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Nonlinear Adaptive Robust Control of Valve-Controlled Symmetrical Cylinder System

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Abstract: A nonlinear adaptive robust control method based on a differentiable LuGre friction model is proposed for the problems of uncertain parameters, nonlinear friction and time-varying disturbances in valve-controlled symmetrical cylinder system. The proposed method can effectively compensate the uncertain nonlinear characteristics and external disturbances in the system by combining adaptive robust control with state and disturbance observations, as well as sliding mode differential technique. The global stability and boundedness of the proposed control method are proven by applying the Lyapunov theory. The simulation results show that this control method can deal with the problems of the uncertain nonlinearities and disturbances within the system. The proposed control method also presents excellent dynamic performance and robustness.

Key words: valve-controlled cylinder system; uncertain parameter; nonlinear characteristic; adaptive robust control

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The valve controlled servo system, which has been widely used in the robot manipulator[1], active suspension system[2] and load simulator[3], possesses large transmission power, fast response speed and high control precision. However, due to the effect of the nonlinear flow pressure characteristics of the servo valve, the bulk modulus of oil, the initial volumes of the hydraulic cylinder, friction and some other factors, there are a lot of parameter uncertainties and nonlinear problems in the valve-controlled servo system[4]. Therefore, the traditional control methods cannot meet the increasing performance requirements of the valve controlled servo system.

Focusing on the parameter uncertainties and nonlinear problems existing in the valve-controlled servo system, many scholars have carried out all kinds of research and tried to apply advanced control methods to improve the performance of the system. Such research endeavors include adaptive sliding mode control[5-6], backstepping neural network control[7] and adaptive robust control[8]. In the above control methods, the adaptive robust control method, applying the tuning function[9] and backstepping design to deal with the mismatched uncertain parameters and unmodeled disturbances in the system, has the advantages of adaptive control and robust control. However, in practical applications, the ever-increasing control gains may cause the problems of controller saturation and output chattering[10]. Therefore, based on adaptive robust control, different kinds of observers[11-12] are applied to compensate for unknown disturbances in the system and reduce the gains of adaptive robust control design.

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In order to improve the dynamic performance of the valve-controlled servo system, the nonlinear friction in the system should be controlled and compensated, and the continuous LuGre model-based friction compensation[13] has been widely used in hydraulic servo system[10,14]. Yin et al. proposed a continuously differentiable LuGre friction model to avoid the constraints of the non-differentiable model to the controller design[15]. To resolve the problems of adaptive robust control design with unknown nonlinear parameters, Guan et al. developed a novel Lyapunov function and presented a nonlinear adaptive robust controller for the hydraulic servo system[16].

Based on the above analysis, this paper comprehensively investigates the effect of uncertain parameters, nonlinear friction and external disturbances on the valve-controlled symmetrical cylinder system, as well as establishes a nonlinear mathematical model based on the continuously differentiable LuGre friction model. On this basis, a nonlinear adaptive robust controller and estimations of states and external disturbances are designed by applying the Lyapunov function, sliding mode differentiator and disturbance observer. Furthermore, the stability and boundedness of the proposed control method are proved. Finally, the simulation results under various conditions show that it has excellent control performance and robustness under different conditions.

1 System Modeling

The valve-controlled symmetrical cylinder system studied in this paper is shown in Fig. 1. It includes a symmetrical hydraulic cylinder, a servo valve, a hydraulic source, several variations of sensors and a controller. The definitions of symbol are shown in Tab. 1.

