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Adaptive Backstepping Control of a Pneumatic System With Unknown Model Parameters and Control Direction

HAI-PENG REN[1], (Member, IEEE), XUAN WANG[1], JUN-TAO FAN[1], AND OKYAY KAYNAK[2], (Fellow, IEEE)
[1]Shaanxi Key Laboratory Complex System Control and Intelligent Information Processing, Xi'an University of Technology, Xi'an 710048, China
[2]Department of Electrical and Electronic Engineering, Boğaziçi University, 34342 Istanbul, Turkey

Corresponding authors: Hai-Peng Ren (renhaipeng@xaut.edu.cn) and Okyay Kaynak (okyay.kaynak@boun.edu.tr)

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ABSTRACT The performance of the pneumatic elements and the micro-controller steadily increases as the price of them decreases. This trend promotes high performance tracking control research on pneumatic servo systems. However, it is very difficult to obtain exact model parameters, which is one of the main obstacles to design a high performance controller. Moreover, in some application cases, the control direction is undetermined because of the possible false or incorrect valve outlets connection. At the same time, the inaccuracy of the proportional valve zero points also degrades the tracking performance of pneumatic servo systems. In this paper, an adaptive backstepping control approach is proposed for a pneumatic position servo system considering unknown model parameters, unknown control direction, and inaccurate valve zero points. The proposed method combines the Nussbaum function and backstepping technique to design a position tracking controller of a pneumatic servo system with the aforementioned uncertainties. By using the Lyapunov method, the designed controller is proved to be stable, and the tracking error asymptotically converges to zero. The experimental results demonstrate the effectiveness and the superiority of the proposed approach as compared with some existing methods, even for negative direction case for which the other methods fail. The performance of the proposed controller is further improved by considering the proportional valve zero points.

INDEX TERMS Pneumatic system, unknown model parameters, inaccurate proportional valve zero point, Nussbaum function, unknown control direction.

I. INTRODUCTION

Pneumatic technology is widely used in various automation fields, due to its low-cost, cleanness, and high power-to-weight ratio properties [1]. With the technology progress, the performance of pneumatic elements, e.g., the proportional valve, is increasing, while the prices of them are decreasing; at the same time, micro-controllers with high performance to price ratio are available for pneumatic servo systems control. These factors promote the high performance pneumatic servo systems research. However, due to the difficulty to obtain the exact parameters of pneumatic servo systems, it is difficult to design a high performance tracking controller. And in some application field, it is an additional obstacle that the designer cannot know the control gain direction for sure, which might be due to the reconnection of the system on site. Moreover, in practical applications, the nominal zero point of the proportional valve might be inaccurate, which will debase the tracking performance due to the integral property of the proportional valve. In such cases, a controller has to be designed without the information of system parameters, control direction and accurate proportional valve zero point. This paper is dedicated to such a case, and gives a design methodology of the tracking controller.

According to the reference signal, pneumatic servo systems can be divided into two categories, positioning control (i.e., step reference) and tracking continuous variant reference. References [2]–[9] were dedicated to positioning control of the pneumatic servo system. In this case, the friction has less impact on the precision than the case of tracking

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control. Proportional-integral-derivative (PID) control was implemented in pneumatic servo systems based on the switch valve [2]. Reference [3] established the state space model of a pneumatic servo system, and implemented a state feedback controller to obtain higher positioning accuracy, as compared to the PID control method. A feedback linearization method was proposed to get better performance compared to the conventional PID controller [4]; however, both methods in [3] and [4] had the disadvantage of requiring all states measurement, especially, the expensive pressure sensor, which increases the structure complexity and the cost of the control system. The predictive functional control was applied for the positioning control and force control of pneumatic servo systems [5], it got good performance. Reference [6] estimated the contraction length and tension of pneumatic artificial muscle (PAM) by using feed-forward neural networks, and implemented fuzzy logic control based on the switch valve. A nonlinear cascaded positioning control was presented for a high-speed linear movement using pneumatic muscle actuators in [7], where Sliding Mode Control (SMC) was used in the inner control loop to control the internal muscle pressure, and the outer control loop achieved the decoupling of the rocker angle and the muscle pressure. It had negligible steady-state position error and slight steady-state pressure error. A constrained finite-time optimal controller based on a multiple PAM model approximation was proposed for pneumatic positioning [8]. Reference [9] presented a nonlinear PID based antagonistic control to compensate the PAM's hysteretic nonlinearity and to improve robustness, it obtained high accuracy and execution speed.

As compared to positioning control, tracking control of pneumatic servo systems is more complicated, especially for tracking sinusoidal-like reference signal. When the velocity of the piston changes its direction, the Coulomb friction force suddenly changes its direction and amplitude, too, which increases the difficulty of the precise control. Due to the strong robustness to system parameter uncertainties and external disturbances, Sliding Mode Control (SMC) was proposed to the tracking control of pneumatic servo systems. A SMC was designed in [10], using third-order both linear and nonlinear models, to get the good results. However, this method was not robust to the payload variations. A second order sliding surface was designed for a pneumatic motion control system in [11], only the simple on/off valves and a position sensor were needed, which improved the overall reliability of the hardware. A SMC was developed using the pressure observer to simplify the structure of the system and reduce the cost [12], but the tracking accuracy was relatively low. A sliding mode position control was proposed based on the averaged model of pneumatic system in [13], which achieved good tracking performance in position control. Adaptive backstepping design can guarantee the global stability of the system and the asymptotic tracking of unknown/time-variant parameters. A backstepping sliding mode control was developed for a pneumatic servo system in [14], which can track both the desired force and stiffness effectively. However, the conventional SMC control is model dependent, it is necessary to know the nonlinear terms and the boundary of uncertainties. A backstepping controller was designed based on multi-input multi-output model for four proportional valves based pneumatic systems [15]. This controller could stabilize the system under the condition of all parameters being known. However, if the parameters change too strongly, for example, the load variation as shown in the paper, the stability cannot be guaranteed. Adaptive state feedback controllers based backstepping design were proposed in [16] and [17], which didn't require either the prior knowledge of system model parameters or uncertainty bounds. The controllers were simple, and had high tracking accuracy. But the closed-loop stability of the system has not been verified. An adaptive backstepping slide mode control was proposed for the Pneumatic Position Servo System in [18]. The method is simple for implementation because the model parameters and the uncertain parameters bounds are not required, the pressure sensor is not needed as well. At the same time, the control accuracy is higher as compared with other five methods. A nonlinear controller was proposed for a pneumatic cylinder by adopting the theories of homogeneity and finite-time stability in [19]. The experimental results showed the effectiveness of this method, but the chattering of control force is large. Feedforward hysteresis compensation and adaptive backstepping control were combined in [20] for a pneumatic muscles and the tracking error was reduced effectively. A hybrid fuzzy-repetitive control method based on an existing PID control was proposed in [21] to achieve good tracking performance. Reference [22] compared the performance of a PID controller plus an adaptive neural network compensator with a fuzzy adaptive PID controller. Both methods in [22] achieved good performance, but the parameters adjustment was very complicated. An extended state observer and active disturbance rejection adaptive control strategy proposed by [23] could be extended to pneumatic system control, however, the full state information was required in this method.

However, to our best knowledge, all existing pneumatic systems control methods assume that control gain direction of the system is known. When the control gain direction of actuators is unknown or changeable, most of the current adaptive control methods would fail to control. Moreover, the difficulty to obtain the exact parameters of the pneumatic servo system impedes the design of high performance tracking controller based on exact model. It is an additional obstacle that the designer cannot know the control force direction for sure. A simple reason might be the wrong connection of the valve output due to the mistakes of the operator. The wrong connection might not attract more attention in the past, because it just causes the piston to move to the protective limitation point. Then the human manipulator just stops the system, moves the piston away from the limitation point and shifts the connection. In some application fields, for example, the pneumatic assistant system for disabled human being, this case becomes complicated because it might damage the device or even hurt people. In this case, if the problem can be solved using controller design approach, it is helpful to form the intelligent mechatronics. Even for the portable pneumatic tools, users would be happy if they don't need to care the connection. The nominal value of the zero point of the proportional valve is usually inaccurate, either due to the environment variation or the D/A converter offset, which is usually ignored by the designer because it is believed that the close loop control would compensate its effect. However, because of the integral property of the proportional valve and the time-varying reference signal (for tracking control), the inaccurate zero point of the valve will degrade the desired performance.

