Modeling Inverse Airflow Dynamics Towards Fast Movement Generation using Pneumatic Artificial Muscle with Long Air Tubes

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Abstract—This study entailed the derivation of an inverse airflow dynamics model that causes pneumatic transmission delay due to the long air tube in pneumatic artificial muscle (PAM) control. The inverse model of the derived airflow dynamics was used in designing a feedforward pressure control command for the proportional pressure control valve. We also modulated the feedback command based on the proposed pressure control for precise force tracking performance. The proposed methods were validated by varying the tube lengths in the pressure tracking task and force tracking task. The tracking tasks with the sinusoidal desired profile of 2.5 Hz were successfully achieved using the proposed method. The results indicate that the proposed method can compensate for the delay due to the airflow dynamics in the long air tube and improve the control performance.

Index Terms—pneumatic artificial muscle, force control, air transmission tube

I. INTRODUCTION

Pneumatic Artificial Muscles (PAMs) have been used in soft robotics research fields [1]–[3] and as actuators for rehabilitation devices [4]–[7] for their mechanical compliance and high power-weight ratio [8]. Furthermore, thanks to large power to weight ratio, PAM systems do not require heavy reduction gears, significantly losing joint backdrivability. In recent years, humanoid robots have been developed that use this flexibility and have succeeded in generating supple human-like movement [9], [10]. However, PAM has the difficulty of precise force generation, especially for fast movements and accurate control to follow a high-frequency trajectory has not been achieved. To improve the motion control performance of robots with PAM, we need to cope with the difficulty in controlling PAM actuators.

The air valve is often placed at a considerable distance from the PAM to reduce the inertia of a robot system and end-effector weight [7], [11]. In experimental setups for skeletal muscle robots using PAM, thin and soft bendable tubes are practical because tubes are installed in the narrow spaces such as the inside of the robot body [10]. These tubes cause pressure transmission delays that degrade the control performance [12]. However, how to compensate for the pressure transmission delay in PAM control has not been carefully studied.

This study derives airflow dynamics model to compensate for pneumatic transmission delay due to the long air tube in the PAM system. The inverse model of the derived airflow dynamics is used to design feedforward pressure control commands for a proportional pressure control valve. We also modulated the feedback force commands to achieve precise force tracking performance even with long pneumatic transmission lines. The precise control performance of the proposed method was validated with different lengths of tubes. The contributions of this paper are as follows:

- Derives an inverse airflow dynamics model for feedforward PAM control.
- Develops a pressure feedforward control approach to improve the force control performance with the airflow model-based control.
- Empirically validates the stability of the proposed PAM pressure control method.

This paper is organized as follows: Section II introduces the related studies. In section III, we present the PAM system.
model composed of a control valve, PAM, and pressure transmission via tube. The proposed control method is introduced in section IV. In section V, experimental setups are explained, whereas in section VI, the experimental results are presented. In section VII, we discuss the performances of the developed control system. Finally, the conclusions are presented in section VIII.

II. RELATED WORKS

In position control of pneumatic systems, it is common to measure the pressure of the actuator and design feedback command to the valve [13], [14]. However, many robot applications that need to use long air tube do not allow us to put a pressure sensor on the actuator side. Therefore, in this study, we aim to control the pneumatic control system with long transmission lines in which the feedback based on the measured pressure inside the actuator is not feasible.

Chou et al. attempted to model pressure transmission as a relationship between the inlet and outlet pressures of the tube [15]. Tagami et al. controlled the inner pressure of PAM, considering the pressure transmission [12]. However, only the pressure loss caused by the friction from the tube’s wall was accounted for in the model. In other words, only the simple manually-tuned first-order transmission delay was considered. Krichel and Sawadony divided the tube model into multiple finite-volume elements and derived a discrete dynamics formulation [16]. Melih et al. constructed the force control method of a pneumatic cylinder with a long transmission line using Krichel’s model [17]. However, due to the time-varying relationship between force and pressure in PAM system, the control approach proposed in [17] cannot be directly applied to PAMs. To control PAMs, we need to explicitly take the time-varying contraction rate of the actuator into account since the pressure-to-force relationship depends on the contraction rate. As a matter of fact, the previous studies have focused on pneumatic cylinder control and a method to handle the difficulty of the time-varying pressure-to-force relationship has not been sufficiently explored.