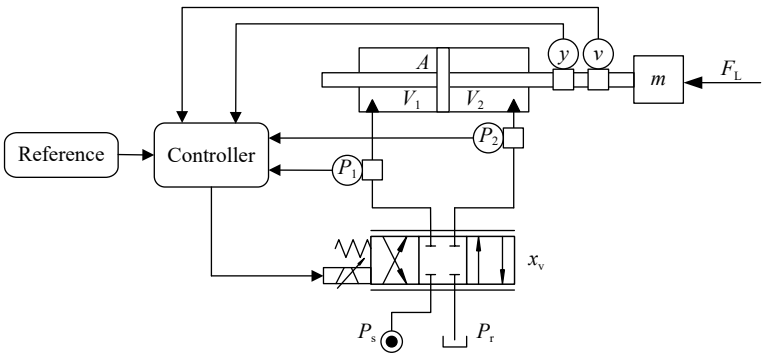


Fig. 1 Schematic diagram of valve-controlled symmetrical cylinder system

Tab. 1 Definitions of symbol

Symbol	Definition	Symbol	Definition
P_1	Pressure of the left cylinder chamber	m	Mass of the load
P_2	Pressure of the right cylinder chamber	y	Displacement of the load
P_s	Supply pressure of hydraulic source	v	Velocity of the load
P_r	Return pressure of hydraulic source	F_L	Unmodeled disturbances
V_1	Volume of the left cylinder chamber	A	Effective area of hydraulic cylinder
V_2	Volume of the right cylinder chamber	x_v	Spool displacement of servo valve

In order to simplify the system model, an assumption that the external leakage of the hydraulic cylinder is small and can be neglected is made. Another assumption is that the servo valve used in the system is a high-frequency servo valve and its dynamic characteristics can be regarded as a proportional element. Lastly, it is assumed that the connection pipeline is short and as such its dynamic characteristics can be ignored.

According to the principle of the valve-controlled symmetrical cylinder system[17], the system's mathematical model can be expressed as

$$\begin{aligned}&(P_1-P_2)A=ma+F_{\mathrm{f}}+F_{\mathrm{L}}\\&\frac{V_{10}+Ay}{\beta_{\mathrm{e}}}\frac{\mathrm{d}P_1}{\mathrm{d}t}=guR_1(P_1,u)-Av-C_{\mathrm{t}}(P_1-P_2)\\&\frac{V_{20}-Ay}{\beta_{\mathrm{e}}}\frac{\mathrm{d}P_2}{\mathrm{d}t}=Av+C_{\mathrm{t}}(P_1-P_2)-guR_2(P_2,u)\end{aligned}\tag{1}$$

where a is the acceleration of the load, F_{f} is the friction force of the hydraulic cylinder. V_{10} and V_{20} are the initial volumes of the left and right chambers of the hydraulic cylinder, respectively. β_{e} is the oil elastic modulus, and C_{t} is the leakage coefficient of the hydraulic cylinder. u is the servo valve control input and g is the servo valve control gain. The expressions of g, $R_1(P_1,u)$ and $R_2(P_2,u)$ can be found in Ref.[5].

The friction force F_{f} can be expressed by the continuously differentiable LuGre friction model

$$\begin{aligned}F_{\mathrm{f}}&=\sigma_0 z+\sigma_1\dot{z}+\sigma_2 v\\\dot{z}&=v-vz/N(v)\\N(v)&=(F_{\mathrm{s}}-F_{\mathrm{c}})\left[\tanh(a_1v)-\tanh(a_2v)\right]+\\&\quad F_{\mathrm{c}}\tanh(a_3v)\end{aligned}\tag{2}$$

where σ_0, σ_1 and σ_2 are the weighting factors of friction characteristic, and z is the average deformation of the bristles. F_{s} is the static friction force and F_{c} is the coulomb friction force. a_1, a_2 and a_3 are the shape factors that characterize the friction model.

By defining state variables $\boldsymbol{x}=\begin{bmatrix}x_1 & x_2 & x_3 & z\end{bmatrix}^{\mathrm{T}}$ and $x_1\triangleq y$, $x_2\triangleq v$, $x_3\triangleq(P_1-P_2)A/m$, the system's nonlinear mathematical model can be expressed as

$$\begin{aligned}\dot{x}_1&=x_2\\\dot{x}_2&=x_3-\theta_1 z-\theta_2(x_2-x_2/g(x_2)z)-\theta_3x_2-d\\\dot{x}_3&=\frac{-\theta_6x_2-\theta_7x_3}{-A^2x_1^2-\theta_6x_1+\theta_7}+\\&\quad\frac{\theta_8R_1+\theta_9R_2+\theta_{10}(R_2-R_1)x_1}{-A^2x_1^2-\theta_4x_1+\theta_5}u\\\dot{z}&=x_2-\frac{x_2}{g(x_2)}z\end{aligned}\tag{3}$$

where β and d are the equivalent elastic modulus and equivalent disturbance, respectively. $\beta=\beta_{\mathrm{e}}A/m$ and $d=F_{\mathrm{L}}/m$. $\theta_1=\sigma_0/m$, $\theta_2=\sigma_1/m$, $\theta_3=\sigma_2/m$, $\theta_4=A(V_{10}-V_{20})$, $\theta_5=V_{10}V_{20}$, $\theta_6=\beta A(V_{10}+V_{20})$, $\theta_7=C_{\mathrm{t}}\beta m(V_{10}+V_{20})/A$, $\theta_8=g\beta V_{20}$, $\theta_9=g\beta V_{10}$ and $\theta_{10}=g\beta A$.

