Adaptive Backstepping Control of Pneumatic Servo System

Hai-Peng Ren
Department of Information and Control Engineering
Xi'an University of Technology
Xi'an, China
E-mail renhaipeng@xaut.edu.cn

Chao Huang
Department of Information and Control Engineering
Xi'an University of Technology
Xi'an, China
E-mail huangchao823@163.com

***Abstract*—Because of its high power-to-weight ratio, safe, clean and other advantages, pneumatic servo systems are widely applied in the factory automation and the other applications. However, due to the compressibility of gas and friction, the high precision control is a challenge and key problem of pneumatic servo system. In this paper, an adaptive backstepping controller is designed to control rodless cylinder using the proportional valve. The method has no requirement of the pressure sensors and the derivative calculation of the reference signal. The controller can be designed without the prior knowledge about the system model and the boundary of the model uncertainty. The controlled experiments are performed, the results show that the proposed method achieves better precision compared to some slide mode controllers.**

***Keywords*—pneumatic system, position tracking, backstepping design, parameter adaption, slide mode control**

I. INTRODUCTION

Using the compressed air as the working medium, the pneumatic actuators have the advantages of long operating life, safety, cleanliness, high power-to-weight ratio and so on. The pneumatic technology has been widely applied to various fields. In particular, the pneumatic technology plays an irreplaceable role in the mobile robot, mechanical arm, and other machinery and automation fields. With the rapid development of the pneumatic components and the microprocessor, the better actuators and the faster controller with low cost are available for the performance improvement of the pneumatic system. As a consequence, these give more application prospects to the pneumatic servo system. But the compressibility of air and the friction force make it difficult to achieve accurate position control of a pneumatic actuator. Using high performance method based on the special hardware to improve the pneumatic servo system precision becomes a new trend of the pneumatic technology.

Reference [2] implements PID control in the pneumatic system based on the switch valve. Reference [3] proposes a PID controller with a feedback of the pressure difference between two chambers to improve the tracking performance. Pu *et al*. has established the state space model of the pneumatic system, and implemented a state feedback controller to get better tracking performance[4]. Yin *et al.* proposes a feedback linearization method to get better performance comparing to the ordinary PID controller. Both the method in [3] and [4] has the disadvantage of requiring all states measurement and feedback, which increases the cost of the control system. Model reference adaptive control methods are also proposed to use in the pneumatic servo systems in [6-7]. As a robust method to deal with the model uncertainty of the system, slide mode controllers (SMC) have been applied to the pneumatic servo system, which is widely considered to be a suitable method for pneumatic system[8-12]. Reference [10] designs a SMC using the third-order linear and nonlinear model, respectively, to get the expected results. But this method is not robust to the payload variations. Reference [11] designs a SMC based on the second-order slide surface, and uses the sign of the surface to switch the on-off valve, therefore, simplifies the controller structure. In addition, the authors proposed a scheme to reduce the switch times with the cost of the a bit larger tracking error. But this method also needs pressure sensor. Reference [12] proposes a SMC based on the pressure observer to simplify the controller structure and to reduce the cost. It has limited tracking precision. Reference [13] establishes a multi-input-multi-output model including four proportional valves, and designs a backstepping controller. This controller can stabilize the system under the condition of all parameters being known. If the parameters change, the stability cannot be guaranteed.

In this paper, we propose an adaptive backstepping design methodology for the pneumatic position servo system using proportional valve. The method can implement tracking control without prior knowledge of the pneumatic system model. By this way, the complicated trail procedure for adjusting PID parameters is avoided. No pressure sensors or the calculation of the reference derivative is needed. Comparison with two SMCs in [10] and [11], the method proposed in this paper has a better tracking precision.

II. PNEUMATIC POSITION SYSTEM ADAPTIVE BACKSTEPPING TRACKING CONTROLLER DESIGN

The working principle of pneumatic servo system is shown in Fig. 1. The pump is used to provide the compressed air, which flows into the left side or the right side of the cylinder through the regulation of a proportional valve. The input voltage of the proportional valve adjusts the mass flow into each side of the cylinder in order to obtain the pressure

Support in part by Specialized Research Fund for the Doctoral Program of Higher Education (No. 20126118110008), Shaanxi KJXX project (No. 2011kjxx18).

difference between two cylinder sides that drives the piston and payload. The displacement of the payload is detected by the potentiometer, which converts the displacement into 0-10V voltage signal. The displacement signal is then fed into the computer through a general purpose analog signal converter card (including A/D converter and D/A converter). The control signal is given to the proportional valve through D/A converter, thus, a feedback control can be implemented based on such a hardware configuration.