In this paper, an adaptive state feedback control is designed for the pneumatic servo system with unknown model parameters, control gain direction and inaccurate zero point of the proportional valve by using adaptive backstepping method and Nussbaum function. The proposed method doesn't require the model parameters and the control gain direction. Using Lyapunov stability theorem, the designed adaptive controller is proved to be stable, the closed-loop controller can ensure that all signals in the system are bounded and asymptotic convergent. The proposed method is used to track three reference signals to show the effectiveness of the proposed method. The first feature of the proposed method is that the method is effective even the control gain direction is unknown or different from that is assumed in the controller design stage. The second feature of the proposed method is that the better tracking accuracy is achieved by taking the valve zero point into account, as compared to some existing methods.

This paper is organized as follows. In section II, the mathematical model of the pneumatic servo system and Nussbaum-type function are introduced briefly. In section III, the proposed backstepping controller with Nussbaum-type function and the stability proof of the controlled system are given. In section IV, the experimental results of the proposed controller and its comparison with other existing controllers are presented to show the effectiveness and superiority of the proposed method from the viewpoint of the tracking accuracy. Finally, in section V, conclusions are given.

II. PNEUMATIC SYSTEM CONFIGURATION AND ITS MODEL

The working principle of the pneumatic servo system is shown in Fig. 1. Where the air pump provides compressed air, the proportional valve controls the flow of the pressured air into Chamber A and Chamber B. The computer gives control signal to the proportional valve via D/A converter in the data acquisition card in computer, thus controls the pressure difference of two Chambers in order to drive the payload movement. The potentiometer is used to measure the displacement of the payload. The displacement signal is fed into the computer through A/D converter in the same data acquisition card. The user interface is designed on computer, it displays the payload position and the reference signals [26]–[28].

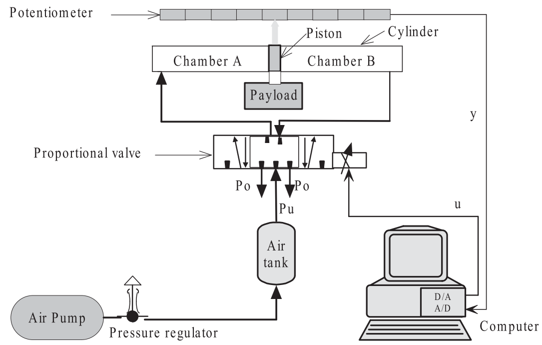


FIGURE 1. Pneumatic servo control experiment system.

The model of the pneumatic servo system is given by [4], [10], [16], [17]:

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

where, $\dot{m}_a$ and $\dot{m}_b$ are the gas mass flow rates into Chambers A and B, respectively, p_a and p_b are pressures in Chambers A and B, respectively. A_a and A_b are the cross section area of the two sides of the piston corresponding to Chambers A and B, respectively. y is the payload displacement, y_o is the initial payload displacement, M is the total mass of the payload and piston, F_f is the friction force. K is the specific heat ratio, R is the ideal gas constant, T is the air temperature, u is the input voltage of the proportional valve.

$f_a(u, p_a)$, $f_b(u, p_b)$ are nonlinear functions of the upper stream and lower stream pressure of Chambers A and B, respectively, which are given as:

$$\begin{cases} f_a(u, p_a) = \sqrt{p_u - p_a}(c_{a_1}u + c_{a_2}u^2) \\ f_b(u, p_b) = \sqrt{p_b - p_0}(c_{b_1}u + c_{b_2}u^2), \end{cases} \tag{2}$$

where p_u is the upstream pressure, p_0 is the atmospheric pressure c_{a_1}, c_{a_2}, c_{b_1}, c_{b_2} are constants related to the air property.

In order to facilitate analysis and system design, $f_a(u, p_a)$ and $f_b(u, p_b)$ are linearized, the friction F_f and other unmodeled factors are treated as disturbance, and the inaccurate zero point of the proportional valve is also considered, then the above nonlinear mathematical model (1) is simplified as a third-order linear model given by:

$$\dddot{y}(t) = a_1y(t) + a_2\dot{y}(t) + a_3\ddot{y}(t) + b(u(t) + \Delta u(t)) + d(t), \tag{3}$$

where, a_1, a_2 and a_3 are the unknown parameters; b is the control gain, whose value and sign are completely unknown; $d(t)$ is the disturbance; $\Delta u(t)$ is the zero point of proportional valve.

Define the state variables as $x_1(t) = y(t)$, $x_2(t) = \dot{y}(t)$ and $x_3(t) = \ddot{y}(t)$, which represent the displacement, the velocity and the acceleration of the payload, respectively. The third-order linear model can then be written as:

$$\begin{cases} \dot{x}_1(t) = x_2(t) \\ \dot{x}_2(t) = x_3(t) \\ \dot{x}_3(t) = a_1x_1(t) + a_2x_2(t) + a_3x_3(t) \\ +b(u(t) + \Delta u(t)) + d(t) \\ y(t) = x_1(t), \end{cases} \tag{4}$$

The Nussbaum-type function is introduced as follows.

If function has the following properties [25]:

$$\begin{cases} \lim\limits_{s\to\infty} \sup \dfrac{1}{s}\displaystyle\int_{s_0}^{s} N(\xi)d\xi = +\infty \\ \lim\limits_{s\to\infty} \inf \dfrac{1}{s}\displaystyle\int_{s_0}^{s} N(\xi)d\xi = -\infty, \end{cases} \tag{5}$$

then it is called Nussbaum-type function. There are many functions satisfying the above condition, such as $\xi^2\cos(\xi)$, $e^{\xi^2}\cos(\xi)$ and $\ln(\xi+1)\cos(\sqrt{\xi+1})$.

III. ADAPTIVE TRACKING CONTROLLER DESIGN BASED ON BACKSTEPPING METHOD AND NUSSBAUM GAIN

For tracking control of pneumatic servo system (4), the control goal is to make the output $y(t)$ track the reference signal $y_d(t)$, that is $\lim\limits_{t\to\infty} [y(t) - y_d(t)] = 0$. Before designing a controller, we give the following assumption.

Assumption 1: the up to third order derivative of the reference signal y_d is piecewise continuous and bounded.

A. THE BACKSTEPPING DESIGN WITH NUSSBAUM GAIN

The third-order linear model of pneumatic servo system can be rewrite as:

$$\begin{cases} \dot{x}_1(t) = x_2(t) \\ \dot{x}_2(t) = x_3(t) \\ \dot{x}_3(t) = a_1x_1(t) + a_2x_2(t) + a_3x_3(t) \\ +bu(t) + d_1(t) \\ y(t) = x_1(t), \end{cases} \tag{6}$$

where $d_1 = d + b\Delta u$ is the uncertainty including the friction and the inaccurate zero point of the proportional valve.

Define the error variables as:

$$\begin{cases} z_1(t) = x_1(t) - y_d(t) \\ z_2(t) = x_2(t) - \alpha_1 \\ z_3(t) = x_3(t) - \alpha_2, \end{cases} \tag{7}$$

where α_1 and α_2 are the virtual control variables of x_2 and x_3, respectively [26], [27].