It can be seen from Eq.(3) that the valve-controlled symmetrical cylinder system is a complex nonlinear system. It not only contains uncertain parameters, unknown friction states and external disturbances, but also includes the parameters that cannot be parameterized linearly. For such a complex nonlinear system, it is a challenging task to design a controller with satisfactory performance.

2 Controller Design

The design goal of this paper is to design a nonlinear adaptive robust controller for the established nonlinear mathematical model of the valve-controlled symmetric cylinder system, so that the system output can track the displacement reference $x_{\mathrm{d}}(t)$ as closely as possible, even in the presence of uncertain parameters, nonlinear friction, and external disturbances.

To facilitate the controller design, the following assumptions are made.

Assumption 1 The reference signal $x_{\mathrm{d}}(t)\in\mathrm{C}^2$ is bounded.

Assumption 2 Uncertain parameters and equivalent disturbances are bounded, namely

$$\theta_{i\min}\le\theta_i\le\theta_{i\max},\quad i=1,\cdots,10,\ \ |d|\le D$$

where $\theta_{i\max}$ and $\theta_{i\min}$ are the maximum and minimum values of uncertain parameters θ_i, and D is a positive constant.

Assumption 3 P_1 and P_2 are bounded and are satisfied by $P_{\mathrm{r}}\le P_1\le P_{\mathrm{s}}$ and $P_{\mathrm{r}}\le P_2\le P_{\mathrm{s}}$. Meanwhile, the volumes of the two chambers of the hydraulic cylinder V_1, V_2 are always positive and $V_1=V_{10}+Ax_1>0$, $V_2=V_{20}-Ax_1>0$.

2.1 Parameter estimation and discontinuous projection

2.1.1 Estimation of the unmeasured state variable

In the LuGre friction model, the deforma-

tion of the bristles,z, is an unmeasured state and the state z is associated with two sub-items in the LuGre model, hence a dual observer[14] can be used to estimate the unmeasured state z.

The dual observer can be expressed as

$$\begin{aligned}\frac{\mathrm{d}\hat{z}_1}{\mathrm{d}t} &= v - \frac{v}{N(v)}\hat{z}_1 + \iota_1\\ \frac{\mathrm{d}\hat{z}_2}{\mathrm{d}t} &= v - \frac{v}{N(v)}\hat{z}_2 + \iota_2\end{aligned} \tag{4}$$

where $\hat{z}_1$ and $\hat{z}_2$ are the estimated values of the unmeasured state z, respectively. ι_1 and ι_2 are the adaptive update laws to be designed later.

2.1.2 Sliding mode differentiator

Since the derivatives of the state variables and the control gains in the controller are complicated to solve because of uncertain parameters, sliding mode differentiators[18] are used to estimate the derivatives of the state variables and the control gains. The structure of the sliding mode differentiators can be expressed as

$$\begin{cases}\dot{w} = -\rho(v,w)\mathrm{sign}(w-v)\\ \hat{\dot{v}} = -\dfrac{\rho(v,w)}{s+\tau}\mathrm{sign}(w-v)\end{cases} \tag{5}$$

where w is the estimated value of the variable v and $\rho(v,w)$ is the gain of the sliding mode differentiator. τ is the first-order filter time constant and s is the Laplace operator. The sliding mode differentiator can make the estimated value follow the true value after a finite time, so the estimated derivative $\hat{\dot{v}}$ is bounded and there is a positive constant δ satisfying that $\delta = \sup|\dot{v} - \hat{\dot{v}}|$.

