2 =\leq 0$ as long as $|s| > \sigma$, and otherwise $\dot{V}_2$ will be $-\eta s^2 / \sigma - ds$.

Moreover, considering that $|s| \leq \sigma$, $|d| \leq D$ and $|s| \leq \sigma$, $\dot{V}_2 \leq -|s|(\eta - D) \leq 0$. Thus, the Eq. (29) can realize the tracking control that the tracking error system firstly arrive at the sliding mode in the finite time and then reach the origin in the finite time.

4. Experiments and results

4.1. Introduction of experimental platform

As shown in Fig. 2, the experimental platform is composed of a single-rod hydraulic cylinder, two pressure sensors are used to measure two cylinder chambers, a displacement sensor (KEYENCE IL-300) providing accurate and continuous motion measurement for the end-effector, an electro-hydraulic servo valve (AVIC FF101-6) utilized in actuation of the hydraulic actuator, the external load is a tension spring which connects respectively with the fixed block and the end-effector, and a hydraulic supply with $P_s = 4$ MPa, etc. In order to ensure the control algorithms are conducted in real-time, they are run in Matlab/Simulink Real-time Workshop environment.

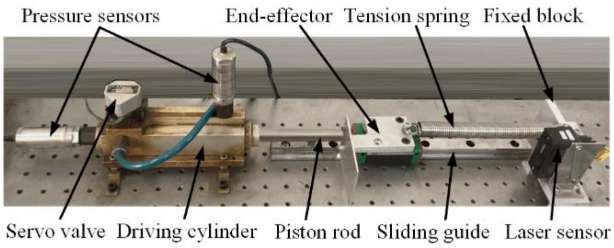

Fig. 2 Experiment platform of the EHSS

Furthermore, a host computer and a target computer are equipped in the industrial PC control system, and they exchange data through serial communication. In this configuration, the analog feedback signal from the displacement sensor is acquired via an analog-to-digital (A/D) card (Advantech PCL-818HD, which can provide 16 channels of single-ended analog input based on ISA bus). After that the acquired analog signals is processed by the designed controller, and the control command u is sent to the digital-to-analog (D/A) card (Advantech PCL-726, with 6 channels and 12 bits analog output channel). Finally, the control output signal is transformed into an analog signal and send to the corresponding servo valve through a power amplifier. The sampling rates for the host and target control layers are all set to 1 kHz.

4.2. Trajectory tracking experiments and results

Ignoring the influence of damping and the elastic damping in cylinder, details about EHSS parameters are set as follows: m=4kg, k=0, A_1=4.9×10^{-4}m^2, A_2=2.9×10^{-4}m^2, P_s=4×10^6 Pa, β_e=8×10^8 Pa, C_d=0.8, w=0.012m, C_i=5×10^{-13}m^3/(s·Pa), V_{10}=V_{20}=4.5×10^{-5}m^2, K_v=1×10^{-3}m/V. For the proposed ASMC algorithm, parameters of adaptive laws are set to γ_1=0, γ_2=2×10^{-6}, γ_3=5×10^{-7}, and γ_4=1×10^{-11}. The initial and extreme boundary values of system parameters are selected according to Table 1.

In order to illustrate its displacement tracking responses problem, the desired position was guided by a sinusoid wave trajectory as $y = 35 + 25\sin(\pi t / 2)$. When the system is stable without the external load, the position tracking response and corresponding parameters estimation of EHSS are as shown in Figs. 3 and 4, respectively. As Fig. 3 demonstrates that the cylinder rod can perform well in tracking the desired curve with errors within 1mm. Meanwhile, the uncertain parameters are dynamic adjusted in certain bounded values, as shown in Fig. 4.

Table 1

Uncertain parameters of the EHSS

Parameters	Initial value	Lower value	Upper value
φ_1	0	-	-
φ_2	1025	500	1200
φ_3	125	100	200
φ_4	0.01	0.001	0.1