The derivatives of the above variables are:

$$\begin{cases} \dot{z}_1(t) = z_2(t) + \alpha_1 - \dot{y}_d(t) \\ \dot{z}_2(t) = z_3(t) + \alpha_2 - \dot{\alpha}_1 \\ \dot{z}_3(t) = a_1x_1(t) + a_2x_2(t) + a_3x_3(t) + bu(t) \\ +d_1(t) - \dot{\alpha}_2. \end{cases} \tag{8}$$

We design a backstepping controller using the steps as follows:

Step 1: Define the first Lyapunov function candidate as:

$$v_1(t) = \frac{1}{2}{z_1}^2(t), \tag{9}$$

whose derivative is:

$$\dot{v}_1(t) = z_1(t)\dot{z}_1(t) = z_1(t)z_2(t) + z_1(t)\alpha_1 - \dot{y}_d(t)z_1(t). \tag{10}$$

Select α_1 as:

$$\alpha_1 = \dot{y}_d(t) - c_1z_1(t), \tag{11}$$

where c_1 is a positive constant. Then the time derivative of v_1 is:

$$\dot{v}_1(t) = z_1(t)z_2(t) - c_1{z_1}^2(t). \tag{12}$$

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

Step 2: Choose the second Lyapunov function candidate as:

$$v_2(t) = v_1(t) + \frac{1}{2}{z_2}^2(t), \tag{13}$$

whose derivative is:

$$\begin{aligned} \dot{v}_2(t) &= \dot{v}_1(t) + z_2(t)\dot{z}_2(t) \\ &= z_1(t)z_2(t) - c_1{z_1}^2(t) + z_2(t)z_3(t) + \alpha_2z_2(t) - \dot{\alpha}_1z_2(t). \end{aligned} \tag{14}$$

Select α_2 as:

$$\alpha_2 = \dot{\alpha}_1 - c_2z_2(t) - z_1(t), \tag{15}$$

where c_2 is a positive constant. Then the time derivative of $v_2(t)$ becomes:

$$\dot{v}_2(t) = z_2(t)z_3(t) - c_1{z_1}^2(t) - c_2{z_2}^2(t). \tag{16}$$

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

Step 3: Set $\boldsymbol{\theta} = [1\ \ a_1\ \ a_2\ \ a_3]^T$, $\hat{\boldsymbol{\theta}}$ is the estimation of $\boldsymbol{\theta}$. Choose the third Lyapunov function candidate as:

$$v_3(t) = v_2(t) + \frac{1}{2}{z_3}^2(t) + \frac{1}{2}\tilde{\boldsymbol{\theta}}^T\boldsymbol{\Gamma}^{-1}\tilde{\boldsymbol{\theta}} + \frac{1}{2\epsilon}{\tilde{d}_1}^2(t), \tag{17}$$

where, $\tilde{\boldsymbol{\theta}} = \hat{\boldsymbol{\theta}} - \boldsymbol{\theta}$ is parameter estimation error vector. $\boldsymbol{\Gamma}$ is a positive defined matrix. $\tilde{d}_1(t) = \hat{d}_1(t) - d_1(t)$ is the estimated error of $d_1(t)$, $\hat{d}_1(t)$ is the estimated value of $d_1(t)$.

Let $\boldsymbol{\omega}(t) = [z_2(t) - \dot{\alpha}_2\ \ x_1(t)\ \ x_2(t)\ \ x_3(t)]^T$. Then the derivative of $v_3(t)$ is given as:

$$\begin{aligned} \dot{v}_3(t) &= \dot{v}_2(t) + z_3(t)\dot{z}_3(t) + \tilde{\boldsymbol{\theta}}^T\boldsymbol{\Gamma}^{-1}\dot{\hat{\boldsymbol{\theta}}} + \frac{1}{\epsilon}\tilde{d}_1(t)\dot{\hat{d}}_1(t) \\ &= -c_2{z_2}^2(t) - c_1{z_1}^2(t) + z_3(t)(bu + \boldsymbol{\theta}^T\boldsymbol{\omega} + \hat{d}_1(t)) \\ &\quad + \tilde{\boldsymbol{\theta}}^T\boldsymbol{\Gamma}^{-1}\dot{\hat{\boldsymbol{\theta}}} + \frac{1}{\epsilon}\tilde{d}_1(t)(\dot{\hat{d}}_1(t) - \epsilon z_3(t)). \end{aligned} \tag{18}$$

The control law and adaptive law are given as follows:

$$u(t) = N(\xi)(c_3 z_3(t) + \hat{\boldsymbol{\theta}}^T \boldsymbol{\omega}) - \hat{d}_1(t)/b, \tag{19}$$
$$N(\xi) = \xi^2 \cos(\xi), \tag{20}$$
$$\dot{\xi}(t) = z_3(t)(c_3 z_3(t) + \hat{\boldsymbol{\theta}}^T \boldsymbol{\omega}), \tag{21}$$
$$\dot{\hat{\boldsymbol{\theta}}} = \boldsymbol{\Gamma\omega} z_3(t), \tag{22}$$
$$\dot{\hat{d}}_1(t) = \epsilon z_3(t). \tag{23}$$

where c_3 is a positive constant. $N(\xi)$ is a Nussbaum-type function, in this paper, an even Nussbaum function, $\xi^2\cos(\xi)$, is used. In the controller, $x_1 = y$ is measured using the position potentiometer, x_2 and x_3 are calculated from x_1 using Eular method.

B. STABILITY ANALYSIS OF THE CONTROL SYSTEM

A theorem about the stability of control system (4) and (19)-(23) is given as follows:

Theorem 1: System (4) is stable and all signals in the closed-loop system are bounded, by using control law (19), (20) and parameter adaptive laws (21), (22), (23), meanwhile $\lim\limits_{t\to\infty} [y(t) - y_d(t)] = 0$.

To prove Theorem 1, we introduce a lemma:

Lemma 1 [24]: Let $v(\cdot)$ and $\xi(\cdot)$ be smooth functions defined on $[0, t_f)$, and satisfy $v(t) > 0$, $\forall t \in [0, t_f)$. $N(\xi)$ is a smooth even Nussbaum-type function. If the following inequality holds:

$$v(t) \leq c_0 + \int_0^t (bN(\xi) + 1)\dot{\xi} d\tau, \quad \forall t \in [0, t_f), \tag{24}$$

where b is a non-zero constant, c_0 is a appropriate constant, then $v(t)$, $\xi(t)$ and $\int_0^t (bN(\xi) + 1)\dot{\xi} d\tau$ must be bounded on $[0, t_f)$.

Proof of Theorem 1:

Substituting from (19) to (23) into (18), we obtain:

$$\begin{aligned}\dot{v}_3(t) &= -\sum_{i=1}^{2} c_i z_i{}^2(t) + z_3(t)[bN(\xi)(c_3 z_3(t) + \hat{\boldsymbol{\theta}}^T \boldsymbol{\omega}) + \boldsymbol{\theta}^T \boldsymbol{\omega}] \\ &\quad + \tilde{\boldsymbol{\theta}}^T \boldsymbol{\Gamma}^{-1} \dot{\hat{\boldsymbol{\theta}}} \\ &= -\sum_{i=1}^{2} c_i z_i{}^2(t) + bN(\xi)\dot{\xi} + z_3(t)\boldsymbol{\theta}^T \boldsymbol{\omega} + \tilde{\boldsymbol{\theta}}^T \boldsymbol{\Gamma}^{-1} \dot{\hat{\boldsymbol{\theta}}}.\end{aligned} \tag{25}$$

On the right side of (25), we firstly add, then subtract $\dot{\xi}$, then by taking (22) into account, we obtain:

$$\begin{aligned}\dot{v}_3(t) &= -\sum_{i=1}^{2} c_i z_i{}^2(t) + bN(\xi)\dot{\xi} + z_3(t)\boldsymbol{\theta}^T \boldsymbol{\omega} + \tilde{\boldsymbol{\theta}}^T \boldsymbol{\Gamma}^{-1} \dot{\hat{\boldsymbol{\theta}}} \\ &\quad + \dot{\xi} - z_3(t)(c_3 z_3(t) + \hat{\boldsymbol{\theta}}^T \boldsymbol{\omega}) \\ &= -\sum_{i=1}^{3} c_i z_i{}^2(t) + (bN(\xi) + 1)\dot{\xi}.\end{aligned} \tag{26}$$

