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Adaptive Backstepping Slide Mode Control of Pneumatic Position Servo System

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Abstract: With the price decreasing of the pneumatic proportional valve and the high performance micro controller, the simple structure and high tracking performance pneumatic servo system demonstrates more application potential in many fields. However, most existing control methods with high tracking performance need to know the model information and to use pressure sensor. This limits the application of the pneumatic servo system. An adaptive backstepping slide mode control method is proposed for pneumatic position servo system. The proposed method designs adaptive slide mode controller using backstepping design technique. The controller parameter adaptive law is derived from Lyapunov analysis to guarantee the stability of the system. A theorem is testified to show that the state of closed-loop system is uniformly bounded, and the closed-loop system is stable. The advantages of the proposed method include that system dynamic model parameters are not required for the controller design, uncertain parameters bounds are not need, and the bulk and expensive pressure sensor is not needed as well. Experimental results show that the designed controller can achieve better tracking performance, as compared with some existing methods.

Keywords: pneumatic position servo system, adaptive backstepping design, slide mode control, uncertain parameter, tracking accuracy

1 Introduction

Pneumatic actuators are widely used in the field of industrial automation since they are inexpensive, easily maintained, clean, safe to operate, fast acting, self cooling, easily connected to a payload, and have a high power-to-weight ratio. Yet they have high order time variant dynamics suffering from nonlinear friction force, dead zone due to stiction force caused by sealing ring of cylinder, and weak stiffness due to compressibility of air. These factors offer difficulties of achieving high accuracy tracking control of actuators[1].

A number of works have been done to improve tracking accuracy of pneumatic position servo system. PID control is widely used in pneumatic position servo system, due to its simplicity. VARSECELD, et al[2], implemented a PID controller in the pneumatic system based on the switch valve, which were implemented by adding the bounded integral action, position feed forward, and friction compensation. In a subsequent study, WANG, et al[3], proposed a practical control strategy based on modification of PID controller by adding acceleration feedback and nonlinear compensation. Position compensation algorithm with time delay minimization was used to achieve the position tracking precisely. However, the adaptability of PID control is poor, when the payload or other conditions of the system change, the PID control system might degrade. To improve the performance of PID controller, SALIM, et al[4], proposed a self-regulation nonlinear PID controller to obtain higher tracking accuracy of pneumatic system, by utilizing the characteristic of rate variation of nonlinear gain. In recent work[5–8], neural network PID controller and fuzzy PID controller were used to control the position of pneumatic system. But these methods cannot achieve high trajectory tracking accuracy. The integration of adaptive PID with repetitive control and grey relation compensation were proposed to improve the trajectory tracking accuracy in Refs. [9–10]. However, the expensive pressure sensor is needed in these methods.

REN, et al[11–12], designed two adaptive state feedback controllers based on backstepping to control the position of pneumatic system. These simple controllers had high tracking accuracy without pressure sensor. However, the closed-loop stability of the systems have not been testified. RIACHY, et al[13], proposed a state feedback nonlinear controller based on the concepts of homogeneity and finite-time stability. The method got high precision, but the main disadvantage is the chattering of the control signal.

As a robust method to deal with the model uncertainty, Slide Mode Controller(SMC) has been applied to the pneumatic servo system, which is considered to be a suitable method for pneumatic system control. NGUYEN,

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et al[14], designed a SMC using the position and the pressure feedback in the pneumatic system based on the solenoid valve, where the sign of a second order sliding surface was used to control the solenoid valve state to track the time variant input signal. But this method needed expensive pressure sensor. HODGSON, et al[15], designed two SMCs using the three mode average model and the seven mode average model of pneumatic system respectively, based on four solenoid valves and Pulse Width Modulation(PWM) technique, to achieve good position tracking, but an accurate model of the system was needed in this method.

