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A Method for Improving Position Control Performances of a Pneumatic Cylinder Using On-Off Solenoid Valves

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Abstract

Technology using on-off solenoid valves is currently being applied in controlling pneumatic actuators to replace the use of the pneumatic proportional/servo valves. However, controlling the pneumatic actuators using on-off solenoid valves is often difficult due to the low switching frequency of these valves and the high nonlinearity of the pneumatic system. In this paper, we first re-evaluate a controller proposed by Truong in 2020, in which four pneumatic on-off valves were used. We then modify the Truong's controller by dividing the desired position input of the cylinder into two controlled position intervals and each interval will use a separate control law. Seven operating modes of the four on-off solenoid valves combined with the Pulse Width Modulation method (PWM) are used. The experimental research method is carried out. The modified controller is evaluated by comparing it with the Truong's controller and a controller for a pneumatic system using pneumatic proportional valves at the same conditions of the desired position inputs. Comparison results verify the usefulness of the modified controller.

Keywords: Position control, pneumatic cylinder, on-off solenoid valve, pulse width modulation.

1. Introduction

Pneumatic systems are widely used in industrial applications. The systems offer many advantages such as the ability to provide high speed of the actuators, high power-to-weight ratio, low cost, easy installation and maintenance, and unlimited supply of air [1]. Pneumatic systems are also preferred for use in hot or humid environments, where electric driving systems are not realizable [2].

To precisely control the position of the pneumatic actuators, pneumatic proportional valves are often used [3-6]. These valves allow controlling continuously the flow to the actuator and, therefore, it is easy to obtain the desired position of the actuators. However, due to the high cost of the pneumatic proportional valves and their unavailability on the market, it is difficult to apply and maintain the systems using these valves. A cheaper alternative method to the proportional valves is the use of conventional on-off solenoid valves. These valves are widely used in pneumatic transmission systems and their availability in the market is high. However, when using the on-off solenoid valves, precise motion control of the pneumatic actuators is often difficult due to the on-off valve's response time limitation and the high nonlinearities of the pneumatic system [7]. Therefore, it is very important to study suitable control strategies for the position of the pneumatic actuators using the on-off valves.

Until now, several studies have applied on-off solenoid valves to control the position of pneumatic actuators [8-10]. In these studies, one or two valves are often used and therefore the control performances are limited and the valves' opening and closing cycles are often quite large.

In 2020, Truong [11] has introduced a method of controlling the position of a pneumatic cylinder using four on-off solenoid valves and he has shown that this control method can provide relatively good control performances. However, the effectiveness of the Truong's controller has only been verified for large step desired positions. In this study, we re-evaluate the Truong's controller under different desired positions in the whole range of the cylinder stroke and point out the limitations of this controller. We then propose modifications to the Truong's controller to improve the position control quality of the pneumatic system using on-off solenoid valves. The control performances obtained by the modified controller are compared with those of the Truong's controller and a controller for a pneumatic system using pneumatic proportional valves.

2. Experimental System

Fig. 1 and Fig. 2 show respectively the schematic diagram and the image of the experimental system used in this study to study the position control methods of the pneumatic cylinder. The system consists of a double-acting pneumatic cylinder (1) (Model

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CM2L25-300, SMC). The piston diameter, the rod diameter, and the piston stroke are 0.25, 0.1, and 0.3 m, respectively. The piston rod was connected to an external load (3) which slides on a guiding bar (4). The system used four on-off solenoid pneumatic valves (6) (Model 2V025-08, AIRTAC) to control the piston movement. These valves have two ports and two operating positions. These valves were controlled by 24 VDC electrical signals and can deliver the flow rate up to 100 l/min. They have the maximum switching frequency of 8 Hz. Valve 1 and Valve 2 were connected to the left chamber while Valve 3 and Valve 4 were connected to the right chamber of the cylinder. Valve 1 and Valve 3 allow the supply of compressed air from the compressor into the cylinders' chambers while Valves 2 and 4 allow air to be released from the cylinders' chambers into the atmosphere.