Our idea to cope with this difficulty is using the inner-loop pressure controller rather than the inner-loop force controller adopted in the previous study [17]. In the pressure control domain, we can derive the inverse airflow model to compensate the delay due to the pneumatic transmission even with considering the contraction rate. In deriving the inverse airflow model, we adopted the airflow dynamics model proposed in previous studies [16], [17]. To the best of our knowledge, this is the first study that shows successful experimental results to compensate delay due to the pneumatic transmission of PAM system.

Proportional flow control valves are more popular than proportional pressure control valves in the pneumatic field. Although this study primarily focuses on proportional pressure control valves due to the safety and usability, there is no significant difference between them. The application of the proposed approach to proportional flow control valves and its control results are also presented in Section VII-B.
$T = 298 \text{K}$ is room temperature. We adopted the approximated friction model as follows [17]:

$$f (P_V, \dot{m}, P_P) = \frac{32\eta}{D^2} + 0.158 \frac{Re (P_V, \dot{m}, P_P)^{3/4}}{D^2} \eta,$$

(4)

where $D$ is the diameter of the tube and $\eta = 1.52 \times 10^{-5} \text{m}^2/\text{s}$ is kinematic viscosity. $Re$ is Reynolds number and $\rho$ is the air density in the tube.

Airflow dynamics consists of (2) and (3). A tube is parameterized by $L$ and $S$. Airflow dynamics consists of (2) and (3). The effect of the tube length $L$ appears in (2) as a coefficient on pressure and in (3) as a part of the total volume $V_{total}$.

**B. Pressure-to-force model**

The relationship between the force $F$ of the PAM and $P_P$ and $\epsilon$ is highly nonlinear and difficult to model accurately. Hildebrandt et al. expressed it as:

$$F (P_P, \epsilon) = \left( \sum_{i=0}^{2} b_i \epsilon^i \right) (P_P - P_{atm})$$

$$+ \left( \sum_{i=0}^{3} d_i \epsilon^i + d_4 \epsilon^2 \right),$$  

(5)

using polynomial approximation up to the third order of $\epsilon$ [14]. $P_{atm} = 0.1013 \text{MPa}$ is the atmospheric pressure.

The accuracy of the PAM model can be improved by adding a hysteresis term that includes $\dot{\epsilon}$ [21]. However, due to the vibration caused by $\epsilon$, it is not expected to improve the control performance with a simple implementation. Furthermore, since hysteresis is not the focus of this study, a static model was chosen.

**IV. CONTROLLER DESIGN**

The block diagram of the proposed pressure control valve and the control method of the valve is described in the Appendix.

**B. Pressure controller based on the inverse airflow dynamics**

We use the inverse dynamics of the PAM-tube system introduced in Section III-A to derive the desired pressure $P_d$ for the proportional pressure control valve. From the inverse dynamics of (3), the reference mass flow rate $\dot{m}_r$ inside the tube is derived as

$$\dot{m}_r = \frac{V_{total}}{kRT} \left( \frac{dP_r}{dt} + k \frac{\dot{V}_{pam}}{V_{total}} P_r \right).$$

(8)

The desired valve pressure $P_d$ is then determined as the inverse model of (2) as

$$P_d = P_r + \frac{L}{S} \left( \frac{d\dot{m}_r}{dt} + f (P_V, \dot{m}_r, P_r) \dot{m}_r \right).$$

(9)

Pressure control targeting $P_x$ with $P_d$ as input is composed of (8) and (9). $P_V = P_d$ is achieved by the proportional pressure control valve and the control method of the valve is described in the Appendix.

**C. Stability analysis**

To analyze the stability of the proposed pressure control method, we evaluated the linearized approximations of the PAM control system around a state trajectory: We summarize (3) and (2) as follows:

$$\frac{d\bar{x}}{dt} = A (\epsilon, \bar{x}, P_V) \bar{x} + BP_V$$

(10)

$$A (\epsilon, \bar{x}, P_V) = \begin{bmatrix} -f (x, P_V) & -\frac{\dot{\epsilon}}{2} \\ \frac{k_1 RT}{V_{total}} & \frac{k_2 \dot{V}_{pam}}{V_{total}} \end{bmatrix}$$

$$B = \begin{bmatrix} \frac{\dot{\epsilon}}{2} \\ 0 \end{bmatrix}, \bar{x} = \begin{bmatrix} \bar{m} \\ P_P \end{bmatrix}.$$  

(11)

From (8), (9), and (10), the differential equation of $\bar{x}_r = [\bar{m}_r, \bar{P}_r]^T$ is expressed as

$$\frac{d\bar{x}_r}{dt} = A (\epsilon, \bar{x}_r, P_V) \bar{x}_r + BP_d.$$  

(12)