Recently, servo valves were widely used to drive pneumatic actuators instead of solenoid valves due to the reduced price and increased reliability. BONE, et al[16], proposed two SMCs based on third order both linear and nonlinear models for pneumatic servo system to obtain the expected results. However, these methods were sensitive to the payload variations. TAHERI, et al[17], presented a backstepping slide mode force-stiffness controller based on an accurate mathematical model of the pneumatic servo system. This method had achieved very good tracking performance in force control. ESTRADA, et al[18], proposed an output feedback controller based on impulsive gain and second order slide mode method for pneumatic servo system. This method did not require the derivative of slide mode variable. A gain adjustment mechanism was used to thus reduce the chattering effectively. However, the conventional SMC design method is model dependent, due to the requirement of knowledge about the nonlinear terms and the boundaries of the corresponding uncertain parameters.

EDJEKOUANE, et al[19], proposed a model order reduced controller, which used a second order with delay input to design homogeneous robust controller. However, this reduced model neither decreases the design complexity nor improves the tracking accuracy. GARMSIRI, et al[20], applied an intelligent brain emotional learning control in the pneumatic servo system, which did not need the model information. Yet the learning rate and emotion signal selection is difficult; at the same time, the tracking accuracy is not high. BOUBAKIR, et al[21], proposed a stable linear adaptive controller to control the position of pneumatic system, which didn't need the model parameters information as well. MENG, et al[22], compared three model-based approaches for the compensation of hysteresis. Each approach added feedforward hysteresis compensation in an existing adaptive backstepping control structure for high-speed pneumatic muscles, all approaches could reduce tracking error effectively. The robust adaptive control methods were used to track pneumatic position and pressure trajectory in Refs. [23−25], yet all these methods need pressure sensors.

In this paper, an adaptive backstepping sliding mode control method is proposed for the pneumatic position servo system. The proposed method uses the backstepping design to get a slide mode controller which guarantees the stability of the control system. The controller requires neither the exact system model parameters nor uncertainty boundary of the uncertain parameters. The proposed method uses a simplified linearized model with unknown parameters, which is a universal assumption. This point is superior to the existing slide mode controller. Moreover, the proposed method uses the displacement sensor information without the expensive pressure sensor, which makes the system configuration simple and cheap. The experimental results show that the proposed method can track the different reference signals and achieve higher tracking accuracy as compared to the five existing methods.

This paper is organized as follows. Section 2 introduces the pneumatic system and its simplified linear mathematical model briefly. Section 3 gives the design procedure of the proposed adaptive slide mode controller. Section 4 proves the stability of the control system. The experimental results of the proposed controller and the comparison to the five existing controllers(including two sliding mode controllers, two state feedback controllers and a linear adaptive controller) are given to show the effectiveness and superiority of the proposed method in section 5. Section 6 offers some conclusions.

2 Mathematical Model of Pneumatic Position Servo System

The schematic diagram of the pneumatic position servo system is shown in Fig. 1.

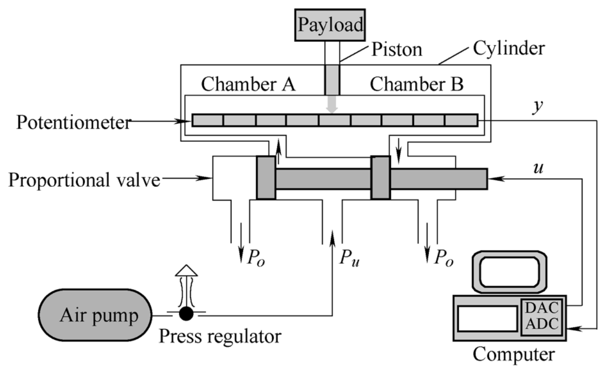


Fig. 1. Schematic diagram of pneumatic position servo system

In Fig. 1, the air pump is used to provide the compressed air, which flows into two chambers of a cylinder(chamber A or chamber B) through the regulation of a proportional valve. The input voltage of the proportional valve adjusts the mass flow into each chamber of the cylinder in order to obtain the pressure difference between the two chambers that drive the piston and payload. The displacement of the payload is measured by the potentiometer. The displacement signal is then fed into the computer through a analog-to-digital converter(ADC). The controller output is given to the proportional valve through a digital-to-analog converter(DAC). The ADC and DAC are integrated into a general purpose converter card installed in the computer.