To measure the piston position for the control process, the system used a position sensor (2) (Model LWH0300, NOVOTECHNIK) with a measuring range of 300 mm and an accuracy of 0.5% F.S. The position sensor was connected in parallel with the pneumatic cylinder. A Programmable Logic Controller (PLC) (5) (Model PLC S7-1200, SEIMEN) was used to communicate with the position sensor and the valves. The position signal was connected to an analog input of the PLC and the control signals of the four valves u_1 to u_4 were connected to four digital outputs of the PLC. A personal computer (Vostro 260, DELL) was connected to the PLC to acquire the signals and to program the system's controller through the TIA Portal software. The time interval for the program was 0.1 s. Compressed air was fed into the system from the air compressor through an air preparation unit. The supply pressure was set at 6 bar.

The PLC can allow receiving and processing signals only in a minimum time interval of 0.1 s. Therefore, to obtain a more accurate signal from the sensor, an independent Analog to Digital Converter (ADC) (8) (Advantage USB 4711A) was used. The signal from the position sensor was connected to an analog input of the ADC and the ADC was connected to the computer via a USB port. The data collection program was implemented by Microsoft Visual C ++ software. The position signal was recorded to the computer every 0.0011 s.

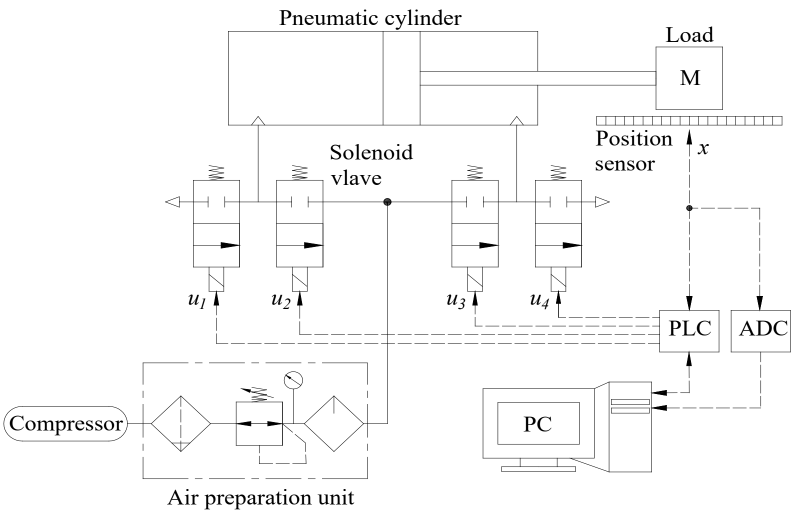

Fig. 1. Schematic of pneumatic system.

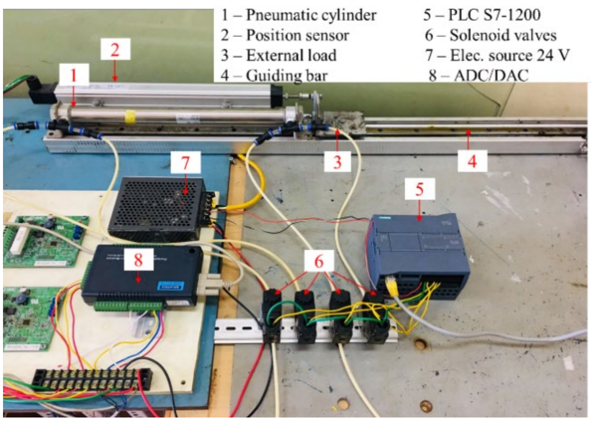

Fig. 2. Image of the experimental system.

3. Controller Design

3.1. Controller Proposed in Previous Study

Truong [11] proposed a position controller of the pneumatic cylinder in the experimental system in Fig. 1. The schematic of the closed-loop control system is shown in Fig. 3 and the control law is shown in Table 1. The control signals of the four pneumatic valves (u_i = 0, i = 1-4) take two values: 1, when the valve is opened (On state), and 0, when the valve is closed (Off state). In this controller, the operating modes of the valves are selected depending on the interval of the position error e between the desired position x_d and the actual position x_t of the piston. Five error intervals of the piston position are considered, including $e \leq -\alpha$, $-\alpha < e < -\beta$, $-\beta \leq e \leq \beta$, $\beta < e < \alpha$, and $e \geq \alpha$. Corresponding to these five error intervals, five operating modes of the four valves are used, including modes M4, M2, M1, M3 and M5 as shown in Table 1. Valves' operation in each mode is shown in Table 2. In Table 1, if the position error $|e| \geq \alpha$, i.e., the error is in the maximum error interval, two modes M4 and M5 are used. In these modes, one valve (Valve 1 or Valve 3) is opened to supply air to the cylinder chambers and one valve (Valve 2 or Valve 4) is opened to exhaust air from the cylinder chambers to the atmosphere. By using these modes, the maximum flow rates are obtained, and this causes the piston to move as quickly as possible to the desired position to achieve minimum rise time. When the position error reduces and falls into the error interval $\beta < |e| < \alpha$, two models M2 and M3 are used. In these modes, only one Valve 1 or Valve 3 is opened intermittently according to a certain pulse to supply air from the compressor into the cylinder chambers while other valves are closed. The cycle of the valve opening pulse is selected as follows:

$$T = \frac{e}{\lambda} \tag{1}$$

where λ is the coefficient that determines the width of the control pulse. The value of λ was chosen as 0.0001. This value was chosen based on the maximum switching frequency of the valve. By using Modes 2 and 3, the air is supplied intermittently into the cylinder chamber, and thus, the piston is gradually decelerated when it is close to the desired position. Finally, when the position error is in the interval $|e| \leq \beta$, i.e., the smallest allowable error range, mode M1 is used. By using this mode, the piston is fixed at the desired position.

In Table 1, α and β are the limited position errors. They are given by

$$\begin{aligned} \alpha &= 0.8\,|\,x_d - x_t\,| \\ \beta &= 0.02\,|\,x_d - x_t\,| \end{aligned} \tag{2}$$

Fig. 4a shows control results obtained by the experiment of the controller with a constant desired position $x_d = 250$ mm. The results show that the piston can track the desired position with fast setting time (0.35 s), small overshoot (2%) and small position error in steady state (1.5 mm, corresponding to a relative position error of 0.6%). For a low desired position $x_d = 50$ mm, the controller exhibits poor control results as shown in Fig. 4b. The position overshoot obtained in this case is very high (130%). This result indicates that the controller proposed by Truong [11] is only suitable for the highly desired positions and is not suitable for the low desired positions.

Table 1. Control law in the Truong's controller

Position error $e = x_t - x_d$	**Operating modes**	**Cycle**
$-\alpha \geq e$	M4 (Valves 1 and 4 ON)	
$-\alpha < e < -\beta$	M2 (Valve 1 ON)	$T = e/\lambda$
$-\beta \leq e \leq \beta$	M1 (Valves 1,2,3,4 OFF)	
$\beta < e < \alpha$	M3 (Valve 3 ON)	$T = e/\lambda$
$\alpha \leq e$	M5 (Valves 2 and 3 ON)	

Table 2. Operating modes of four pneumatic valves

	M1	M2	M3	M4	M5
u_1	0	1	0	1	0
u_2	0	0	0	0	1
u_3	0	0	1	0	1
u_4	0	0	0	1	0

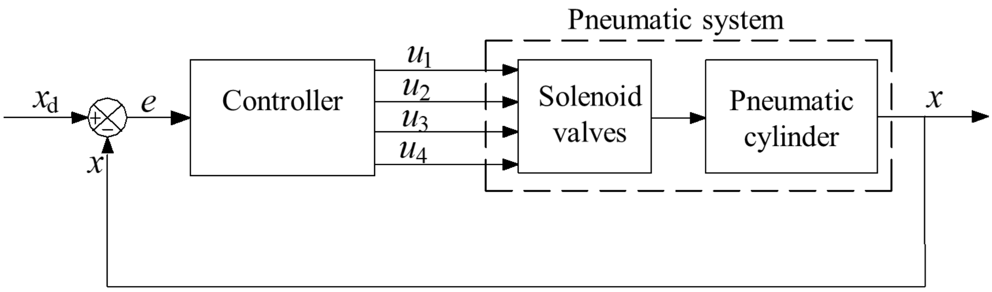

Fig. 3. Schematic of the closed-loop control system.