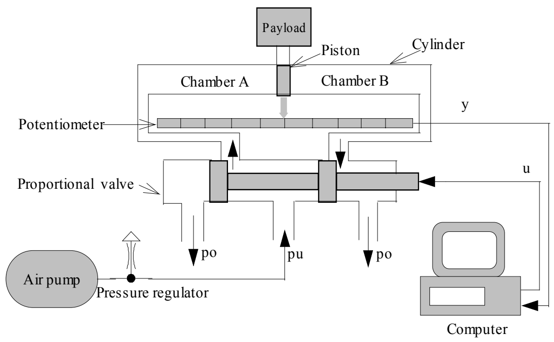


Fig 1.System structure

The dynamic model of the pneumatic servo system is given as follows[3,5,10]

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

where m_a and m_b are the mass flow rates into the chamber A and chamber B, respectively. p_a and p_b are the pressures of the chamber A and chamber B, respectively. A_a and A_b are the cross-section area corresponding to the two sides of the piston. 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, $f_a(.)$ and $f_b(.)$ are nonlinear functions of the upper stream and lower stream pressure of the chamber A and chamber B, K,R and T are the constants, u is the input voltage of the proportional valve. Here, we ignore the dynamics of the proportional valve.

By ignoring the friction, and linearizing nonlinear function $f_a(.)$ and $f_b(.)$, we obtain a third-order linear model of the pneumatic system as follows

$$\dddot{y} + a_2\ddot{y} + a_1\dot{y} + a_0 y = b_0 u, \tag{2}$$

where a_0, a_1, a_2 and b_0 are unknown parameters.

Rewrite (2) as

$$\begin{cases} \dot{x}_1 = x_2 \\ \dot{x}_2 = x_3 \\ \dot{x}_3 = -a_0 x_1 - a_1 x_2 - a_2 x_3 + b_0 u \\ y = x_1 \end{cases}. \tag{3}$$

The control objective is to make the displacement of the payload and piston, y, track the desired reference input.

Define the following error variables

$$\begin{cases} z_1 = x_1 - y_m \\ z_2 = x_2 - \alpha_1 \\ z_3 = x_3 - \alpha_2 \end{cases}, \tag{4}$$

where α_1, α_2 are virtual control variables.

We design a backstepping control for the pneumatic actuator as follows:

Step 1, select the first Lyapunov function

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

The derivative of the first Lyapunov function is

$$\dot{V}_1 = z_1 z_2 + \alpha_1 z_1 - \dot{y}_m z_1. \tag{6}$$

Design the first virtual control as

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

where $c_1 > 0$ is the constant design parameter.

Substituting (7) into (6), we obtain

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

If $z_2 = 0$, then the first subsystem is stable.

Step 2, select the second Lyapunov function

$$V_2 = V_1 + \frac{1}{2} z_2^2. \tag{9}$$

The derivative of the second Lyapunov function is

$$\dot{V}_2 = z_1 z_2 - c_1 z_1^2 + z_2 z_3 + z_2 \alpha_2 - z_2 \dot{\alpha}_1. \tag{10}$$

Design the second virtual control as

$$\alpha_2 = -z_1 + \dot{\alpha}_1 - c_2 z_2, \tag{11}$$

where $c_2 > 0$ is the constant design parameter.

By substituting (11) into (10), we obtain

$$\dot{V}_2 = z_2 z_3 - c_1 z_1^2 - c_2 z_2^2. \tag{12}$$

If $z_3 = 0$, then the subsystem consisting of two variables z_1 and z_2 is stable.