The controller programmed in the computer regulates mass flow rate into chambers A and B through DAC in order to drive the piston and payload. The control objective is to drive the position of the payload to track a desired trajectory.

The dynamic model of the pneumatic position servo system is given as follows[11–12]:

$$\begin{cases} \dot{m}_a = f_a(u, p_a), \\ \dot{m}_b = f_b(u, p_b), \\ KRT\dot{m}_a = Kp_a A_a \dot{y} + A_a(y_0 + y)\dot{p}_a, \\ KRT\dot{m}_b = -Kp_b A_b \dot{y} + A_b(y_0 - y)\dot{p}_b, \\ M\ddot{y} = p_a A_a - p_b A_b - F_f, \end{cases} \tag{1}$$

where $\dot{m}_a$ and $\dot{m}_b$ are the mass flow rates into chambers A and B, respectively. p_a and p_b are pressures of chambers A and B, respectively. A_a and A_b are the cross section area of the piston with respect to chambers A and B. y is the payload displacement, y_0 is the initial payload displacement, M is the total mass of the payload and piston, F_f is the friction force, $f_a(u, p_a)$ and $f_b(u, p_b)$ are nonlinear functions of the upper stream and lower stream pressure of chambers A and B, respectively. K is the specific heat ratio, R is the ideal gas constant, and T is the air temperature. u is the input voltage of the proportional valve.

By ignoring the friction and linearizing the nonlinear function $f_a(u, p_a)$ and $f_b(u, p_b)$, the third-order linear model of the pneumatic position system is obtained as follows[11]:

$$\begin{cases} \dot{x}_1 = x_2, \\ \dot{x}_2 = x_3, \\ \dot{x}_3 = a_1 x_1 + a_2 x_2 + a_3 x_3 + (1/b)u, \\ y = x_1, \end{cases} \tag{2}$$

where a_1, a_2 and a_3 are unknown parameters, and b is the unknown control gain. Here, in fact, the output, y, is the position, x_2 is the velocity, and x_3 is the acceleration of the payload. The point to be noticed here is that it is very difficult to derive the exact value of these parameters due to the manufacture and other design reasons. Therefore, it is very difficult to achieve desired performance for the methods which need to know the exact parameters.

For tracking control of pneumatic position servo system (2), the control goal is to make the output y track the reference signal y_m asymptotically. Before designing the controller, we give the following assumptions.

Assumption 1: The sign of b is known.

Assumption 2: The reference signal y_m, and its up to n-th order derivatives are piecewise continuous and bounded, where n is the order of system (2).

3 Adaptive Slide Mode Controller Design Based on Backstepping Method

An adaptive backstepping sliding mode controller is designed for system (2) as follows.

Define the general error variables as

$$\begin{cases} z_1 = x_1 - y_m, \\ z_2 = x_2 - \alpha_1, \\ z_3 = x_3 - \alpha_2, \end{cases} \tag{3}$$

where α_1 and α_2 are virtual control variables.

Step 1: The time derivative of z_1 is

$$\dot{z}_1 = \dot{x}_1 - \dot{y}_m = x_2 - \dot{y}_m . \tag{4}$$

The first Lyapunov function is defined as

$$V_1 = \frac{1}{2} z_1^2 , \tag{5}$$

whose derivative is

$$\dot{V}_1 = z_1 \dot{z}_1 = z_1 (x_2 - \dot{y}_m) . \tag{6}$$

Select virtual control α_1 as

$$\alpha_1 = \dot{y}_m - c_1 z_1 , \tag{7}$$

where $c_1 > 0$ is a design parameter. Then the time derivative of V_1 becomes

$$\dot{V}_1 = z_1 z_2 - c_1 z_1^2 . \tag{8}$$

If $z_2 = 0$, then $\dot{V}_1 = -c_1 z_1^2$ and z_1 are guaranteed to converge to zero asymptotically.