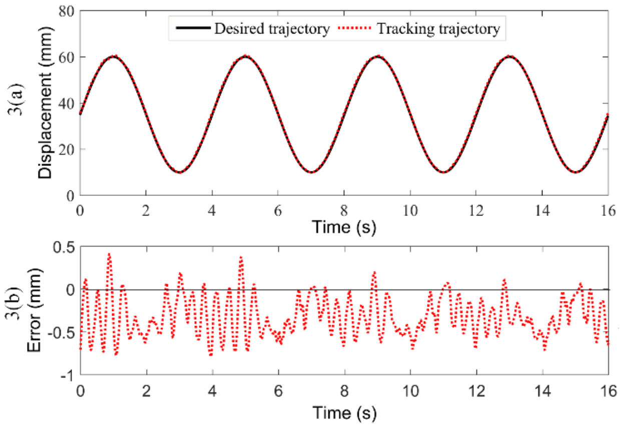

Fig. 3 Position tracking and errors of the developed ASMC controller

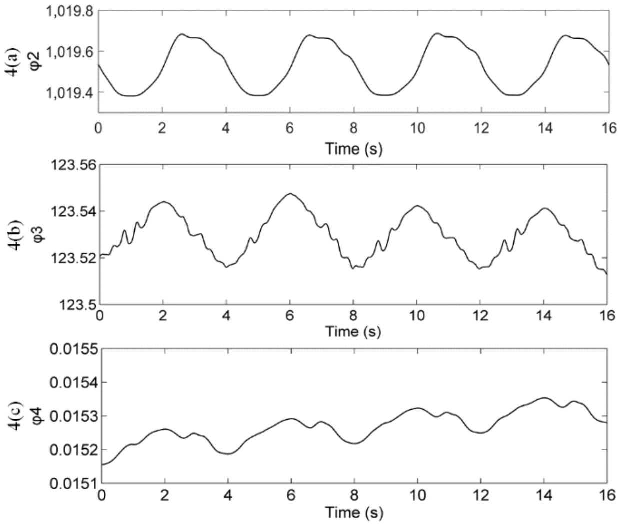

Fig. 4 Estimation values of three uncertain parameters

To validate the effectiveness of the proposed ASMC strategy, a traditional PID controller and an exponent approaching SMC algorithm are conducted to compare with it. In particular, parameters of these two controllers are well-tuned for the sake of best tracking performance. When the elastic load is zero, the motion trajectories and tracking errors of three controllers are shown in Fig. 5. As shown in Fig. 5, a, these controllers can make the end effector of the hydraulic cylinder follow the designed trajectory strictly. The tracking errors of the piston rod is not more than 3 mm as described in Fig.5b, which indicates that these controllers can obtain good dynamic response performance under no load. Specifically, for PID and SMC, the numerical range of trajectory error is between – 2.6 mm and 2.2 mm. The tracking error of ASMC is relatively very small, mostly within the measurement resolution of 1 mm, which verifies the high-performance nature of the proposed ASMC control strategy.

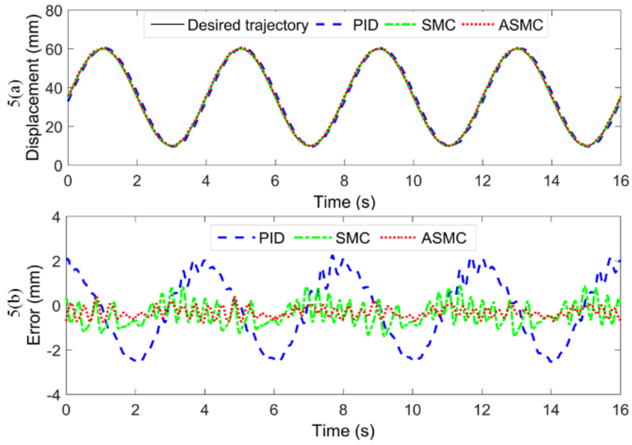

Fig. 5 Comparative results for position tracking control without external disturbance

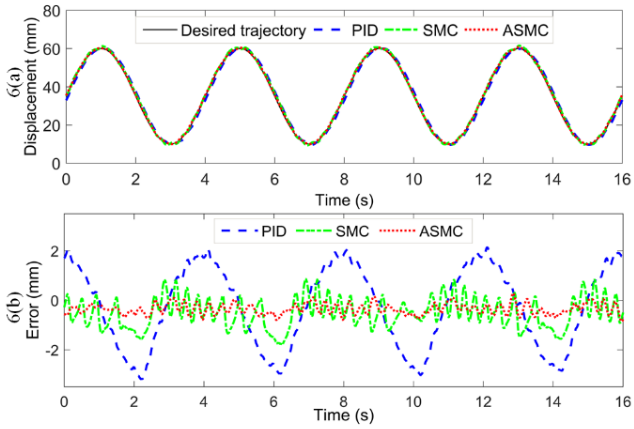

Fig. 6 Comparative results for position tracking control with external disturbance

Furthermore, to evaluate the robustness of the designed controller, the external disturbance is add by a tension spring to the control system. The dynamic behavior of the proposed controller is remaining better than the other two controllers as shown in Fig. 6. As is illustrated in Fig. 6a and Fig. 6, b, the PID controller cannot handle such a disturbance well and a large tracking error over 3 mm is exhibited. Though there are also some changes for SMC controller in terms of tracking error, it does not change so much as the PID controller. In contrast, the tracking error of the proposed ASMC is almost invariable. This experimental phenomenon denotes that both of the SMC controller and the proposed ASMC have more capability to suppress disturbance. Moreover, since the parametric estimation laws is used to adapt the actual hydraulic parameter with uncertainties, the proposed controller is more effective for the parametric uncertainties adaptation and load disturbance suppression comparison with the other two controllers.

5. Conclusion

In case of external disturbance and model uncertainty, this paper presents a discontinuous projection-based ASMC controller with variable sliding surface gain for position tracking control of an electro-hydraulic servo system. The dynamic model of the valve-controlled system is first established and the corresponding state-space equation is obtained. Next, the parametric adaptive estimation law and discontinuous projection algorithm are designed to estimate unknown parameters of the EHSS, which can effectively overcome the influence caused by the parameter uncertainty. After that, the sliding surface with variable gain and the saturation function are designed to modify the controller, with the purpose of solving the chattering problem and improving the robustness of the system. Finally, experimental results demonstrated that the designed controller has better tracking performance when compared with PID controller and the SMC controller.

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ADAPTIVE SLDING MODE CONTIONAL OF A NONLINEAR ELECTRO-HYDRAULIC SERVO SYSTEM FOR POSITION TRACING

S u m m a r y

In view of the electro-hydraulic position servo system with parameter uncertainty and bounded disturbances, an improved adaptive sliding mode control scheme is proposed. The mathematical model of the valve-controlled system is first constructed with consideration of the external disturbance, matched and mismatched unknown parameters. Then, the parametric adaptive estimation laws are established by Lyapunov technique to estimate the generalized uncertainty parameters, and the discontinuous projection algorithm is used to ensure the boundedness of parameter estimation. In order to eliminate the chattering phenomenon in sliding mode control, the saturation function is designed to replace the sign function and the gain coefficient is adjustable on the sliding surface. Finally, the comparative experimental results clarify that the proposed control scheme has better control performance than the PID controller and the SMC controller.

Key words: Electro-hydraulic servo system, parameter uncertainty, bounded disturbances, adaptive sliding mode control, trajectory tracing.

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