2.1.3 Discontinuous projection of deterministic boundary parameters

The presence of disturbances may cause the adaptive estimation of uncertain parameters to diverge and undermine the stability of the system. In order to avoid this problem, a commonly used method is to use discontinuous projection to constrain the uncertain parametric estimation within a deterministic boundary, i.e.,

$$\mathrm{Proj}_{\hat{\theta}_i}(\tau_i) = \begin{cases}0, & \hat{\theta}_i = \theta_{i\max} \text{ and } \tau_i > 0\\ 0, & \hat{\theta}_i = \theta_{i\min} \text{ and } \tau_i < 0\\ \tau_i, & \text{otherwise}\end{cases} \tag{6}$$

where τ_i is the adaptive update rate of parameter estimation. From Assumption 2, it can be proven that the discontinuous projection of deterministic boundary parameters satisfies the following characteristics, namely,

$$\begin{aligned}&\widehat{\theta}_i \in \left\{\widehat{\theta}_i : \theta_{i\min} \le \widehat{\theta}_i \le \theta_{i\max}\right\}, \ i = 1,\cdots,10\\ &\tilde{\theta}_i^{\mathrm{T}}\left[\gamma_i^{-1}\mathrm{Proj}_{\hat{\theta}}(\gamma_i\tau_i) - \tau_i\right] \ge 0, \ \ \forall\tau_i\end{aligned} \tag{7}$$

2.2 Design of nonlinear adaptive robust controller

The position tracking error e_1 and auxiliary error e_2, e_3 of valve-controlled symmetric system can be defined as

$$e_1 = x_1 - x_{\mathrm{d}},\ e_2 = x_2 - x_{2\mathrm{d}},\ e_3 = x_3 - x_{3\mathrm{d}} \tag{8}$$

where $x_{2\mathrm{d}}$ and $x_{3\mathrm{d}}$ are virtual controls that can be expressed as

$$x_{2\mathrm{d}} = \dot{x}_{\mathrm{d}} - k_1 e_1,\ x_{3\mathrm{d}} = x_3 - e_1 - k_1\dot{e}_1 - k_2 e_2 \tag{9}$$

Here, k_1 and k_2 are the controller gains and k_2 can be expressed as $k_2 = \eta_2 + s_1$, where η_2 is a positive constant and s_1 is a damping term. s_1 will need to be designed in a manner that satisfies $s_1 e_2^2 \geqslant -\varepsilon_1 - e_2\psi_1$, in which ε_1 is a positive constant and its value is determined by the allowable error of e_2 and the initial states of the system. ψ_1 can be expressed as

$$\begin{aligned}\psi_1 = &-\theta_1\tilde{z}_1 - \tilde{\theta}_1\hat{z}_1 - \tilde{\theta}_2 x_2 - \tilde{\theta}_3 x_2 +\\ &\theta_2\tilde{z}_2 x_2/N(x_2) + \tilde{\theta}_2\hat{z}_2 x_2/N(x_2) - \tilde{d}\end{aligned} \tag{10}$$

where $\tilde{z}_1$ and $\tilde{z}_2$ are the estimation errors of unknown state z and satisfies that $\tilde{z}_1 = z - \hat{z}_1$, $\tilde{z}_2 = z - \hat{z}_2$. The disturbance estimation error can be expressed as $\tilde{d} = d - \hat{d}$ and the disturbance estimation can be expressed as $\hat{d} = x_3 - \ddot{x}_{\mathrm{d}} - \hat{\theta}_1\hat{z}_1 - \hat{\theta}_3 x_2 - \hat{\theta}_2(x_2 - x_2/N(x_2)\hat{z}_2)$.

The designed nonlinear adaptive robust controller is

$$u = \frac{1}{\hat{\boldsymbol{\alpha}}_{h_1}^{\mathrm{T}}\boldsymbol{h}_1}\left(-\frac{1}{2}\hat{\boldsymbol{\alpha}}_{\mathrm{g}}^{\mathrm{T}}\dot{\boldsymbol{h}}_{\mathrm{g}}e_3 - \hat{\boldsymbol{\alpha}}_{h_2}^{\mathrm{T}}\boldsymbol{h}_2 + \hat{\boldsymbol{\alpha}}_{\mathrm{g}}^{\mathrm{T}}\boldsymbol{h}_{\mathrm{g}}\phi_u - e_2 - k_3 e_3\right) \tag{11}$$