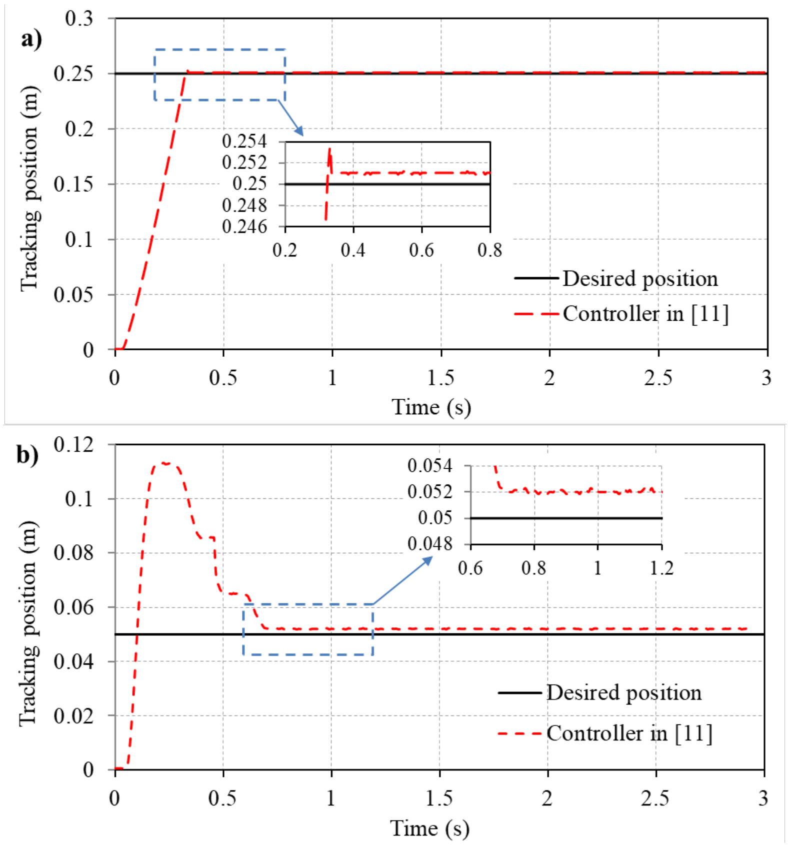

Fig. 4. Control results by the controller in previous study with different constant desired position inputs: a) x_d = 250 mm, b) x_d = 50 mm.

3.2. Modified Controller

In the controller proposed by Truong [11], only one switching rule of the operating modes of the four valves was applied to all the desired position. When piston moves with high speed by using the modes M4 or M5, switching the state of valves from mode M4 to mode M2 or from mode M5 to mode M3 is difficult to reduce immediately the piston speed and it causes high overshoot value. Therefore, in this study, we modify the controller proposed by Truong [11] to improve the control ability of the piston position. Table 3 shows the switching rules of the operating modes of the four valves in the modified controller.

We divide the desired position into two different ranges: small, desired position range $x_d \le$ 50 mm and large desired position range $x_d >$ 50 mm. Each range will have an individual control law. To the small desired position range $x_d \le$ 50 mm, when the position error $e \le -\alpha$, instead of using the mode M4 of the controller in [11], we use a new mode M6 of the valves in which the air is supplied into both the cylinder chambers at the same time (Valves 1 and 3 are opened). Due to the differences in the area of the piston between the two cylinder chambers, the piston can move at a low speed.

Table 3. Control law of the modified controller

Desired position range: $x_d \le$ 50 mm		
$e = x_d - x_t$	**Operating modes**	**Pulse cycle**
$-\alpha \ge e$	Valve 1 ON, Valve 3 ON	
$-\alpha < e < -\beta$	Valve 1* ON	T
$-\beta \le e \le \beta$	Valves 1,2,3,4 OFF	
$\beta < e < \alpha$	Valve 3* ON	T
$\alpha \le e$	Valve 1 ON*, Valve 3 ON	T
Desired position range: $x_d >$ 50 mm		
$e = x_d - x_t$	**Operating modes**	**Pulse cycle**
$-\alpha \ge e$	Valve 1 ON, Valve 4 ON	
$-\alpha < e < -\beta$	Valve 1 ON, Valve 3* ON	T
$-\beta \le e \le \beta$	Valves 1,2,3,4 OFF	
$\beta < e < \alpha$	Valve 3 ON, Valve 1* ON	T
$\alpha \le e$	Valve 3 ON, Valve 2 ON	

Note: '' Indicating the use of the Pulse Width Modulus method*

When the position error falls into the range, we use the mode M2 but Valve 1 is opened intermittently according to a pulse with a cycle *T*. By using this mode, the piston speed is quickly reduced and thus it can eliminate the overshoot behavior that occurs by using the controller in study [11]. On the contrary, when the position error $e \geq \alpha$, the mode M5 in [11] is replaced by a new mode, in which Valve 3 is opened continuously and Valve 1 is opened intermittentl with a *T* cycle pulse. Next, when the positon error falls into the range $\beta < e < \alpha$, the mode M3 used in [11] is replaced by a mode that Valve 3 is opened intermittently with a *T* cycle pulse. Finally, when the position error falls into the smallest range $-\beta \leq e \leq \beta$ the mode M1 is used, similar to the control mode used in [11].