Step 2: The time derivative of z_2 is

$$\dot{z}_2 = x_3 - \dot{\alpha}_1 . \tag{9}$$

The second Lyapunov function is defined as

$$V_2 = V_1 + \frac{1}{2} z_2^2 , \tag{10}$$

whose derivative is

$$\dot{V}_2 = \dot{V}_1 + z_2 \dot{z}_2 = z_1 z_2 - c_1 z_1^2 + z_2 (x_3 - \dot{\alpha}_1) . \tag{11}$$

Select virtual control α_2 as

$$\alpha_2 = \dot{\alpha}_1 - c_2 z_2 - z_1 , \tag{12}$$

where $c_2 > 0$ is a design parameter. Then the time derivative of V_2 becomes

$$\dot{V}_2 = z_2 z_3 - c_1 z_1^2 - c_2 z_2^2 . \quad (13)$$

If $z_3 = 0$, we have $\dot{V}_2 = -c_1 z_1^2 - c_2 z_2^2$, and thus both z_1 and z_2 are guaranteed to converge to zero asymptotically.

Step 3: Define the sliding surface as

$$s = k_1 z_1 + k_2 z_2 + z_3 , \quad (14)$$

where $k_1 > 0$, $k_2 > 0$ are design parameters.

The third Lyapunov function is defined as

$$V_3 = V_2 + \frac{1}{2} b s^2 \geqslant 0 . \quad (15)$$

Define $\tau_1 = ba_1$, $\tau_2 = ba_2$, $\tau_3 = ba_3$, then the adaptive sliding mode controller is given by

$$u = -\hat{b}k_1(x_2 - \dot{y}_m) - \hat{b}k_2(x_3 - \dot{\alpha}_1) - \hat{\tau}_1 x_1 - \hat{\tau}_2 x_2 - \hat{\tau}_3 x_3 + \hat{b}\dot{\alpha}_2 - c_3 s - c_4 \,\mathrm{sgn}(s), \quad (16)$$

where $c_3 > 0$, $c_4 > 0$ are design parameters.

The adaptive law of unknown parameters is

$$\begin{cases} \dot{\hat{\tau}}_1 = (1/\beta_1) s x_1, \\ \dot{\hat{\tau}}_2 = (1/\beta_2) s x_2, \\ \dot{\hat{\tau}}_3 = (1/\beta_3) s x_3, \\ \dot{\hat{b}} = (1/\lambda)[k_1(x_2 - \dot{y}_m) + k_2(x_3 - \dot{\alpha}_1) - \dot{\alpha}_2] s, \end{cases} \quad (17)$$

where $\beta_i > 0 (i = 1,2,3)$ and λ are adaptive gains. $\hat{\tau}_1$, $\hat{\tau}_2$, $\hat{\tau}_3$ and $\hat{b}$ are the estimated value of τ_1, τ_2, τ_3 and b.

4 Stability Analysis of the Control System

A theorem about stability of the close-loop system (2) with controller (16) and (17) is given as follows.

Theorem 1: For the system (2), if the controller design parameters satisfy the following inequality:

$$\begin{aligned} & c_i > 0 (i = 1,2,3), k_1 > 0, k_2 > 0, \\ & c_1 c_2 + c_1 c_3 k_2^2 + c_2 c_3 k_1^2 > 0, \\ & c_1 c_2 c_3 + c_1 c_3 k_2 - (c_1 + c_3 k_1^2)/4 > 0. \end{aligned} \quad (18)$$

The output y can track the reference signal y_m, and the system is asymptotically stable.