where $\hat{\boldsymbol{\alpha}}_{\mathrm{g}} = \left[1\ \ \hat{\theta}_4\ \ \hat{\theta}_5\right]^{\mathrm{T}}$, $\boldsymbol{h}_{\mathrm{g}} = [-A^2x_1^2\ \ -x_1\ \ 1]^{\mathrm{T}}$, $\dot{\boldsymbol{h}}_{\mathrm{g}} = [-2A^2x_1x_2\ \ -x_2\ \ 0]^{\mathrm{T}}$, $\hat{\boldsymbol{\alpha}}_{h1} = \left[\hat{\theta}_6\ \ \hat{\theta}_7\right]^{\mathrm{T}}$, $\boldsymbol{h}_1 = [-x_2\ \ -x_3]^{\mathrm{T}}$, $\hat{\boldsymbol{\alpha}}_{h_2} = \left[\hat{\theta}_8\ \hat{\theta}_9\ \hat{\theta}_{10}\right]^{\mathrm{T}}$, $\boldsymbol{h}_2 = [R_1\ R_2\ (R_2 - R_1)x_1]^{\mathrm{T}}$ and ϕ_u

can be expressed as $\phi_u = (-k_1^3 + 2 + k_2)\, e_1 + \left(k_1^2 - 1 + k_1 k_2 + k_2^2 - \hat{\dot{k}}_2\right) e_2 + \hat{\dot{x}}_3 - (k_1 + k_2)\, e_3$.

Meanwhile, $\hat{\dot{x}}_3$ is the derivative estimate value of x_3 obtained by the sliding mode differentiator. $\tilde{\dot{x}}_3$ is the estimation error of $\dot{x}_3$, and $\tilde{\dot{x}}_3 = \dot{x}_3 - \hat{\dot{x}}_{33}$. $\hat{\dot{k}}_2$ is the derivative estimation obtained by the same way as $\hat{\dot{x}}_{33}$. $\tilde{\dot{k}}_2$ is the estimate error of $\dot{k}_2$ and $\tilde{\dot{k}}_2 = \dot{k}_2 - \hat{\dot{k}}_2$. k_3 is the controller gain that needs to be designed and k_3 can be expressed as $k_3 = \eta_3 + s_2$, where η_3 is a positive constant and s_2 is a damping term. s_2 needs to be designed in a manner that satisfies $s_2 e_3^2 \geqslant -\varepsilon_2 - e_3 \psi_2$, in which ε_2 is a positive constant and its value is determined by the allowable error of e_3 and the initial states of the system. ψ_2 can be expressed as

$$\begin{aligned}\psi_2 =& \frac{1}{2}\tilde{\boldsymbol{\alpha}}_g^{\mathrm{T}} \dot{\boldsymbol{h}}_g e_3 + \tilde{\boldsymbol{\alpha}}_{h_1}^{\mathrm{T}} \boldsymbol{h}_1(x) u + \tilde{\boldsymbol{\alpha}}_{h_2}^{\mathrm{T}} \boldsymbol{h}_2 + \\ & \boldsymbol{\alpha}_g^{\mathrm{T}} \boldsymbol{h}_g \left((k_1 + k_2)\, \psi_1 - \tilde{x}_3 + e_2 \tilde{k}_2\right) - \tilde{\boldsymbol{\alpha}}_g^{\mathrm{T}} \boldsymbol{h}_g \phi_u\end{aligned} \tag{12}$$

2.3 Stability proof of the controller

The parameter adaptive laws are selected as

$$\begin{cases} \dot{\hat{\theta}}_1 = \mathrm{proj}_{\theta_1}\left(-\gamma_1 \hat{z}_1 \left(e_2 + (k_1 + k_2)\, \hat{\alpha}_g^{\mathrm{T}} h_g(x) e_3\right)\right) \\ \dot{\hat{\theta}}_2 = \mathrm{proj}_{\theta_2}\left(\gamma_2 \left(-x_2 + x_2 / N(x_2)\, \hat{z}_2\right)\right. \\ \qquad \left.\left(e_2 + (k_1 + k_2)\, \hat{\alpha}_g^{\mathrm{T}} h_g(x) e_3\right)\right) \\ \dot{\hat{\alpha}}_3 = \mathrm{proj}_{\theta_3}\left(-\gamma_3 x_2 \left(e_2 + (k_1 + k_2)\, \hat{\alpha}_g^{\mathrm{T}} h_g(x) e_3\right)\right) \\ \dot{\hat{\alpha}}_g = \mathrm{proj}_{\alpha_s}\left(T_g \left(\frac{1}{2} \dot{h}_g(x) e_3^2 - h_g(x) \phi_u e_3\right)\right) \\ \dot{\hat{\alpha}}_{h_2} = \mathrm{proj}_{\alpha_4}\left(T_{h_1} h_{h_1}(x) u e_3\right) \\ l_1 = -\left(e_2 + (k_1 + k_2)\, \hat{\alpha}_g^{\mathrm{T}} h_g(x) e_3\right) \\ l_2 = \left(e_2 + (k_1 + k_2)\, \hat{\alpha}_g^{\mathrm{T}} h_g(x) e_3\right) x_2 / N(x_2) \end{cases} \tag{13}$$