By subtracting (11) from (10) and assuming $P_V \approx P_d$,

$$\frac{d}{dt} (x - x_r) = A (\epsilon, x, P_V) x - A (\epsilon, x, P_V) x_r$$

(13)

is obtained. Defining $f' (x, P_V)$ as $f' (x, P_V) = f (x, P_V) \dot{m}$, $f' (x, P_V) \dot{m}$ is linearized using the Taylor expansion of $f' (x, P_V)$ around a certain point $\bar{x} = [\bar{m}, \bar{P}_V]^T$ with the input $P_V$ at that time as

$$f (x, P_V) \dot{m} = f' (\bar{x}, \bar{P}_V) (\dot{m} - \bar{m}) + \frac{\partial f' (\bar{x}, \bar{P}_V)}{\partial m} (\dot{m} - \bar{m})$$

(14)

Similarly, $f (x, P_V) \dot{m}_r$ is linearized as

$$f (x, \bar{P}_V) \dot{m}_r = f' (\bar{x}, \bar{P}_V) \dot{m}_r + \frac{\partial f' (\bar{x}, \bar{P}_V)}{\partial m_{\dot{m}_r}} (\dot{m}_r - \bar{m})$$

(15)

$$+ \frac{\partial f' (\bar{x}, \bar{P}_V)}{\partial P_{\dot{m}_r}} (P_r - \bar{P}_r).$$  

(16)
By substituting (13) and (14) to (12), the dynamics of $e_x = x - x_o$ is expressed as follows:

$$\frac{de_x}{dt} = A'(t)e_x,$$

$$A'(t) = A'(\epsilon, \bar{x}, \bar{P}_V) = \begin{bmatrix} -\frac{\partial f'(\bar{x}, \bar{P}_V)}{\partial \bar{P}_V} - S - k_2 \frac{\partial P}{\partial P} \\ -k_1 \frac{\partial P}{\partial P} \end{bmatrix}. $$

For (15), the transition matrix $\Phi_{A'}(t, \tau)$ $(t \geq \tau)$ can be defined as follows:

$$\frac{d}{dt} \Phi_{A'}(t, \tau) = A'(t) \Phi_{A'}(t, \tau),$$

$$\Phi_{A'}(t, t) = I.$$

Then, the error at time $t$, $e_x(t)$, can be derived from the error at time $\tau$, $e_x(\tau)$, as

$$e_x(t) = \Phi_{A'}(t, \tau) e_x(\tau).$$

Here, we discuss the stability of the error dynamics (15) with the transition matrix $\Phi_{A'}$ as suggested in [22]. If the condition

$$\exists M > 0 \text{ s.t.} \| \Phi_{A'}(t, \tau) \| \leq M \quad \forall t, \tau$$

is satisfied, then the error dynamics (15) is considered Lyapunov stable. Furthermore, if the condition

$$\lim_{t \to \infty} \| \Phi_{A'}(t, \tau) \| = 0$$

is satisfied, then (15) is considered asymptotically stable.

V. EXPERIMENT

We conducted two trajectory-tracking experiments to evaluate the proposed method. First, we assessed the pressure control performance of the inner-loop controller of our PAM control system (see Fig.3). Then, we evaluated the force control performance of the whole PAM control system, including the inner and outer-loop controllers.

A. Experimental setup

Figure 4 shows our experimental setup and sensor allocations. The PAM bladder (cut from Festo DMSP-20, diameter: 20mm, length: 325mm) plugged with custom-made PAM both ends was mounted on a stage and connected to a proportional pressure control valve (Norgren VP5010B) with an air tube (inner diameter: $D = 5\text{mm}$). The valve pressure $P_V$ was measured by a pressure sensor attached to the exit port of the proportional pressure control valve. We also measured the pressure inside the PAM $P_P$ using a pressure sensor connected to the PAM via a thin air tube. The observed value of $P_P$ was used only for evaluation and not for control. The contracted length of the PAM $\Delta l$ was measured by a linear encoder attached at the bottom of a linear slider to derive the contraction rate $\epsilon$. The generated force $F$ by the PAM was measured by a load cell attached to the PAM tip. The control frequency of the system was 250 Hz. The supply pressure from the compressor (Hitachi SRL-A3.7DW) to the control valve was maintained around 0.78MPa.
B. Controller settings