Step3, select the third Lyapunov function as

$$V_3 = V_2 + \frac{1}{2} z_3^2. \tag{13}$$

The derivative of the third Lyapunov function is

$$\begin{aligned}\dot{V}_3 = -c_2 {z_2}^2 - c_1 z_1^2 + z_2 z_3 - a_0 z_1 z_3 - a_0 y_m z_3 - \\ a_1 z_2 z_3 - a_1 \alpha_1 z_3 - a_2 z_3 z_3 - \\ a_2 \alpha_2 z_3 + b_0 u z_3 - \dot{\alpha}_2 z_3\end{aligned} . \tag{14}$$

If the parameters of the system are known, then by using the controller

$$\begin{aligned}u = \frac{1}{b_0}(-z_2 + a_0 z_1 + a_0 y_m + a_1 z_2 + \\ a_1 \alpha_1 + a_2 \alpha_2 + \dot{\alpha}_2 - c_3 z_3 - a_2 z_3)\end{aligned} , \tag{15}$$

then we have

$$\dot{V}_3 = -c_2 {z_2}^2 - c_1 z_1^2 - c_3 z_3^2 .$$

Therefore, the whole system is stable.

If the parameters are unknown, using the estimated variables to replace the known parameters, we obtain the adaptive controller as follows

$$\begin{aligned}u = \frac{1}{\hat{b}_0}(-z_2 + \hat{a}_0 z_1 + \hat{a}_0 y_m + \hat{a}_1 z_2 + \\ \hat{a}_1 \alpha_1 + \hat{a}_2 \alpha_2 + \dot{\alpha}_2 - \hat{c}_3 z_3 - \hat{a}_2 z_3)\end{aligned} . \tag{16}$$

The adaptive law of the unknown parameters is

$$\begin{cases}\dot{\hat{b}}_0 = (-\lambda) z_1 y_m \hat{b}_0^2 \\ \dot{\hat{a}}_0 = \beta_0 z_1 x_1 \\ \dot{\hat{a}}_1 = \beta_1 z_1 x_2 \\ \dot{\hat{a}}_2 = \beta_2 z_1 x_3\end{cases} , \tag{17}$$

where $\lambda > 0$, $\beta_i > 0, i = 0,1,2$ are the positive adaption gains.

III. EXPERIMENTAL RESULTS

A. *Experimental setup*

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, a air pump, a pressure regulator, and a computer with a general purpose analog signal converter card (including A/D converter and D/A converter). The photo of the experimental devices is shown in Fig. 2.

A computer operation interface is programmed using the Visual Basic development environment, which can be used to display position response curve, $y(t)$, the desired position curve, $y_m(t)$, the curve of the manipulated variable, u(t), and to select the type and parameters of the reference signal, *etc*.

B. *Experimental Results*

In the experimental setup, the adaptive backstepping controller in Eqs. (16) and (17) is used. The derivative operations are implemented using Eular equation. The controller parameters $c_1 = c_2 = 30$, the adaptive gains $\lambda = \beta_0 = \beta_1 = \beta_2 = 1$. Since the input voltage of the proportional valve has a 5.1685 V bias voltage, the controller output is added the bias voltage. In order to decrease the energy consumption, the controller output is limited in a range of [-U_{max}, U_{max}]. The reference output zero is set to 236.2mm.

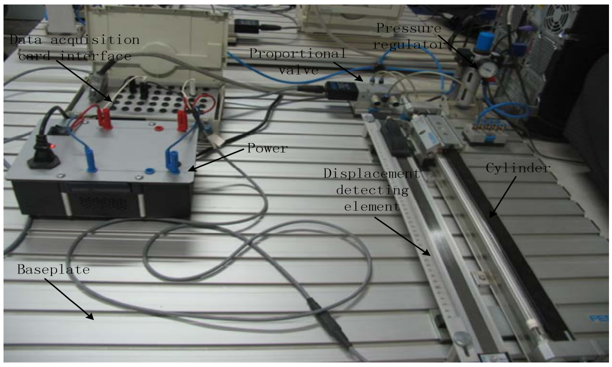


Fig. 2 Experiment equipments

When the reference output is S-curve given by

$$y_m(t) = -(A/\omega^2)\sin(\omega t) + (A/\omega)t , \tag{18}$$

where $A = 55.825, w = 0.5\pi$, the experimental result of the proposed method is shown in Fig.3 (a).

When the reference input is sinusoidal signal given by

$$y_m(t) = 111.65 \sin 0.5\pi t , \tag{19}$$

the result of the proposed method is shown in Fig. 3 (b).