Integrate both sides of (26), we have:

$$v_3(t) = v_3(0) - \sum_{i=1}^{3} c_i \int_0^t z_i{}^2(t) d\tau + \int_0^t (bN(\xi) + 1)\dot{\xi} d\tau. \tag{27}$$

Since $\sum_{i=1}^{3} c_i \int_0^t z_i{}^2(t) d\tau$ is nonnegative, then we have the following inequality:

$$v_3(t) \leq v_3(0) + \int_0^t (bN(\xi) + 1)\dot{\xi} d\tau. \tag{28}$$

Based on Lemma 1, we conclude that $v_3(t)$, $\xi(t)$ and $\int_0^t (bN(\xi) + 1)\dot{\xi} d\tau$ are all bounded on $[0, t_f)$. From (27), one has that $\sum_{i=1}^{3} c_i \int_0^t z_i{}^2(t) d\tau$ is also bounded for all $t \geq 0$. Because $v(t)$ and $\xi(t)$ are bound for all $t \geq 0$, the derivative of $z_i{}^2(t)(i = 1, 2, 3)$ is also bounded. Because the terms in (27) are all continuous, using the Barbalat's lemma [28], we know that the tracking error $z_i(t)$ tends to zero when $t \to \infty$, that is $\lim\limits_{t\to\infty} [y(t) - y_d(t)] = 0$.

End of proof.

IV. EXPERIMENTAL RESULTS

A. EXPERIMENTAL PLATFORM

The pneumatic servo system produced by Festo company. The experiment setup is shown in Fig. 2. The system includes a double-acting rodless pneumatic cylinder, a electric power to supply the power to electric element, a five-way proportional valve, a potentiometer, an air tank, a data acquisition interface to connect the signal wire to computer system and a manual pressure regulator. In the cylinder, the slide mass is 2.7 Kg, the piston diameter is 25 mm, the effective force is 200 N at 6 bar air pressure supply. The nominal operating pressure of the valve is 6 bar, the maximum flow rate of the valve at the nominal pressure is 700 L/min. The resolution of the potentiometer is less than 0.01 mm, the linear error of the potentiometer is less than 0.07% of full scale, the effective electrical working distance is 457 mm. The valve control voltage is 0-10 V. The output of the potentiometer is also 0-10 V. The bit length of A/D and D/A converter is 12. The capacity of the air tank is 400 mL.

As shown in Fig. 3, two outlets of the proportional valve should be connected to Chamber A and Chamber B of the cylinder, respectively. We define the positive connection direction as shown in Fig. 3 (a), and the negative connection direction as shown in Fig. 3 (b). The match between two valve outlets and two chambers determines the control gain direction. If the connection match between the valve outlets and the chambers changes, it would cause the reverse of the control gain direction, and consequently lead to the controlled system failure, even the disaster in some practical applications. In this paper, an adaptive control is proposed to avoid the failure caused by the reversed control gain direction.

Considering the characteristic of the proportional valve, the controller output of all methods is limited to

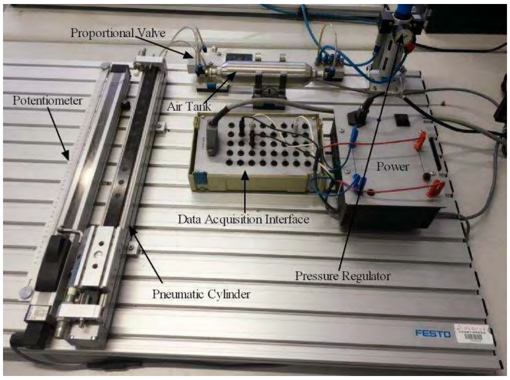
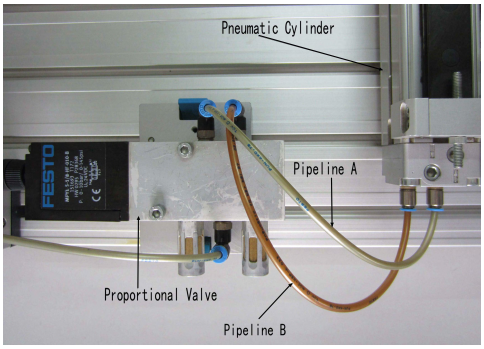
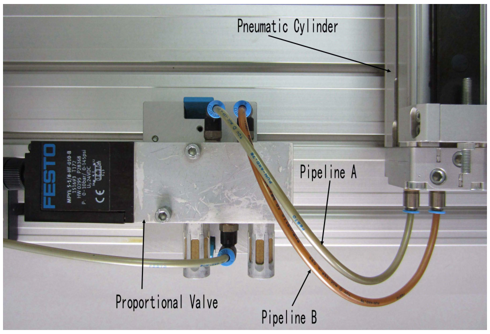


FIGURE 2. Pneumatic servo control experiment system.



(a)



(b)

FIGURE 3. Control direction definition of the pneumatic platform. (a) Positive direction. (b) Negative direction.

$[-U_{\max}, U_{\max}]$, where $U_{\max} = 1.96$V. The payload position $y = x_1$ is measured by the analog displacement sensor directly. The sampling period of the controller is $\Delta t = 10$ms. Before the experiment, the slider is forced to move to the middle point of the cylinder, and the position should be used as the reference signal zero point.

The reference signals are defined as follows:

1) Reference signal 1:

$$y_d(t) = A_1 \sin(\omega_1 t), \tag{29}$$

where $A_1 = 167.5$ mm, $\omega_1 = 0.5\pi$ rad/s. This is a sinusoidal reference signal.

2) Reference signal 2:

$$y_d(t) = \begin{cases} -(A_2/\omega_2{}^2)\sin(\omega_2 t) + (A_2/\omega_2)t, & t < 4 \\ 142.157mm, & t \geq 4 \end{cases} \tag{30}$$

where $A_2 = 55.825$ mm/s^2, $\omega_2 = 0.5\pi$ rad/s. This is a S-curve reference signal.

3) Reference signal 3:

$$y_d(t) = A_3[\sin(2\omega_3 t) + \sin(\omega_3 t) + \sin(4\omega_3 t/7) + \sin(\omega_3 t/3) + \sin(4\omega_3 t/17)], \tag{31}$$

where $A_3 = 167.475$ mm, $\omega_3 = 0.5\pi$ rad/s. This is a multi-frequency sinusoidal reference signal.

B. EXPERIMENT RESULTS OF THE PROPOSED METHOD

The pneumatic system is controlled by the proposed method to track the three types of reference signals. Before each test, the piston is forced to middle point of the cylinder by close loop control. Therefore, the initial conditions are set to $x_2(0) = 0$, $x_3(0) = 0$, $\xi(0) = 0$. The controller parameters are set as follows $c_1 = c_2 = 60$, $c_3 = 0.01$, $\mathbf{\Gamma} = diag([0.01\ \ 0.01\ \ 0.01])$. The corresponding experimental results for the positive control direction as the connection shown in Fig. 3(a) are given in Fig. 4, and the corresponding experimental results for the negative control direction as the connection shown in Fig. 3(b) are given in Fig. 5. Subplots (a), (b) and (c) of Figs. 4-5 are corresponding to the reference signal 1, 2 and 3, respectively. In Figs. 4-5, the black dash line is a reference signal, the blue solid line is the actual output of the system in the upper panel, the middle panel shows the tracking error. The red dash line is adaptive parameter ξ waveform and the blue solid line is Nussbaum gain $N(\xi)$ waveform in the lower panel of each plot. According to Figs. 4-5, we know that the proposed method can achieve effective tracking control for both control directions.