To the large desired position range $x_d > 50$ mm, the control law in [11] is used. However, to improve control performance, we modify the operating modes of the valves in the two error intervals $-\alpha < e < -\beta$ and $\beta < e < \alpha$. For the controller in [11], if only Valve 1 is opened and Valve 3 is closed (mode M2), or Valve 3 is opened and Valve 1 is closed (mode M3), the piston speed will reduce rapidly when the mode M4 or mode M5 is switched to the mode M2 or the mode M3. This results in a jerky movement of the piston. So, in the modified controller, when the position error $-\alpha < e < -\beta$, Valve 1 is opened continuously to supply air to the cylinder chamber while Valve 3 is opened intermittently with a *T* cycle pulse. And vice versa, if the positon error $\beta < e < \alpha$, Valve 3 is opened continuously to supply air to the cylinder chamber while Valve 1 is opened intermittently with a T cycle pulse. These combinations of the valves can gradually reduce the piston speed, leading the piston to run smoothly when switched from fast pusher mode (modes M4 or M5). The control parameters α, β, and *T* are also calculated from equations (1) and (2).

4. Control Results

To evaluate the control performances of the modified controller, its control performances are compared with those of the controller proposed in [11]. Step and sinusoidal position inputs are used. Fig. 5 shows the control results of the piston position of both the modified controller and the controller in [11] with a step position input x_d = 250 mm. The comparison result indicates that the modified controller offers faster rise time and smaller position tracking error than those of the controller in [11]. The rise time achieved by the modified controller is 0.29 s while that achieved by the controller in [11] is 0.32 s. In addition, the position error is 0.5 mm for the modified controller and is 1.5 mm for the controller in [11]. Fig. 6 compares the position control results of the two controllers at a low desired position x_d = 30 mm. The results also show that the control quality is significantly improved when using the modified controller. To the controller proposed in [11], the position overshoot is 180% and the maximum position error in steady state is 2 mm. However, to the modified controller, the position overshoot is 3.33% and the maximum position error in steady state is 1.36 mm.

Fig. 7 shows the comparison of the control performances of the two controllers with a sinusoidal desired position x_d = 150 + 100sin ($2\pi ft$) mm at the frequency f = 0.1 Hz. Fig. 7a shows the position tracking results and Fig. 7b shows the tracking errors. Fig. 7a shows that the piston can follow the desired position with the same frequency for both controllers. However, the modified controller provides much better control results comparing to those of the controller in [11]. For the modified controller, the piston moves smoothly and creates small control error (maximum error in steady state is 6 mm). While, for the controller in [11], the piston moves jerkily and creates large control error (maximum error in steady state is 22 mm).

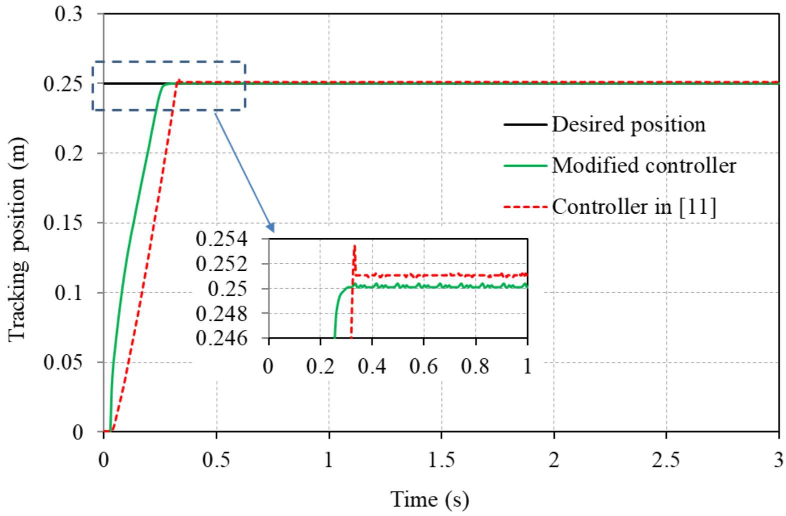

Fig. 5. Control results of two controllers with a desired position input x_d = 250 mm.