For the pressure control task, we compared the following two control methods: 1) Our proposed method using (9) to derive the desired pressure command \( P_d \) for the proportional pressure control valve. 2) Baseline method that did not explicitly consider the existence of the long air tube. The desired pressure commands \( P_d \) for the two approaches are derived as follows:

\[
P_d = \begin{cases} 
P_r + \frac{L}{S} \left( \frac{dn_i}{dt} + f(P_V, \dot{n}_r, P_r, \dot{n}_r) \right) & \text{(Proposed)} \\
P_r & \text{(Baseline)}
\end{cases}
\]

We derived \( \frac{dn_i}{dt} \) by the derivative of \( \dot{n}_r \). Therefore, we applied a first-order low-pass filter \( 1/(1 + s \tau) \) with a time constant \( \tau = 0.05 \) to \( \frac{dn_i}{dt} \), to eliminate the high-frequency noise due to the numerical differentiation.

For the force control task, we compared the two control methods: 1) Our proposed method and 2) Baseline method. Feedback gains for each force controller were tuned manually for the sinusoidal desired trajectory with the air tube length of 5 m and \( \nu = 0.5 \text{Hz} \). The gains of proposed method and baseline method were tuned based on Ziegler–Nichols method [23]. Owing to the lack of differential operation, the gains of the baseline method are less prone to oscillations, which allows for higher gains than that proposed. The gains are listed in Table I.

<table>
<thead>
<tr>
<th>TABLE I: Feedback gain settings</th>
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<tr>
<td>( K_p )</td>
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<td>( K_i )</td>
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The identified parameters used in the controllers are shown in Table II. The value of \( V_0 \) was obtained from measurement and the other parameters were identified from the prior experiment with a 4 m tube. For the identification of \( V_1 \) and \( V_2 \), the minimization of the square error between measured \( P_P \) and the estimated value \( \hat{P}_P \) was conducted to experimental data when valve pressure was generated as the sinusoidal waves of 1, 2, 3, 4 Hz under the free condition. \( \hat{P}_P \) is estimated by (2) and (3) with the estimated mass flow rate \( \hat{n}_r \). For the identification of \( d_i \) \((i = 0, 1, \ldots, 4) \) and \( b_i \) \((i = 1, 2, 3) \), the minimization of the square error between measured \( F \) and the estimated value \( \hat{F} \) was conducted to experimental data when PAM was extended slowly and gradually under constant PAM pressure conditions (0.2 MPa to 0.8 MPa in 0.1 MPa increments).

<table>
<thead>
<tr>
<th>TABLE II: Identified parameters</th>
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<tr>
<td>( V_0 = 1.02 \times 10^{-4} \text{m}^3 )</td>
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<tr>
<td>( V_1 = 1.04 \times 10^{-3} \text{m}^3 )</td>
</tr>
<tr>
<td>( V_2 = -2.35 \times 10^{-3} \text{m}^3 )</td>
</tr>
<tr>
<td>( b_0 = 2.83 \times 10^{3} \text{N/mPa} )</td>
</tr>
<tr>
<td>( b_1 = -9.26 \times 10^{3} \text{N/mPa} )</td>
</tr>
<tr>
<td>( b_2 = 3.13 \times 10^{3} \text{N/mPa} )</td>
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C. Trajectory tracking tasks

We evaluated the pressure and force control performances using sinusoidal trajectory tracking experiments.

1) Pressure tracking task: First, the pressure tracking performance was evaluated based on the sinusoidal trajectory tracking tasks with different bias levels and frequencies: \( P_r (t) = A_p \sin 2\pi \nu t + B_p \), where the amplitude \( A_p = 0.08 \text{MPa} \), bias level \( B_p = 0.36, 0.46, 0.56 \text{MPa} \), and frequencies \( \nu = 0.5, 1.0, 1.5, 2.0, 2.5 \text{Hz} \). The values of \( A_p \) and \( B_p \) were set around the pressure range of the force tracking task in Section V-C. As in Fig. 5, we conducted the target pressure tracking task in three different conditions:

i) Free condition: one side of the PAM was fixed while the other could freely move on the ball linear guide.

ii) Fixed condition: both sides of the PAM were fixed.

iii) Antagonistic condition: one side of the PAM was fixed, and the other was connected to the antagonistic PAM. The inner pressure of the antagonistic PAM was maintained at a constant value 0.4 MPa so that we can verify the pressure and force tracking performance of the proposed method with wide range of PAM contraction rate in our experimental setup. In the experimental setup of this paper, the pressure was set to stretch and contract in response to the range of forces on the controlled PAM. The antagonistic condition emulated the agonistic-antagonistic system composed of two PAMs [24]. The pressure tracking tasks at every condition were conducted with a 7 m long tube. In practice, we often used several meter long air tube in our rehabilitation studies [5]–[7], [11], [25].