C. COMPARISON TO SOME EXISTING METHODS

To show the superiority of the proposed method, the comparison to other five methods is given in this section. Methods in [10], [11], and [16]–[18] also is conducted to track the three types of reference signals, respectively. The details introduction and parameter values for the comparison methods are described in appendix.

The parameters (including the parameters of the proposed method and the comparing methods) are all carefully tuned with many times trial and error procedure to get nice performance for the nominal experiment condition (without initial position change and payload variation).

In order to compare the steady state tracking error of these five methods quantitatively, we defined two indices, one is a root mean square error ($RMSE$) [29]–[32] defined as:

$$RMSE = \sqrt{\frac{1}{N}\sum_{k=N_1}^{N_2} e_k{}^2}, \tag{32}$$

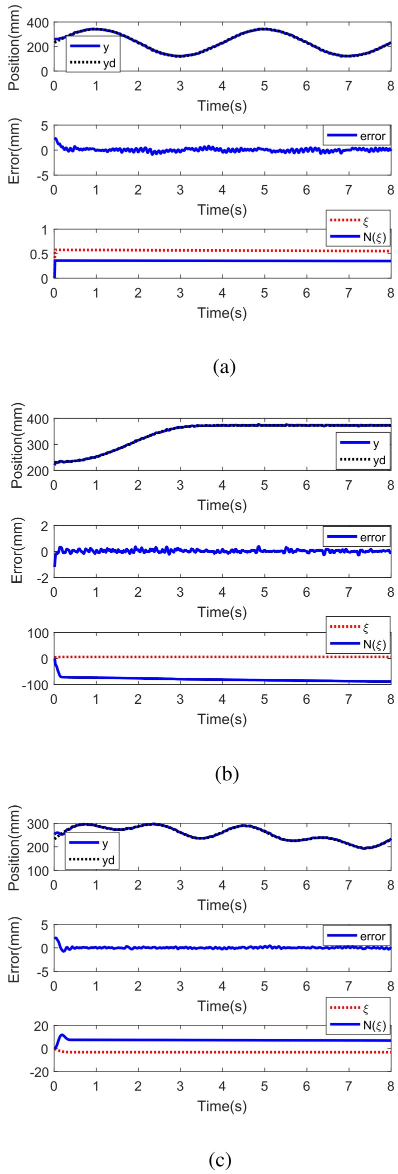


FIGURE 4. **Experimental results using the proposed method for positive direction. (a) Tracking reference signal 1. (b) Tracking reference signal 2. (c) Tracking reference signal 3.**

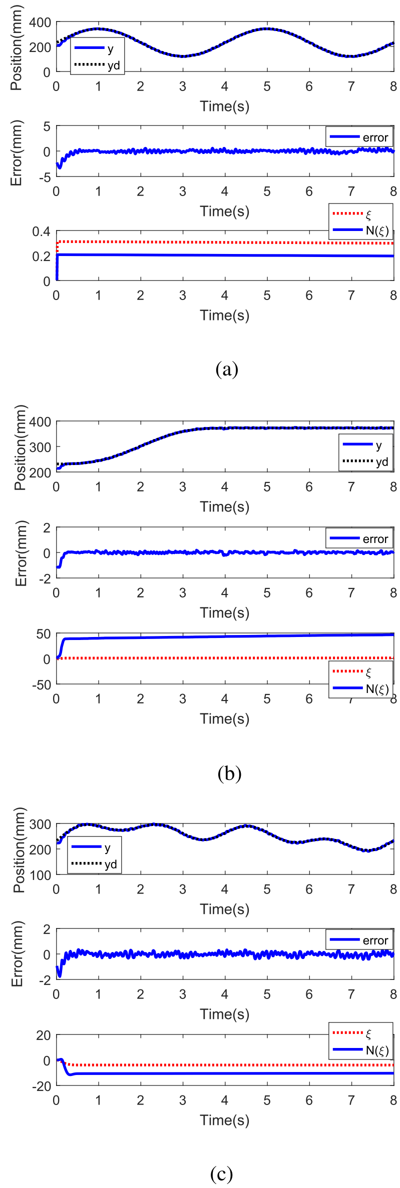


FIGURE 5. **Experimental results using the proposed method for negative direction. (a) Tracking reference signal 1. (b) Tracking reference signal 2. (c) Tracking reference signal 3.**

another is the average absolute error (AAE) defined as

$$AAE = \frac{1}{N}\sum_{k=N_1}^{N_2}\left|{e_k}^2\right|, \tag{33}$$

where $N = N_2 - N_1 + 1$, N_1 is the start time of the steady state considered, N_2 is the end time of the steady state considered, $e_k = y_d(k\Delta t) - y(k\Delta t)$ is the tracking error at kth sampling time. In this paper, $N_1 = 200$, and $N_2 = 1500$.

In order to avoid the influence of random factors such as noise, every experiment is done at least 30 times to get average $RMSE$, maximum $RMSE$, average AAE, and maximum AAE. The quantitative comparison results are given in Tables 1 to 3.

To further check the effect of the extra payload and the piston initial position on the performance of the tracking, the payload mass is set equal to the slide mass plus an extra 2.5Kg payload, and the initial position of the piston is changed from 225mm to 170mm. The tracking performance comparison of the comparison methods for positive control direction and the proposed method for both positive and negative control directions are given in Tables 4 to 9.

From the quantitative comparison given in all tables, we conclude that the average $RMSE$, the max $RMSE$, the average AAE, and the maximum AAE of the proposed method are less than those of the comparison methods in [10], [11], and [16]–[18] and the method without considering the inaccurate zero point of the valve under the same load

TABLE 1. Tracking error comparison for the reference signal 1 without extra payload.

Method		Avg.RMSE	Max.RMSE	Avg.AAE	Max.AAE
Considering zero point	positive direction	1.8321	1.9877	1.4834	1.5662
	negative direction	1.9356	1.9912	1.5299	1.6399
Without considering zero point	positive direction	1.9435	2.0145	1.6295	1.6880
	negative direction	2.0747	2.1686	1.6747	1.8047
Method in Reference [10]		2.2213	2.3517	1.7866	1.9534
Method in Reference [11]		2.3823	2.4365	2.3092	2.3485
Method in Reference [16]		2.3301	2.3761	2.0084	2.1021
Method in Reference [17]		2.2491	2.2966	2.1407	2.1914
Method in Reference [18]		2.0350	2.0995	2.0386	2.0479

TABLE 2. Tracking error comparison for the reference signal 2 without extra payload.

Method		Avg.RMSE	Max.RMSE	Avg.AAE	Max.AAE
Considering zero point	positive direction	0.5873	0.6136	0.4902	0.5482
	negative direction	0.6054	0.6230	0.5304	0.5775
Without considering zero point	positive direction	0.5908	0.6054	0.5355	0.5962
	negative direction	0.6128	0.6296	0.5616	0.5756
Method in Reference [10]		0.6302	0.6879	0.6319	0.6791
Method in Reference [11]		0.6295	0.7054	0.6571	0.7053
Method in Reference [16]		0.6251	0.6908	0.6335	0.6948
Method in Reference [17]		0.6122	0.6729	0.5948	0.6375
Method in Reference [18]		0.6051	0.6270	0.5891	0.8178

TABLE 3. Tracking error comparison for the reference signal 3 without extra payload.

Method		Avg.RMSE	Max.RMSE	Avg.AAE	Max.AAE
Considering zero point	positive direction	0.9195	0.9647	0.7003	0.7560
	negative direction	0.9255	0.9930	0.7004	0.7602
Without considering zero point	positive direction	0.9412	0.9574	0.7743	0.7922
	negative direction	0.9545	0.9811	0.7575	0.7875
Method in Reference [10]		1.0796	1.1308	0.9186	1.0527
Method in Reference [11]		1.1458	1.1972	1.3237	1.3662
Method in Reference [16]		1.1563	1.1998	1.1428	1.2046
Method in Reference [17]		0.9725	1.1125	1.0729	1.1358
Method in Reference [18]		1.0414	1.0496	1.0207	1.0531

TABLE 4. Tracking error comparison for the reference signal 1 with extra payload.