By choosing the Lyapunov function as follows

$$\begin{aligned} V_a =& \frac{1}{2} e_1^2 + \frac{1}{2} e_2^2 + \frac{1}{2} \alpha_g^{\mathrm{T}} h_g(x) e_3^2 + \\ & \frac{1}{2}\left(\tilde{\alpha}_g^{\mathrm{T}} T_g^{-1} \tilde{\alpha}_g + \tilde{\alpha}_{h1}^{\mathrm{T}} T_{h1}^{-1} \tilde{\alpha}_{h1} + \tilde{\alpha}_{h2}^{\mathrm{T}} T_{h2}^{-1} \tilde{\alpha}_{h2}\right) + \\ & \frac{1}{2}\left(\theta_1 \tilde{z}_1^2 + \theta_2 \tilde{z}_2^2 + \frac{1}{\gamma_1} \tilde{\theta}_1^2 + \frac{1}{\gamma_2} \tilde{\theta}_2^2 + \frac{1}{\gamma_3} \tilde{\theta}_3^2\right) \end{aligned} \tag{14}$$

the derivative of Eq. (14) with respect to time when combined with Eqs. (9) (13) will yield

$$\begin{aligned} \dot{V}_a =& -k_1 e_1^2 - k_2 e_2^2 - k_3 e_3^2 + e_2 \tilde{d} - \frac{x_2}{N(x_2)} \tilde{z}_1^2 - \\ & \frac{x_2}{N(x_2)} \tilde{z}_2^2 + e_3\left(\alpha_g^{\mathrm{T}} h_g(x)\left((k_1 + k_2)\, \tilde{d} - \tilde{x}_3 + \tilde{k}_2\right)\right) - \\ & e_3\left(\alpha_g^{\mathrm{T}} h_g(x)\left((k_1 + k_2)\left(\psi_1 - \tilde{d}\right)\right)\right) \leq \\ & -k_1 e_1^2 - \eta_2 e_2^2 - \eta_3 e_3^2 - \frac{x_2}{N(x_2)} \tilde{z}_1^2 - \\ & \frac{x_2}{N(x_2)} \tilde{z}_2^2 + \varepsilon_1 + \varepsilon_2 \leq -2\lambda V_a + \Delta \end{aligned} \tag{15}$$

where λ and Δ can be expressed as $\lambda = \min\left\{k_1, \eta_2, \frac{\eta_3}{\alpha_g^{\mathrm{T}} h_g(x)}, \frac{x_2}{N(x_2)\, \theta_{1\max}}, \frac{x_2}{N(x_2)\, \theta_{2\max}}\right\}$ and $\Delta = \varepsilon_1 + \varepsilon_2 + \lambda\left(\tilde{\alpha}_g^{\mathrm{T}} T_g^{-1} \tilde{\alpha}_g + \tilde{\alpha}_{h1}^{\mathrm{T}} T_{h1}^{-1} \tilde{\alpha}_{h1} + \frac{1}{\gamma_1} \tilde{\theta}_1^2 + \tilde{\alpha}_{h2}^{\mathrm{T}} T_{h2}^{-1} \tilde{\alpha}_{h2} + \frac{1}{\gamma_2} \tilde{\theta}_2^2 + \frac{1}{\gamma_3} \tilde{\theta}_3^2\right)$.

From Eq. (15), we can conclude that

$$V_a \leqslant \exp(-\lambda t) V_a(0) + \frac{\Delta}{\lambda}[1 - \exp(-\lambda t)] \tag{16}$$

As can be seen from Eq. (16), V_a is bounded. With Assumptions 1 and 2, we can see that the system parameters and the state variables are bounded, and the control input and output are also always bounded. Therefore, all the signals of the closed-loop system are bounded, and the tracking error eventually converges to a known bound Δ/λ.