Proof of theorem 1

The time derivative of s is

$$\dot{s} = k_1(x_2 - \dot{y}_d) + k_2(x_3 - \dot{\alpha}_1) + a_1 x_1 + a_2 x_2 + a_3 x_3 + (1/b)u - \dot{\alpha}_2 . \quad (19)$$

Then, the time derivative of V_3 is

$$\dot{V}_3 = \dot{V}_2 + bs\dot{s} = -c_1 z_1^2 - c_2 z_2^2 + z_2 z_3 + s[bk_1(x_2 - \dot{y}_m) = +bk_2(x_3 - \dot{\alpha}_1) + b(a_1 x_1 + a_2 x_2 + a_3 x_3) + u - b\dot{\alpha}_2]. \quad (20)$$

Rewrite Eq. (20) as

$$\dot{V}_3 = -c_1 z_1^2 - c_2 z_2^2 + z_2 z_3 + s[bk_1(x_2 - \dot{y}_m) + bk_2(x_3 - \dot{\alpha}_1) + \tau_1 x_1 + \tau_2 x_2 + \tau_3 x_3 + u - b\dot{\alpha}_2]. \quad (21)$$

Assume the system Lyapunov function to be

$$V = V_3 + \frac{1}{2}\lambda_1 \tilde{\tau}_1^2 + \frac{1}{2}\lambda_2 \tilde{\tau}_2^2 + \frac{1}{2}\lambda_3 \tilde{\tau}_3^2 + \frac{1}{2}\lambda_4 \tilde{b}^2 \geqslant 0, \quad (22)$$

where $\tilde{\tau}_1 = \tau_1 - \hat{\tau}_1$, $\tilde{\tau}_2 = \tau_2 - \hat{\tau}_2$, $\tilde{\tau}_3 = \tau_3 - \hat{\tau}_3$, and $\tilde{b} = b - \hat{b}$ are the parameters estimation error.

The time derivative of V is

$$\dot{V} = -c_1 z_1^2 - c_2 z_2^2 + z_2 z_3 + s[bk_1(x_2 - \dot{y}_m) + bk_2(x_3 - \dot{\alpha}_1) + \tau_1 x_1 + \tau_2 x_2 + \tau_3 x_3 + u - b\dot{\alpha}_2] + \lambda_1 \tilde{\tau}_1(-\dot{\hat{\tau}}_1) + \lambda_2 \tilde{\tau}_2(-\dot{\hat{\tau}}_2) + \lambda_3 \tilde{\tau}_3(-\dot{\hat{\tau}}_3) + \lambda_4 \tilde{b}(-\dot{\hat{b}}). \quad (23)$$

Substituting Eq. (16) into Eq. (23) obtains

$$\dot{V} = -c_1 z_1^2 - c_2 z_2^2 + z_2 z_3 - c_3 s^2 - c_4 |s| + \tilde{b}[sk_1(x_2 - \dot{y}_m) + sk_2(x_3 - \dot{\alpha}_1) - s\dot{\alpha}_2 - \lambda\dot{\hat{b}}] + \tilde{\tau}_1(sx_1 - \beta_1\dot{\hat{\tau}}_1) + \tilde{\tau}_2(sx_2 - \beta_2\dot{\hat{\tau}}_2) + \beta_3\tilde{\tau}_3(sx_3 - \beta_3\dot{\hat{\tau}}_3). \quad (24)$$

Substituting Eq. (17) into Eq. (24) obtains

$$\dot{V} = -c_1 z_1^2 - c_2 z_2^2 + z_2 z_3 - c_3 s^2 - c_4 |s| = -\boldsymbol{E}^{\mathrm{T}}\boldsymbol{Q}\boldsymbol{E} - c_4 |s|, \quad (25)$$

where $\boldsymbol{E} = [z_1, z_2, z_3]^{\mathrm{T}}$.

$$\boldsymbol{Q} = \begin{pmatrix} c_1 + c_3 k_1^2 & k_1 k_2 c_3 & k_1 c_3 \\ k_1 k_2 c_3 & c_2 + c_3 k_2^2 & k_2 c_3 - 1/2 \\ k_1 c_3 & k_2 c_3 - 1/2 & c_3 \end{pmatrix}. \quad (26)$$

If the controller parameters $k_1, k_2, c_i (i = 1,2,3)$ satisfy Eq. (18), then matrix $\boldsymbol{Q}$ is a positive definite.