Method		Avg.RMSE	Max.RMSE	Avg.AAE	Max.AAE
Considering zero point	positive direction	2.0475	2.0954	1.5988	1.6145
	negative direction	2.0927	2.1277	1.7322	1.7693
Without considering zero point	positive direction	2.0576	2.0983	1.6758	1.7325
	negative direction	2.1102	2.1262	1.7839	1.8491
Method in Reference [10]		2.2295	2.3539	1.9228	1.9742
Method in Reference [11]		2.4118	2.4509	2.3541	2.3961
Method in Reference [16]		2.3558	2.4013	2.0492	2.1129
Method in Reference [17]		2.2586	2.3105	2.1944	2.2415
Method in Reference [18]		2.3054	2.5421	1.8872	1.9723

TABLE 5. Tracking error comparison for the reference signal 2 with extra payload.

Method		Avg.RMSE	Max.RMSE	Avg.AAE	Max.AAE
Considering zero point	positive direction	0.6247	0.6299	0.5730	0.5911
	negative direction	0.6347	0.6399	0.5902	0.5993
Without considering zero point	positive direction	0.6313	0.6794	0.6193	0.6414
	negative direction	0.6428	0.6882	0.6405	0.6694
Method in Reference [10]		0.6521	0.7053	0.7696	0.9741
Method in Reference [11]		0.7643	0.7943	0.7835	0.8141
Method in Reference [16]		0.6591	0.6985	0.6484	0.7048
Method in Reference [17]		0.6521	0.6821	0.6704	0.7254
Method in Reference [18]		0.6626	0.7984	0.6599	0.6987

TABLE 6. Tracking error comparison for the reference signal 3 with extra payload.

Method		Avg.RMSE	Max.RMSE	Avg.AAE	Max.AAE
Considering zero point	positive direction	0.9760	0.9872	0.7179	0.7845
	negative direction	0.9975	1.0522	0.7204	0.8666
Without considering zero point	positive direction	1.0236	1.0957	0.7358	0.9587
	negative direction	1.0881	1.1218	0.7796	0.9613
Method in Reference [10]		1.1128	1.1356	1.1035	1.1394
Method in Reference [11]		1.1696	1.2064	1.3511	1.4001
Method in Reference [16]		1.1675	1.2003	1.1484	1.2196
Method in Reference [17]		1.1046	1.1187	1.0929	1.1391
Method in Reference [18]		1.0432	1.2134	0.9944	1.0242

TABLE 7. Tracking error comparison for the reference signal 1 with extra payload and changed initial position.

Method		Avg.RMSE	Max.RMSE	Avg.AAE	Max.AAE
Considering zero point	positive direction	2.1606	2.2554	1.7326	1.8277
	negative direction	2.2037	2.2748	1.7628	1.8359
Without considering zero point	positive direction	2.2427	2.2853	1.8230	1.8649
	negative direction	2.2731	2.3208	1.8469	1.8973
Method in Reference [10]		2.6483	2.7709	2.3359	2.5350
Method in Reference [11]		2.4668	2.5333	2.9169	3.4553
Method in Reference [16]		2.4174	2.7427	2.2030	2.3634
Method in Reference [17]		2.3984	2.5122	2.2245	2.4367
Method in Reference [18]		2.3287	2.6523	1.8347	1.8975

TABLE 8. Tracking error comparison for the reference signal 2 with extra payload and changed initial position.

Method		Avg.RMSE	Max.RMSE	Avg.AAE	Max.AAE
Considering zero point	positive direction	0.8384	0.8582	0.6772	0.6902
	negative direction	0.8529	0.8886	0.7541	0.7887
Without considering zero point	positive direction	0.8671	0.8893	0.8519	0.9215
	negative direction	0.8945	0.9232	0.8631	0.9338
Method in Reference [10]		0.9441	0.9789	0.9424	0.9818
Method in Reference [11]		0.9898	1.0202	0.9660	1.1959
Method in Reference [16]		0.9394	0.9665	0.9258	0.9674
Method in Reference [17]		0.9267	0.9819	0.9001	0.9340
Method in Reference [18]		0.8990	0.9332	0.8754	0.9424

TABLE 9. Tracking error comparison for the reference signal 3 with extra payload and changed initial position.

Method		Avg.RMSE	Max.RMSE	Avg.AAE	Max.AAE
Considering zero point	positive direction	1.1096	1.1354	0.8674	0.8966
	negative direction	1.1187	1.1553	0.8803	0.9184
Without considering zero point	positive direction	1.1365	1.1671	0.8972	1.0951
	negative direction	1.1592	1.1880	0.9250	1.1247
Method in Reference [10]		1.2470	1.2749	1.1733	1.2358
Method in Reference [11]		1.2597	1.3034	1.4216	1.5128
Method in Reference [16]		1.2421	1.3852	1.2563	1.3119
Method in Reference [17]		1.2398	1.2761	1.1530	1.2296
Method in Reference [18]		1.1398	1.1939	0.9333	1.1245

and initial position condition. The controller designed in this paper has better tracking performance than the other methods for both the positive control direction and the negative control direction.

V. CONCLUSION

The proposed method considers unknown model parameters, unknown control direction and inaccurate valve zero point at the same time. This method does not require expensive pressure sensor, which simplifies the system structure and reduces the cost. By using Lyapunov method and linearized model, the stability of the proposed controller is proved. The proposed method is robust to the payload and initial position variations. The proposed method is effective even though the control direction is unknown or changed. The proposed method is superior to the corresponding method without considering the zero point of the valve. Compared with some existing methods, the experimental results verify the superiority of the proposed method.

APPENDIX

The details introduction for the comparison methods are given as following:

In [10], two model-based sliding-mode position tracking control algorithms for pneumatic cylinder actuators were proposed. The sliding-mode controller based on linearized model is given as:

$$u_{10} = \dddot{y}_d + n_2\ddot{y} + n_1\dot{y} - 2\lambda(\ddot{y} - \ddot{y}_d) - \lambda^2(\dot{y} - \dot{y}_d)/n_0 - k_{s1}sat(S/\phi), \quad (34)$$

where $S = (\ddot{y}_d - \ddot{y}) + 2\lambda(\dot{y}_d - \dot{y}) + \lambda^2(y_d - y)$, $n_2 = 29.5$, $n_1 = 218.43$, $n_0 = 5531.3$, $\lambda = 50$, $k_{s1} = 976$, $\phi = 0.05$.

In [11], the sliding mode control based on second order sliding surface was proposed. The controller is given as:

$$u_{11} = \begin{cases} -k_{s2}sign(s), & |s| > \varepsilon \\ 0, & |s| < \varepsilon, \end{cases} \quad (35)$$

where $s = \ddot{e} + 2\xi\omega\dot{e} + \omega^2 e, k_{s2} = 1.96$, $\omega = 50$, $\xi = 1$.

In [16], an adaptive backstepping controller was designed without the knowledge of the accurate model of the pneumatic system, which simplifies the design procedure. The controller in [16] is given as:

$$u_{16} = \frac{1}{\hat{b}}\bar{u}, \quad (36)$$

$$\bar{u} = \alpha_2 + \dddot{y}_d, \quad (37)$$

$$\dot{\frac{1}{\hat{b}}} = -\gamma sgn\frac{1}{(b_0)}(b_0)\bar{u}z_3, \quad (38)$$

$$\dot{\hat{A}} = \Gamma \boldsymbol{x} z_3, \quad (39)$$

where $\boldsymbol{x} = [x_1\ x_2\ x_3]^T$, $A = [a_1\ a_2\ a_3]^T$, $c_1 = c_2 = 50$, $\lambda = 1$, $\Gamma = diag[\,1\ 1\ 1\,]$.