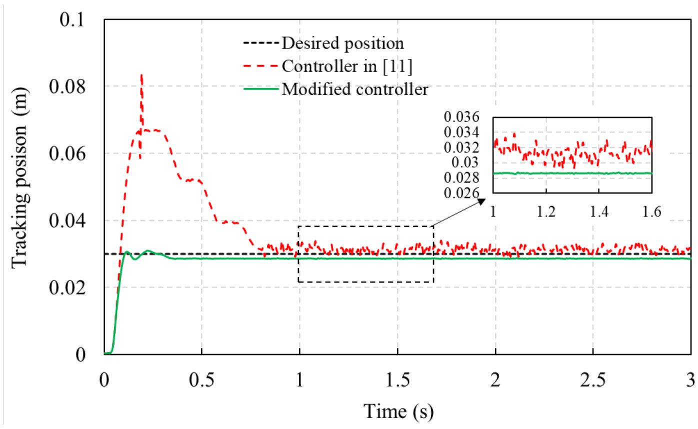

Fig. 6. Control results of two controllers with a desired position input x_d = 30 mm.

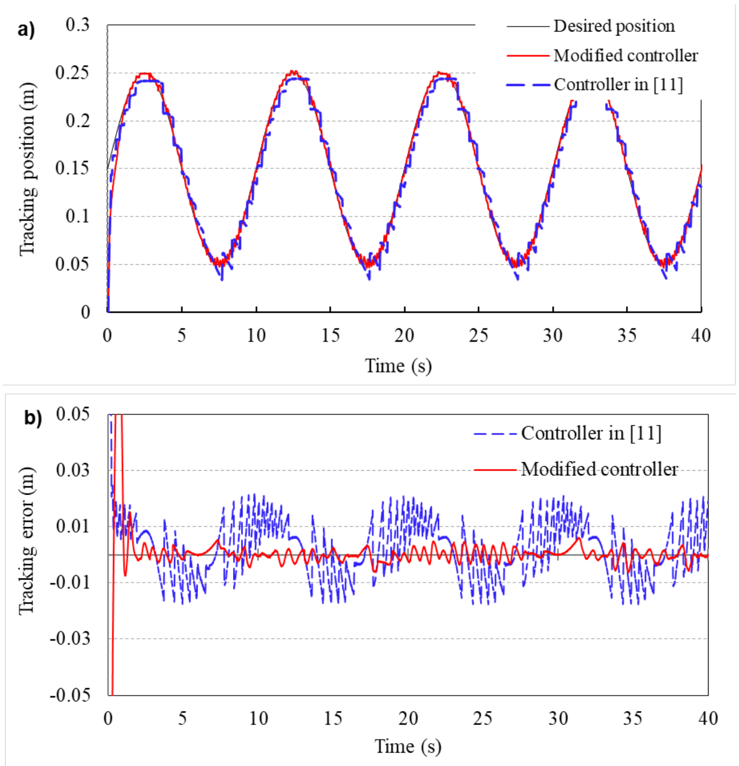

Fig. 7. Control results of two controllers at a desired position input $x_d = 150 + 100\sin(2\pi f t)$ mm with the frequency f = 0.1 Hz.

To further evaluate the usefulness of the modified controller, the control performances of the modified controller are compared with those obtained by a controller for a pneumatic system using pneumatic proportional valves in [12]. In [12], the position control system using two pneumatic proportional valves and a multi-surface sliding control method combined with friction compensation was applied. Fig. 8 shows the comparison of the control performance between the modified controller and the controller in [12]. The control results are obtained with a sinusoidal desired position x1d = 150 + 100sin ($2\pi ft$) mm with frequency f = 0.1 Hz. The dashed line shows the control result of the modified controller, and the dot-dash line shows the control result achieved by the control system using the proportional valves [12]. The results indicate that the modified controller can provide equivalent control errors to those of the controller in [12]. The maximum errors in steady state of both controllers are less than 6 mm. With the modified controller, the piston can track the desired position smoother than that of the controller in [12]. However, with the high frequency cases, for an example at 0.5 Hz as shown in Fig. 9, the modified controller provides much larger control errors than that obtained by the controller in [12]. The switching frequency of the on-off valves used in the experimental system is low and therefore the valves' response is not fast enough to respond to the high frequency of the desired position input. This states that the system with the four on-off solenoid valves and the modified controller proposed in this study are only suitable for either step desired positions or variable desired positions with low operating frequency.