3 Simulation and Discussion

A co-simulation model of valve-controlled symmetrical cylinder system was established using Matlab and Amesim software. The co-simulation diagram of valve-controlled symmetrical cylinder system is shown in Fig. 2. The control input range of the servo valve is $[-10\ 10]$ mA and its frequency response is 20 Hz. The supply pressure of the hydraulic system is 21 MPa and the return pressure is 0 MPa. The hydraulic cylinder parameters are 200 mm/125 mm/600 mm. The other parameters of the hydraulic system are listed as follows: $\beta_e = 1.6$ GPa, $V_{10} = 4.3 \times 10^{-3}$ m^3, $V_{20} = 4.3 \times 10^{-3}$ m^3, $g = 2.52 \times 10^{-7}$ m$^3 \cdot$ s$^{-1} \cdot$ V$^{-1} \cdot$ Pa$^{-\frac{1}{2}}$, $C_t = 2 \times 10^{-20}$ m$^3 \cdot$ s$^{-1} \cdot$ Pa^{-1}, $m = 1\,600$ kg.

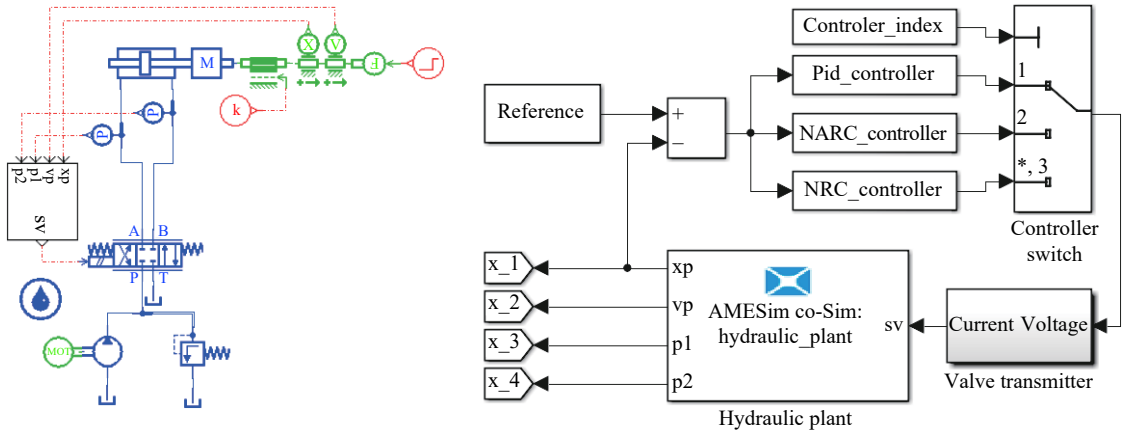
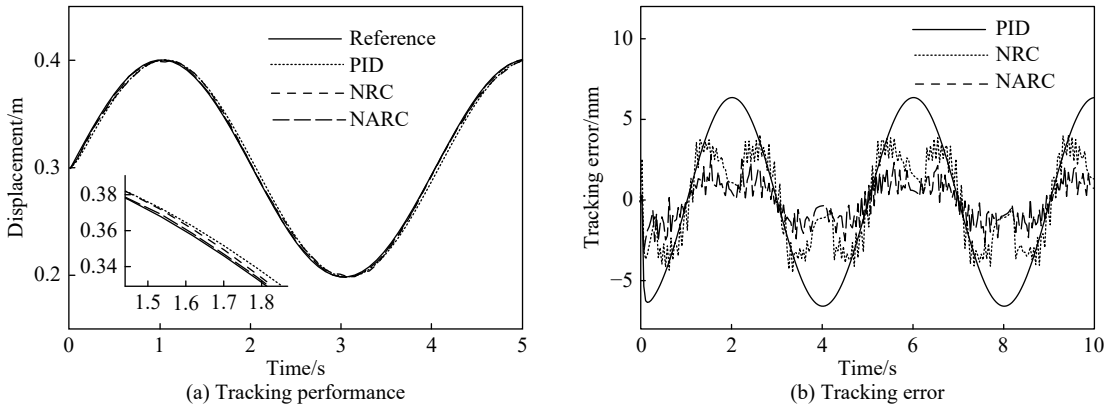