Reference [17] also employed adaptive backstepping method, the controller in [17] is given as:

$$u_{17} = \frac{1}{\hat{b}}(-z_2 + \hat{a}_1 z_1 + \hat{a}_1 y_d + \hat{a}_2 z_2 + \hat{a}_2\alpha_1 + \hat{a}_3\alpha_2 + \dot{\alpha}_2 - c_3 z_3 - \hat{a}_3 z_3). \quad (40)$$

where z_1, z_2, z_3 are error variables, which are defined as:

$$\begin{cases} z_1 = x_1 - y_m \\ z_2 = x_2 - \alpha_1 \\ z_3 = x_3 - \alpha_2, \end{cases} \quad (41)$$

And the adaptive law of the unknown parameters is:

$$\begin{cases} \dot{\hat{b}} = -\lambda z_1 y_d \hat{b}^2 \\ \dot{\hat{a}}_1 = \beta_1 z_1 x_1 \\ \dot{\hat{a}}_2 = \beta_2 z_1 x_2 \\ \dot{\hat{a}}_3 = \beta_3 z_1 x_3 \end{cases}, \quad (42)$$

where $c_1 = c_2 = c_3 = 50$, $\lambda = \beta_1 = \beta_2 = \beta_3 = 1$.

In [18], an adaptive backstepping sliding mode control was proposed given by:

$$u_{18} = -\hat{b}k_1(x_2 - \dot{y}_m) - \hat{b}k_2(x_3 - \dot{\alpha}_1) - \hat{\tau}_1 x_1 - \hat{\tau}_2 x_2 - \hat{\tau}_3 x_3 + \hat{b}\dot{\alpha}_2 - c_3 s - c_4 sgn(s), \quad (43)$$

Define the sliding surface as:

$$s = k_1 z_1 + k_2 z_2 + z_3, \quad (44)$$

where z_1, z_2, z_3 are error variables, which are defined as:

$$\begin{cases} z_1 = x_1 - y_m \\ z_2 = x_2 - \alpha_1 \\ z_3 = x_3 - \alpha_2. \end{cases} \tag{45}$$

Select virtual control as:

$$\begin{cases} \alpha_1 = \dot{y}_m - c_1 z_1 \\ \alpha_2 = \dot{\alpha_1} - c_2 z_2 - z_1. \end{cases} \tag{46}$$

The adaptive law of the unknown parameters is:

$$\begin{cases} \dot{\hat{\tau}}_1 = (1/\beta_1) s x_1 \\ \dot{\hat{\tau}}_2 = (1/\beta_2) s x_2 \\ \dot{\hat{\tau}}_3 = (1/\beta_3) s x_3 \\ \dot{\hat{b}} = (1/\lambda)[k_1(x_2 - \dot{y}_m) + k_2(x_3 - \dot{\alpha}_1) - \dot{\alpha}_2] s, \end{cases} \tag{47}$$

where $c_1 = c_2 = 60$, $c_3 = 0.01$, $c_4 = 60$, $k_1 = k_2 = 1$, $\beta_1 = \beta_2 = \beta_3 = 1, \lambda = 1$.

All above controllers are consistent to the expression in the references, although the variables might be different for different controllers. It would not affect audience for understanding the fact.

REFERENCES

[1] M. Turkseven and J. Ueda, "An asymptotically stable pressure observer based on load and displacement sensing for pneumatic actuators with long transmission lines," *IEEE/ASME Trans. Mechatronics*, vol. 22, no. 2, pp. 681–692, Apr. 2017. doi: 10.1109/TMECH.2016.2582222.

[2] R. B. van Varseveld and G. M. Bone, "Accurate position control of a pneumatic actuator using on/off solenoid valves," *IEEE/ASME Trans. Mechatronics*, vol. 2, no. 3, pp. 195–204, Sep. 1997. doi: 10.1109/3516.622972.

[3] J. Wang, D. J. D. Wang, P. R. Moore, and J. Pu, "Modelling study, analysis and robust servo control of pneumatic cylinder actuator systems," *IEE Proc.-Control Theory Appl.*, vol. 148, no. 1, pp. 35–42, Jan. 2001. doi: 10.1049/ip-cta:20010238.

[4] Y. Yin and J. Wang, "A nonlinear feedback tracking control for pneumatic cylinders and experiment study," in *Proc. Amer. Control Conf.*, Baltimore, MD, USA, Jun. 2009, pp. 3476–3481. doi: 10.1109/ACC.2009.5160340.

[5] K. Osman, A. 'A. M. Faudzi, M. F. Rahmat, N. D. Mustafa, and K. Suzumori, "Predictive functional controller design for pneumatic actuator with stiffness characteristic," in *Proc. IEEE/SICE Int. Symp. Syst. Integrat.*, Kobe, Japan, Dec. 2013, pp. 641–646. doi: 10.1109/SII.2013.6776700.

[6] T. Leephakpreeda, "Fuzzy logic based PWM control and neural controlled-variable estimation of pneumatic artificial muscle actuators," *Expert Syst. Appl.*, vol. 38, no. 6, pp. 7837–7850, Jun. 2011. doi: 10.1016/j.eswa.2010.12.120.

[7] H. Aschemann and D. Schindele, "Sliding-mode control of a high-speed linear axis driven by pneumatic muscle actuators," *IEEE Trans. Ind. Electron.*, vol. 55, no. 11, pp. 3855–3864, Nov. 2008. doi: 10.1109/TIE.2008.2003202.

[8] G. Andrikopoulos, G. Nikolakopoulos, I. Arvanitakis, and S. Manesis, "Piecewise affine modeling and constrained optimal control for a pneumatic artificial muscle," *IEEE Trans. Ind. Electron.*, vol. 61, no. 2, pp. 904–916, Feb. 2014. doi: 10.1109/TIE.2013.2254094.

[9] G. Andrikopoulos, G. Nikolakopoulos, and S. Manesis, "Advanced nonlinear PID-based antagonistic control for pneumatic muscle actuators," *IEEE Trans. Ind. Electron.*, vol. 61, no. 12, pp. 6926–6937, Dec. 2014. doi: 10.1109/tie.2014.2316255.

[10] G. M. Bone and S. Ning, "Experimental comparison of position tracking control algorithms for pneumatic cylinder actuators," *IEEE/ASME Trans. Mechatronics*, vol. 12, no. 5, pp. 557–561, Oct. 2007. doi: 10.1109/TMECH.2007.905718.

[11] T. Nguyen, J. Leavitt, F. Jabbari, and J. E. Bobrow, "Accurate sliding-mode control of pneumatic systems using low-cost solenoid valves," *IEEE/ASME Trans. Mechatronics*, vol. 12, no. 2, pp. 216–219, Apr. 2007. doi: 10.1109/TMECH.2007.892821.

[12] N. Gulati and E. J. Barth, "A globally stable, load-independent pressure observer for the servo control of Pneumatic actuators," *IEEE/ASME Trans. Mechatronics*, vol. 14, no. 3, pp. 295–306, Jun. 2009. doi: 10.1109/TMECH.2008.2009222.

[13] S. Hodgson, M. Tavakoli, M. T. Pham, and A. Leleve, "Nonlinear discontinuous dynamics averaging and PWM-based sliding control of solenoid-valve pneumatic actuators," *IEEE/ASME Trans. Mechatronics*, vol. 20, no. 2, pp. 876–888, Apr. 2015. doi: 10.1109/tmech.2014.2326601.

[14] B. Taheri, D. Case, and E. Richer, "Force and stiffness backstepping-sliding mode controller for pneumatic cylinders," *IEEE/ASME Trans. Mechatronics*, vol. 19, no. 6, pp. 1799–1809, Dec. 2014. doi: 10.1109/TMECH.2013.2294970.

[15] Z. Rao and G. M. Bone, "Nonlinear modeling and control of servo pneumatic actuators," *IEEE Trans. Control Syst. Technol.*, vol. 16, no. 3, pp. 562–569, May 2008. doi: 10.1109/tcst.2007.912127.