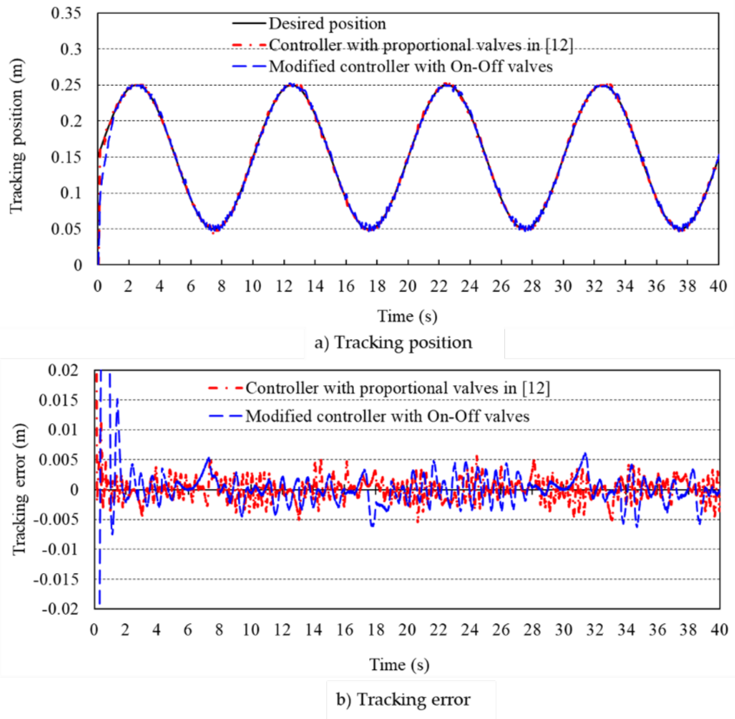

Fig. 8. Comparison of control performances between the modified controller and the controller of the system using proportional valves at a desired position input $x_d = 150 + 100\sin(2\pi ft)$ mm with the frequency $f = 0.1$ Hz.

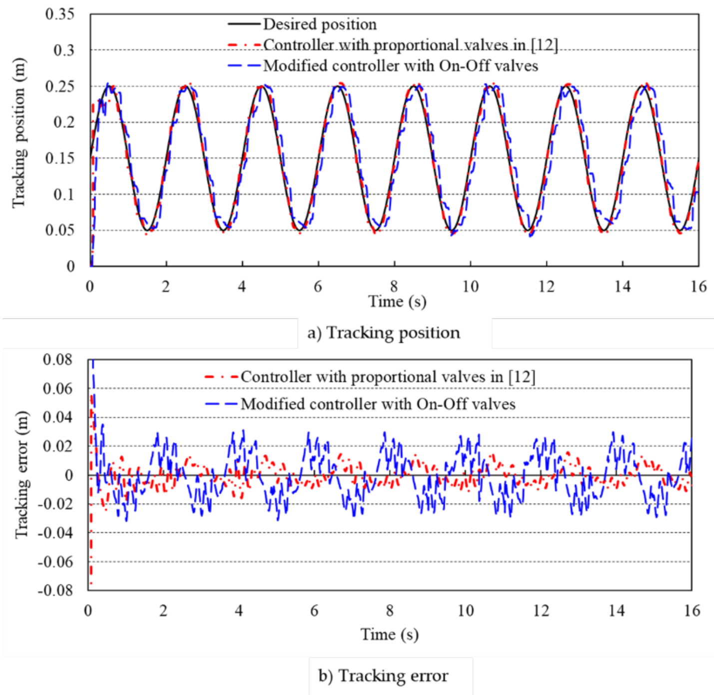

Fig. 9. Comparison of control performances between the modified controller and the controller of the system using proportional valves at a desired position input $x_d = 150 + 100\sin(2\pi ft)$ mm with the frequency $f = 0.5$ Hz.

5. Conclusion

In this study, we re-evaluated a pneumatic cylinder position controller using 4 pneumatic on-off valves introduced by Truong in 2020 and proposed a modified controller based on Truong's controller. Experimental studies were carried out. The study showed the following results: i) the Truong's controller causes large overshoots with low desired position inputs; ii) the modified controller can significantly improve the position control performances including overshoot and control error compared with the Truong's controller; iii) the modified controller can provide equally good control performances to those of the controller using the proportional valves at the operating frequencies lower than or equal to 0.1 Hz.

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