When the reference input is a multi-frequency sinusoidal signal given by

$$\begin{aligned}y_m(t) = 167.475 \sin \pi t + 167.475 \sin 0.5\pi t + \\ 167.475 \sin(2\pi t/7) + 167.475 \sin(\pi t/6) , \\ +167.475 \sin(2\pi t/17)\end{aligned} \tag{20}$$

The result of the proposed method is shown in Fig.3 (c). In Fig.3, the references are given in black dash line, while the experimental outputs are given in dark blue solid line. From the experimental results, the adaptive backstepping controller shows a very nice performance.

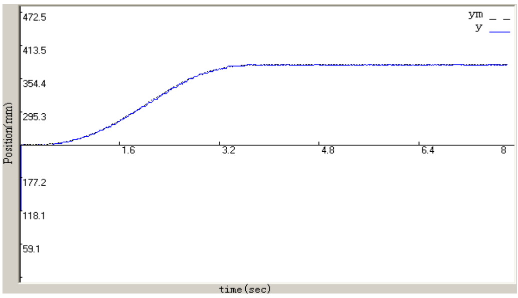
C. *The experimental comparison with the variable structure control method*

Sliding Mode Controller (SMC) is fit for dealing with the uncertainty, as a consequence, many research have been done to use SMC for the pneumatic servo system[8-13]. We design a controlled experiment to compare the results of the proposed method in this paper with two recent SMCs in [10] and [11]. The experimental results are given in Figs. 4 and 5.

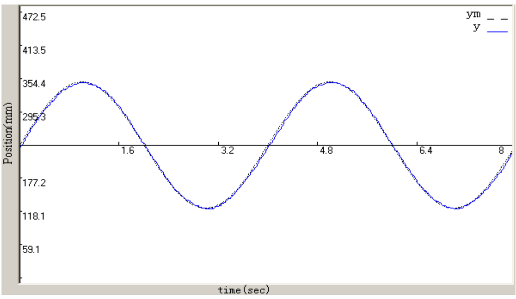
In Fig.4, the controller designed in [10] is used with parameters $a_0 = 0$, $a_1 = 218.436$, $a_2 = 29.5544$,

$b_0 = 5531.3305$, $\lambda = 50$, $k_{s1} = 2.44\times10^4$. In Fig. 4 (a), the reference output is given by Eq.(18), in Fig. 4 (b), the reference output is given by Eq.(19), and in Fig. 4 (c), the reference output is given by Eq.(20), respectively.

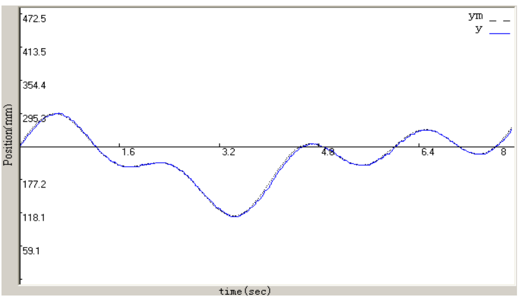
In [11], the controller parameters are given by $k_{s2} = 1$, then $u = 1$ is corresponding to the open valve, and $u = -1$ is corresponding to the closed valve. The experimental setup in this paper uses the proportional valve, so we use $k_{s2} = 1.56$ (v) to control the proportional valve. That means u=1.56V or u =-1.56V according to the states with respect to the sliding surface[11]. The controller parameters in Fig.5 are given as $\xi = 1$, $\omega = 50$. In Fig. 5 (a) , the reference output is given by Eq.(18), in Fig. 5 (b), the reference output is given by Eq.(19), and in Fig. 5 (c), the reference output is given by Eq.(20), respectively.