Define $\boldsymbol{W}=\boldsymbol{E}^{\mathrm{T}}\boldsymbol{QE}$. From Eq. (25), it is known that $\dot{V}\leqslant -\boldsymbol{W}$, so

$$\lim_{t\to\infty}\int_0^t \boldsymbol{W}\mathrm{d}\tau \leqslant V(z_1(0),z_2(0),z_3(0))-V(z_1(\infty),\\ z_2(\infty),z_3(\infty)). \tag{27}$$

Therefore, $\lim_{t\to\infty}\int_0^t \boldsymbol{W}\mathrm{d}\tau$ exists.

If $k_1,k_2,c_i(i=1,2,3)$ are selected in order to ensure $\boldsymbol{Q}$ to be the positive definite matrix and then $\dot{V}\leqslant 0$, it is proven that the system is stable according to the Lyapunov stability theory.

To this end, it is concluded that V is bounded, according to Eq. (27); z_1, z_2, z_3 and $\tilde{\tau}_1$, $\tilde{\tau}_2$, $\tilde{\tau}_3$, $\tilde{b}$ are bounded as well. Since $\lim_{t\to\infty}\int_0^t \boldsymbol{W}\mathrm{d}\tau$ exists, it is concluded that $\lim_{t\to\infty}\boldsymbol{W}=0$ according to the Barbalat theorem. It infers that $z_i\to 0(i=1,2,3)$ as $t\to\infty$. Similarly, $\lim_{t\to\infty}s=0$, the position tracking error of the system is convergent. So the close-loop system is asymptotically stable; the system can achieve asymptotic tracking to the reference signal. The system state will tend to the sliding surface $s=0$ within a limited time period.

5 Experimental Results

5.1 Experimental platform

The experimental hardware consists of a double-acting rodless cylinder with a 25-mm diameter bore and a 450-mm stroke(Festo, model: DGPL-25-450-PPV-A-B-KF–GK-SV), a five-way proportional valve(Festo, model: MPYE-5-1/8-HF-010-B), a potentiometer(Festo, model: MLO-POT-450-TLF), a payload, an air pump, a pressure regulator, and a computer with a general purpose signal conversion card(including ADC and DAC). The photo of the experimental devices is shown in Fig. 2.

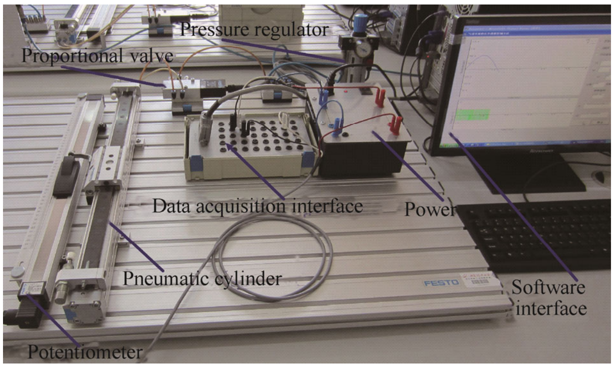


Fig. 2. Photo of the pneumatic position servo system

The controller algorithm is developed using Visual Basic software. A user interface is designed on the computer screen, within which the payload and the reference positions are displayed. The input type selection, parameters setting, etc. can also be done in the interface.

In order to reduce energy consumption, the controller output is saturated within the limited range $[U_{\min},U_{\max}]$. The payload position $y=x_1$ is measured by the analog sensor directly. The sampling time of the controller is 10ms. Before the experiment, the slider is forced to reach the middle point of the chamber.

Define three types of reference signals as follows,

(1) Reference signal 1 is a sinusoidal signal given as

$$y_m = A_1\sin(\omega_1 t), \tag{28}$$

where $A_1=167.475$ and $\omega_1=0.5\pi$.

(2) Reference signal 2 is a s-curve signal given as

$$y_m=-(A_2/\omega_2{}^2)\sin(\omega_2 t)+(A/\omega_2)t, \tag{29}$$

where $A_2=55.825$ and $\omega_2=0.5\pi$.