Fig. 2 Co-simulation diagram of valve-controlled symmetrical cylinder system

In order to verify the tracking performance and robustness of the proposed controller, the nonlinear comparative simulations of adaptive robust controller (NARC), deterministic robust control (NRC) and PID controller were carried out under nominal and disturbance conditions. The parameters of NARC are $k_1 = 100$, $\eta_2 = 50$, $\eta_3 = 25$, $\epsilon_1 = \epsilon_2 = 0.01$, $\gamma_1 = 1$, $\gamma_2 = \gamma_3 = 10^{-6}$, $T_g = \mathrm{diag}([0,\ 10^{-5},\ 10^{-17}])$, $T_{h_1} = \mathrm{diag}([10^{-15},\ 10^{-20},\ 10^{-15}])$, $T_{h_2} = \mathrm{diag}([10^{-17},\ 10^{-17}])$. The choice of s_1 and s_2 can be attained from Ref.[7]. The parameters of NRC are the same as those of NARC expect that all the adaptive gains are equal zero. The parameters of PID controller are selected as $k_p = 300$, $k_i = 1$, $k_d = 1$, which are tuned carefully via error-and-try method.

The reference tracking trajectory is $0.1\sin(0.5\pi t)$ m. The tracking performance and error comparisons of the valve-controlled symmetric cylinder system under nominal condition are shown in Fig. 3. It can be seen that the tracking error of the NARC is much smaller than that of NRC and PID controller. Meanwhile, it can be seen that there is a certain lag between the tracking signal and the reference signal with PID controller. However, NARC can track the reference signal more accurately. Simulation results show that the proposed controller can significantly improve the dynamic performance and tracking accuracy of the valve-controlled symmetric cylinder system.



(a) Tracking performance (b) Tracking error

Fig. 3 Comparisons of tracking performance and error with different controllers under nominal condition

In order to test the robustness of the NARC, an external disturbance $F_L = 50\,000$ N is applied to the system at 3 s. The tracking performance and error comparisons of three kinds of controllers under disturbance condition are shown in Fig. 4. It can be seen that all the controllers achieved good tracking performance, but the tracking error and dynamic performance of NARC are still better than the NRC and PID controller obviously.

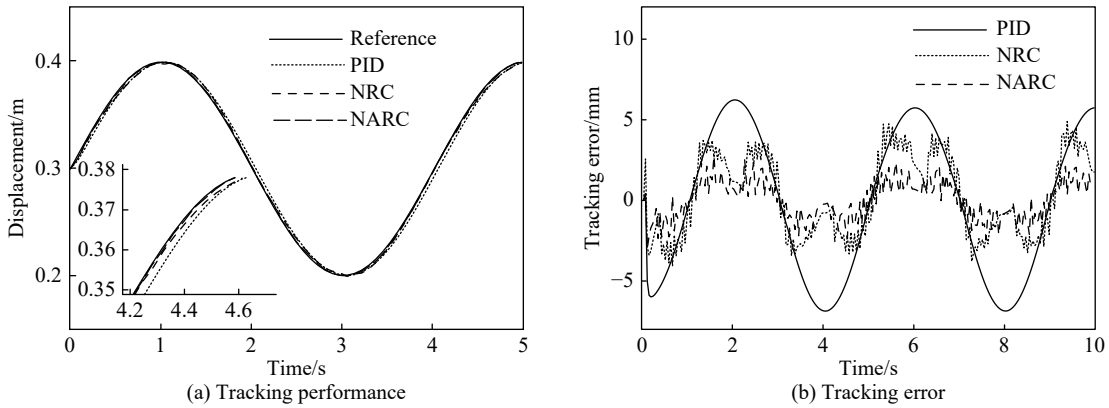


Fig. 4 Comparisons of tracking performance and error with different controllers under disturbance condition

4 Conclusion

A nonlinear adaptive robust controller of valve-controlled symmetrical cylinder is presented in this paper. Firstly, a nonlinear mathematical model of hydraulic system with a continuously differentiable LuGre friction model is established. Then a nonlinear adaptive robust controller is constructed to deal with uncertain nonlinear parameters and external disturbances. Meanwhile, a dual state observer, sliding mode differentiator, and discontinuous projection are used to estimate the unmeasured state, calculate approximate derivate of state and suppress the divergence of parameter estimation respectively. The global stability and boundedness of the proposed controller are proven theoretically by applying a special Lyapunov function. Finally, simulation results show that this controller can maintain excellent dynamic performance and robustness under different conditions.

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