(a) the output and reference for tracking the S-curve



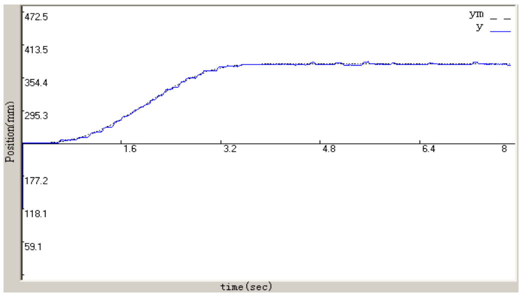
(b) the output and reference for tracking the sinusoidal signal



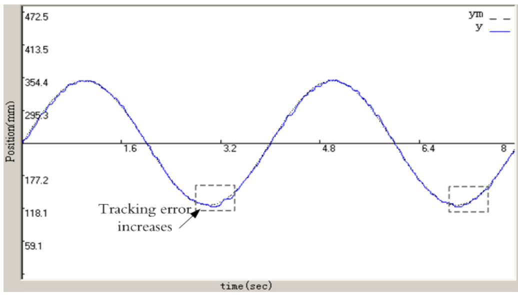
(c) the output and reference for tracking the multi-frequency sinusoidal signal

Fig. 3 the experiment results using the proposed method

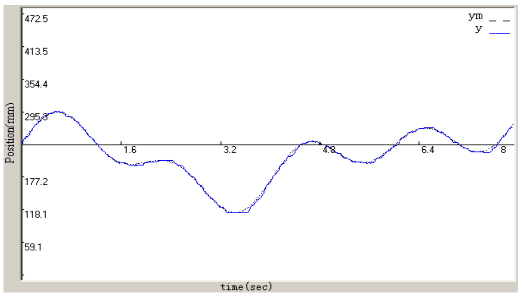
Comparing Fig.3 (a) with Fig.4 (a) and Fig.5 (a), we can see that the proposed method in this paper, shown in Fig.3 (a), can track the S-curve reference more smoothly than the SMC methods in [10] and [11], shown in Fig. 4 (a) and Fig.5 (a) respectively. Comparing Fig. 3 (b) and Fig.3 (c) with Fig.4 (b), Fig.4 (c), Fig.5 (b), and Fig.5 (c), we have that the method proposed in this paper, shown in Fig.3 (b) and Fig. 3 (c), tracks the sinusoidal and multi-frequency sinusoidal reference more smoothly than the SMC methods in [10] and [11] , shown in Fig. 4 (b) Fig.4 (c), Fig.5 (b), and Fig.5 (c), correspondingly. It should be noticed that the tracking errors of SMCs, shown in Fig.4 (b), Fig.4 (c), Fig.5 (b) and Fig.5 (c), are larger than that of the method proposed in this paper, especially near the point where the derivative of the reference output changes its sign, marked in rectangular box in the corresponding figure. In these locations, the Coulomb friction force between the piston and cylinder become active. What is the reason why the method proposed in this paper can overcome this force is not clear now. To explain this reason is a remarkable work, out of the scope of this paper. But, by checking the control force of the proposed method, we find that the control force is a kind of bang-bang control, i.e., the control force oscillates between [$-U_{\max}$, $U_{\max}$] with high frequency. It might be the reason that the high frequency oscillation of the control force lubricates the piston and cylinder, thus, makes the Coulomb friction effect weaken to get better tracking performance.



(a) the output and reference for tracking the S-curve



(b) the output and reference for tracking the sinusoidal signal



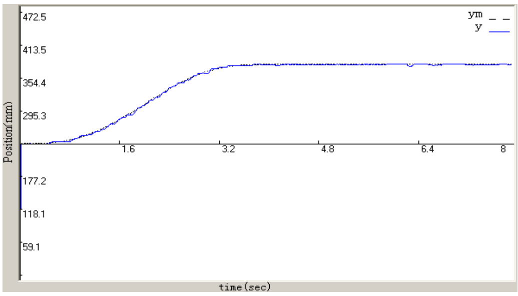
(c) the output and reference for tracking the multi-frequency sinusoidal signal

Fig. 4 Experimental results using the method in [10]

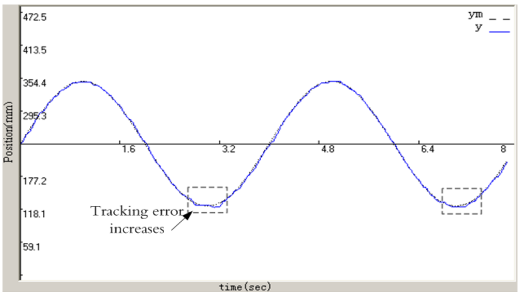
In order to quantitatively compare the tracking performance of the three control methods, we define the Root Mean Square Error (RMSE) as follows

$$\text{RMSE} = \sqrt{\frac{1}{N_2 - N_1 + 1}\sum_{k=N_1}^{N_2} e_k^2}\,, \tag{21}$$