(3) Reference signal 3 is a multi-frequency sinusoidal signal given as

$$y_m = A_3*[\sin(\pi t)+\sin(0.5\pi t)+\sin(2\pi t/7)+\\ \sin(2\pi t/12)+\sin(2\pi t/17)], \tag{30}$$

where $A_3=167.475$.

5.2 Experiment results of the proposed method

Set the controller parameters $c_1=c_2=60$, $c_3=0.01$, $k_1=k_2=1$, $\beta_1=\beta_2=\beta_3=1$, and $\lambda=1$. Pneumatic position servo system is controlled to track the three types of reference signals respectively. The corresponding experimental results are shown in Figs. 3 (a), (b) and (c).

In the Fig. 3, the black dash line is a reference signal, the blue solid line is the actual output of the system. It is known from Fig. 3 that the output of a pneumatic position servo system can track the reference signal with high precision.

5.3 Comparison to some existing methods

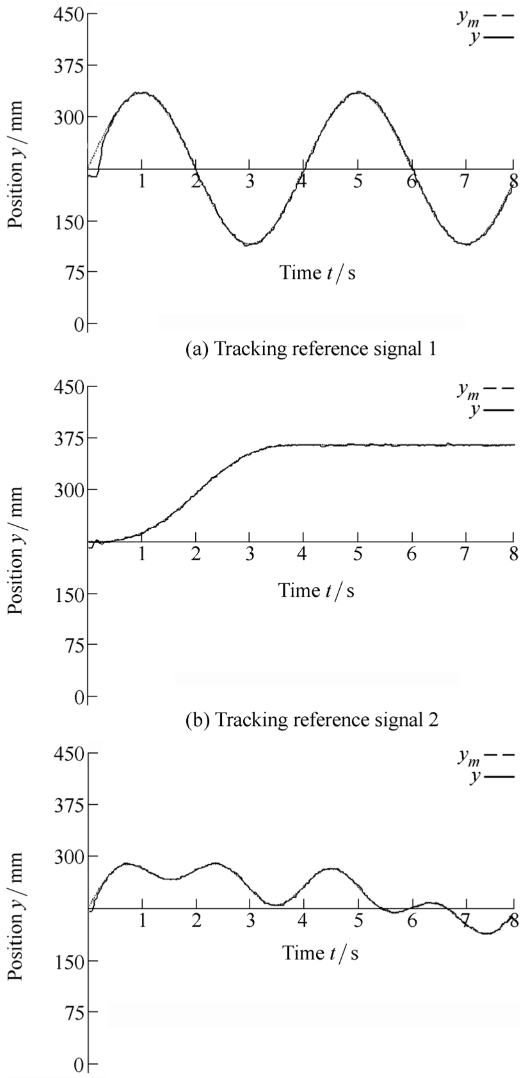
Comparison of the proposed method to the methods in Refs. [11], [12], [14], [16] and [21] is conducted for the three reference inputs, respectively. The controller parameters in Ref. [11] are given as

$$c_1=c_2=50,\ \lambda=1,\ \boldsymbol{\Gamma}=\mathrm{diag}[1\quad 1\quad 1].$$

The controller parameters in Ref. [12] are given as

$$c_1=c_2=50,\ \lambda=\beta_0=\beta_1=\beta_2=1.$$

The controller parameters in Ref. [14] are given as



(a) Tracking reference signal 1

(b) Tracking reference signal 2

(c) Tracking reference signal 3

Fig. 3.　Experimental results using the proposed method

$$\omega = 50,\ \ \xi = 1,\ \ k_{s2} = 1.56(v).$$

The controller parameters in Ref. [16] are given as

$$n_2 = 29.5,\quad n_1 = 218.43,\quad n_0 = 0,\quad m_0 = 5531.3,$$
$$\lambda = 50,\quad k_{s1} = 2440,\quad \phi = 0.05.$$

The controller parameters in Ref. [21] are given as

$$\beta_3 = 3,\quad \beta_2 = 3,\quad \beta_1 = 1,\quad \eta_0 = 5,$$
$$\sigma = 0.1,\quad K = 4,\quad K_0 = 20.$$

These parameters are all dedicatedly adjusted to have nice performance. In order to compare the tracking error of these method quantitatively, a root mean square error is defined as

$$\mathrm{RMSE} = \sqrt{\frac{1}{N}\sum_{k=N_1}^{N_2} {e_k}^2}\ , \tag{31}$$

where $N = N_2 - N_1$, where N_1 is the start time of consideration and N_2 is the end time of consideration, and e_k is the tracking error at k-th time sampling.