[16] H.-P. Ren and C. Huang, "Experimental tracking control for pneumatic system," in *Proc. 39th IEEE IECON*, Vienna, Austria, Nov. 2013, pp. 4126–4130. doi: 10.1109/IECON.2013.6699797.

[17] H.-P. Ren and C. Huang, "Adaptive backstepping control of pneumatic servo system," in *Proc. IEEE Int. Symp. Ind. Electron.*, Taipei, Taiwan, May. 2013, pp. 1–6. doi: 10.1109/ISIE.2013.6563773.

[18] H. Ren and J. Fan, "Adaptive backstepping slide mode control of pneumatic position servo system," *Chin. J. Mech. Eng.*, vol. 29, no. 5, pp. 1003–1009, Sep. 2016. doi: 10.3901/cjme.2016.0412.050.

[19] S. Riachy and M. Ghanes, "A nonlinear controller for pneumatic servo systems: Design and experimental tests," *IEEE/ASME Trans. Mechatronics*, vol. 19, no. 4, pp. 1363–1373, Aug. 2014. doi: 10.1109/TMECH.2013.2280956.

[20] H. Aschemann and D. Schindele, "Comparison of model-based approaches to the compensation of hysteresis in the force characteristic of pneumatic muscles," *IEEE Trans. Ind. Electron.*, vol. 61, no. 7, pp. 3620–3629, Jul. 2014. doi: 10.1109/tie.2013.2287217.

[21] S.-C. Hsu and C.-Y. Lin, "Periodic motion control of a heavy duty pneumatic actuating table using low-cost position sensors and hybrid repetitive Control," in *Proc. IEEE Int. Symp. Ind. Electron.*, Taipei, Taiwan, May. 2013, pp. 1–6. doi: 10.1109/isie.2013.6563590.

[22] B. Dehghan and B. W. Surgenor, "Comparison of fuzzy and neural network adaptive methods for the position control of a pneumatic system," in *Proc. IEEE CCECE*, Regina, SK, Canada, May. 2013, pp. 1–4. doi: 10.1109/CCECE.2013.6567712.

[23] J. Yao and W. Deng, "Active disturbance rejection adaptive control of hydraulic servo systems," *IEEE Trans. Ind. Electron.*, vol. 64, no. 10, pp. 8023–8032, Oct. 2017. doi: 10.1109/TIE.2017.2694382.

[24] Y. Xudong and J. Jingping, "Adaptive nonlinear design without a priori knowledge of control directions," *IEEE Trans. Autom. Control*, vol. 43, no. 11, pp. 1617–1621, Nov. 1998. doi: 10.1109/9.728882.

[25] R. D. Nussbaum, "Some remarks on a conjecture in parameter adaptive control," *Syst. Control Lett.*, vol. 3, no. 5, pp. 243–246, Nov. 1983. doi: 10.1016/0167-6911(83)90021-X.

[26] H. Ye, M. Chen, and Q. Wu, "Flight envelope protection control based on reference governor method in high angle of attack maneuver," *Math. Problems Eng.*, vol. 2015, Jan. 2015, Art. no. 254975. doi: 10.1155/2015/254975.

[27] J. de J. Rubio, G. Gutierrez, H. Perez, and J. Pacheco, "Comparison of three proposed control to accelerate the growth of the crop," *Int. J. Innov. Comput., Inf. Control*, vol. 7, no. 7B, pp. 4097–4114, Jul. 2011.

[28] H. K. Khalil, *Nonlinear System*. Upper Saddle River, NJ, USA: Prentice-Hall, 2002.

[29] J. de J. Rubio, G. Ochoa, J. Pacheco, and R. Balcazar, "Uniform stable observer for the disturbance estimation in two renewable energy systems," *ISA Trans.*, vol. 58, pp. 155–164, Sep. 2015. doi: 10.1016/j.isatra.2015.05.003.

[30] J. D. Rojo, L. F. Carbajal, and J. D. Velasquez, "Streamflow prediction using a forecast combining system," *IEEE Latin Amer. Trans.*, vol. 13, no. 4, pp. 1035–1040, Apr. 2015. doi: 10.1109/TLA.2015.7106354.

[31] J. de J. Rubio, E. Garcia, C. A. Ibanez, and C. Torres, "Stabilization of the robotic arms," *IEEE Latin Amer. Trans.*, vol. 13, no. 8, pp. 2567–2573, Aug. 2015. doi: 10.1109/tla.2015.7332133.

[32] J. P. A. Taco, P. Bernal, S. G. Jimenez, and A. F. Correa, "Systems identification based on resampling periodic signal implemented in fpga," *IEEE Latin Amer. Trans.*, vol. 13, no. 4, pp. 982–988, Apr. 2015. doi: 10.1109/TLA.2015.7106346.

HAI-PENG REN (M'04) was born in Heilongjiang, China, in 1975. He received the bachelor's degree in industry electrical automation, and the master's and Ph.D. degrees in power electronics and power drives from the Xi'an University of Technology, Xi'an, in 1997, 2000, and 2003, respectively. He has been a Visiting Researcher with Kyushu University, Japan, since 2004. He was a post Ph.D. Research Fellow with Xi'an Jiaotong University, from 2005 to 2008. He was an Honorary Visiting Professor with the University of Aberdeen, Scotland, from 2010 to 2011. He was a Professor with the Xi'an University of Technology, from 2008 to 2018, then moved to Xi'an Technological University after 2018. He held 36 China invention patents, one USA patent, and ten pending invention patents. He has published more than 130 journal and conference papers, including papers in *Physical Review Letter*, the IEEE TIE, the IEEE TIA, the IEEE/ACM TCBB, and the IEEE TBCS. His research interests include nonlinear system control, complex networks, communication with nonlinear dynamics, and system biology.

Prof. Ren obtained the National Invention Award of China, 2012. He obtained three science and technology awards from the Government of Shaanxi province. He received the Young Scientist Award of Shaanxi Province, in 2006. His Ph.D. thesis received the excellent Ph.D. dissertation of Shaanxi, in 2006. He also received the Fok Ying Tong (Research) Fund, in 2008. He was a PI of more than 30 projects, including three from the National Natural Science Foundation of China. He is a Leader of the Innovation Research Team of Shaanxi. He is also a Shaanxi Province Special Support Program for Science and Technology Innovation Leader.

XUAN WANG was born in Xi'an, China, in 1994. She received the bachelor's degree in automation from the Xi'an University of Technology, in 2017, where she is currently pursuing the master's degree in control theory and control engineering. She has published one journal and two conference papers. Her research interest includes hydraulic and pneumatic control systems.

JUN-TAO FAN was born in Xi'an, China, in 1991. He received the bachelor's degree in information science and technology and the master's degree in control science and engineering from the Xi'an University of Technology, in 2013 and 2016, respectively. He is currently an Electrical Engineer with Emerson Network Power (Xi'an) Co., Ltd. He has published three journal papers and applied one China invention patent.

OKYAY KAYNAK (SM'90–F'03) received the Ph.D. degree from the Department of Electronic and Electrical Engineering, University of Birmingham, Birmingham, U.K., in 1972.

From 1972 to 1979, he held various positions within the industry. In 1979, he joined the Department of Electrical and Electronics Engineering, Boğaziçi University, Istanbul, Turkey, where he is currently a Professor Emeritus, holding the UNESCO Chair on mechatronics. He is also a 1000 People Plan Professor with the University of Science and Technology Beijing, China. He has holds a long-term (near to or more than a year) Visiting Professor/Scholar positions at various institutions in Japan, Germany, USA, Singapore, and China. He has authored three books, edited five, and authored or coauthored more than 400 papers that have appeared in various journals and conference proceedings. His current research interests include intelligent control and mechatronics.

Dr. Kaynak recently received the Chinese Government's Friendship Award and the Humboldt Research Prize, in 2016. He has served on the Editorial or an Advisory Boards of a number of scholarly journals.