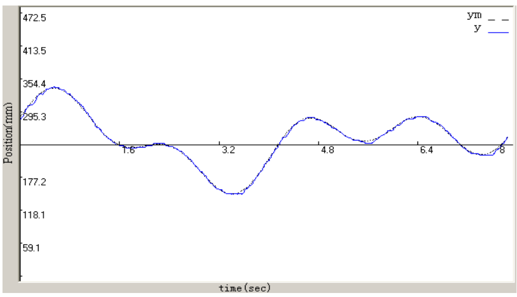
where N_1 and N_2 are the start and the end of the sampling points to be considered, $e_k = y_m(k\Delta T) - y(k\Delta T)$, ΔT is the sampling interval and k is sampling time.



(a) the output and reference for tracking the S-curve



(b) the output and reference for tracking the sinusoidal signal



(c) the output and reference for tracking the multi-frequency sinusoidal signal

Fig. 5 Experimental results using the method in [11]

The RMSEs for different methods and different references are given in Tables 1-3. Table 1 is dedicated to the S-curve reference signal, Table 2 is dedicated to the sinusoidal reference signal, and Table 3 is dedicated to the multi-frequency sinusoidal reference signal. In these tables, five experiments are conducted for each method with the same reference input to avoid stochastic factor in the experiments. The results of these five time experiments are given sequentially from column 2 to column 6, indexed from ***test 1*** to ***test5***, in the corresponding row of the each table. The mean value of the five time EMSEs, indexed by ***avg.***, is given in the last column of the corresponding row in each table. From Tables 1-3, we have that the proposed method possesses the minimum average RMSE for all reference input cases, and the average RMSE of the proposed method is significantly smaller than the other two methods.

From the controlled experiment results given in Figs.3-5 and the quantitative comparisons given in Tables 1-3, we have the conclusion that the proposed method in this paper has better tracking performance comparing with the SMCs in [10] and [11].

Table 1 Tracking RMSE of the S-curve reference signal

	Tracking RMSE(mm)					
	Test 1	***Test 2***	***Test 3***	***Test 4***	***Test 5***	***Avg.***
Proposed mehtod	0.5677	0.6299	0.6201	0.5901	0.6399	0.6095
Method in [10]	0.9236	0.9012	0.8633	1.0267	0.9300	0.9290
Method in [11]	0.9205	1.0160	1.0825	1.0453	1.0380	1.0205

Table 2 Tracking RMSE of the sinusoidal signal

	Tracking RMSE(mm)					
	Test 1	***Test 2***	***Test 3***	***Test 4***	***Test 5***	***Avg.***
Proposed mehtod	1.8750	1.9110	2.0278	1.9756	2.0100	1.9600
Method in [10]	2.8608	2.6066	2.6794	2.6156	2.6896	2.6904
Method in [11]	2.7450	2.6097	2.7333	2.7803	2.8017	2.7340

Table 3 Tracking RMSE of the multi-frequency sinusoidal signal

	Tracking RMSE(mm)					
	Test 1	***Test 2***	***Test 3***	***Test 4***	***Test 5***	***Avg.***
Proposed mehtod	1.1872	1.4742	1.4911	1.4722	1.4205	1.4090
Method in [10]	2.2265	2.1341	2.2206	2.1154	2.0451	2.1483
Method in [11]	2.3604	2.2020	2.1052	1.9833	2.2125	2.1727

IV. CONCLUSION

An adaptive backstepping controller for the position tracking control of the pneumatic servo system is designed in this paper. The controller can track three typical reference signals with higher accuracy compared to two SMCs. Although the controller is designed using the linear model of the pneumatic servo system, the Coulomb friction force does not affect the tracking error significantly. The main features of this method are, firstly, that one can design an adaptive backstepping controller without the knowledge of the actual model of the pneumatic system; secondly, that one can design the controller without the derivative information of the reference signal; thirdly, that one can implement the controller without the expensive pressure sensors or the comprehensive pressure observers, reducing the cost of the control system and simplifying the controller structures. The controller has a good practical application prospects.

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