To avoid the influence of stochastic factors such as noise or the initial conditions, every experiment was performed five times to get average RMSE. The quantitative comparison results are given in Tables 1 to 3.

Table 1.　RMSE comparison of six methods for tracking reference signal 1　　mm

Method	Test 1	Test 2	Test 3	Test 4	Test 5	Avg.
Ref.[11]	2.319 5	2.405 8	2.364 2	2.388 3	2.411 8	2.377 9
Ref.[12]	2.330 4	2.268 5	2.320 5	2.297 8	2.362 9	2.316 0
Ref.[14]	2.453 2	2.473 1	2.481 9	2.487 8	2.491 2	2.477 4
Ref.[16]	2.531 8	2.458 6	2.479 5	2.524 7	2.488 7	2.496 7
Ref.[21]	2.373 0	2.399 1	2.407 9	2.367 3	2.414 2	2.392 3
Proposed	1.980 3	2.081 0	1.995 4	2.099 5	2.019 0	2.035 0

Table 2.　Comparison of six methods for tracking reference signal 2　　mm

Method	Test 1	Test 2	Test 3	Test 4	Test 5	Avg.
Ref.[11]	0.657 4	0.664 5	0.698 1	0.658 2	0.646 0	0.664 8
Ref.[12]	0.705 5	0.711 9	0.662 1	0.655 0	0.666 2	0.680 1
Ref.[14]	0.810 4	0.793 4	0.852 9	0.825 2	0.811 2	0.818 6
Ref.[16]	0.765 5	0.762 5	0.791 4	0.771 3	0.771 2	0.772 4
Ref.[21]	0.788 4	0.828 5	0.806 0	0.797 6	0.825 3	0.809 2
Proposed	0.626 3	0.627 0	0.582 0	0.608 5	0.581 7	0.605 1

Table 3.　Comparison of six methods for tracking reference signal 3　　mm

Method	Test 1	Test 2	Test 3	Test 4	Test 5	Avg.
Ref.[11]	1.114 9	1.157 9	1.147 4	1.110 2	1.152 5	1.136 6
Ref.[12]	1.135 5	1.166 7	1.127 6	1.174 1	1.141 2	1.149 0
Ref.[14]	1.294 5	1.300 6	1.252 9	1.286 8	1.288 7	1.284 7
Ref.[16]	1.242 3	1.249 9	1.273 0	1.287 0	1.282 2	1.266 9
Ref.[21]	1.206 8	1.186 8	1.193 4	1.218 7	1.216 3	1.204 4
Proposed	1.039 5	1.038 0	1.042 3	1.037 7	1.049 6	1.041 4

6　Conclusions

(1) An adaptive backstepping sliding mode control method is proposed for the pneumatic position servo system. The proposed method designs a sliding mode controller using adaptive backstepping technique based on a linear model of the pneumatic system with undetermined parameters. Therefore, the accurate parameters of the system are not required. Moreover, the uncertain parameter boundary needed by the traditional sliding mode controller is not used. This is fit for the pneumatic system because the system parameters, or the range of the parameters, are unavailable to users in general application cases.

(2) The proposed controller employs the parameters adaptation law designed according to Lyapunov analysis to guarantee the stability of the close loop system and the boundedness of the parameters, which is supported by a theorem.

(3) The proposed controller is simple for implementation because only the payload displacement information is needed in the controller to get the high performance. The control system configuration is simple and inexpensive

because no pressure sensors are needed.

(4) The comparison experimental results show that the proposed method can track different reference signals with the highest tracking accuracy in the six methods.

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