To check for length dependence, 3 m and 5 m long tubes were also used for the experiments under the free condition with the bias pressure \( B_p = 0.46 \text{MPa} \).

We further evaluated the pressure tracking performance to the desired trajectory composed of the two sine functions with a 7 m tube under the free condition: \( P_r = 0.07 \sin 2\pi \cdot 1.3t + 0.05 \sin 2\pi \cdot 2.2t + 0.39 \text{MPa} \).

2) Force tracking task: We then evaluated the force tracking performance with sinusoidal target trajectory: \( F_d = A_f \sin 2\pi \nu t + B_{f} \), where the amplitude \( A_f = 100 \text{N} \), bias levels \( B_f = 400 \text{N} \) and frequency \( \nu = 0.5, 1.0, 1.5, 2.0, 2.5 \text{Hz} \). Further, 3, 5, and 7 m tubes were used. The force tracking task was conducted under the antagonistic condition.
VI. RESULTS

A. Pressure tracking task

The error between the reference and actual PAM pressure ($P_r$ and $P_p$, respectively) was evaluated. Fig. 6 compares the control performances of the each method with root mean square error (RMSE). Fig. 7 shows the results of pressure control at 3m and 5m tubes. The proposed method improved the control performance under all three conditions.

Fig. 8 shows an example of the actual pressure tracking performance at $\nu = 2.5$Hz and $B_p = 0.46$MPa. Our proposed method compensates for the delay, while the tracking performance of the baseline approach shows an apparent delay in the reference pressure profile. To further improve the tracking performance of the proposed method, more precise modeling of the system would be necessary.

An example of estimating the value of $P_p$ using the proposed airflow dynamics model constructed in Section III was shown in Fig. 9 for the validation of the proposed method. Fig. 9 shows the behavior of $P_V$ and $P_p$ along with the estimated values of $P_p$ at that time during pressure control (7m tube, $\nu = 2.5$Hz, and $B_p = 0.46$MPa). The delay caused by the tube can be observed from the relationship between $P_V$ and $P_p$. The proposed model (2), (3) with the observed value of $P_V$ can estimate the delay.

Fig. 10 shows the pressure tracking result for the desired trajectory of two sine waves combination. The delay was compensated and even minute desired pressure changes were reproduced by the proposed method.

B. Force tracking task

Fig. 11 shows the tracking performances for a desired sinusoidal trajectory. The control performances of our proposed method were robust against changes of the tube lengths and the frequency, whereas the tracking error of the baseline method increased depending on the situation. When using the baseline method, superiority over the proposed method was observed due to the higher gain under low frequency and short tube conditions, however the error significantly increased with longer tubes and higher frequency.

Fig. 12 shows the force tracking profiles with the 7 m
Fig. 11: Tracking performances for desired sinusoidal force trajectory with three different tube lengths: 3, 5, and 7m. The control performances of our proposed method are robust against changes in tube lengths and frequency.

Fig. 12: Example of force control experiments with 7 m length tube condition at $\nu = 2.5$Hz. Proposed method can track without delay, while baseline method causes significant delays.

Fig. 13: Bode plot of force control results with 7 m air tube. No delay is observed in proposed method, while baseline method was significantly delayed from 1.5Hz as shown in Phase plot.

C. Empirical stability analysis

Fig. 14 shows the matrix norm of $|\Phi A'(t, \tau)|$ from each $\tau$ are contained in the light orange region. Therefore, all norms converged to zero. This result empirically shows that the PAM system controlled by our method is asymptotically stable.

$|\Phi A'(t, \tau)|$ shows different behaviors in Fig.14 for $\tau = T_1 (= 7.91s)$ and $\tau = T_2 (= 8.20s)$. To investigate their difference, the vector fields of (12) with $A'(t = \bar{t})$ as a constant matrix are plotted in Fig. 15. At $t = T_1$, $Re (t = T_1)$ is low and the air flow is characterized by laminar flow. Moreover, $f (t = T_1)$ is also low, because of which the eigenvalue of $A'(t = T_1)$ contains imaginary numbers and the trajectory converges spirally as shown in Fig. 15a. At $t = T_2$, $Re (t = T_2)$ is high and the air flow is characterized by turbulent flow. Moreover, $f (t = T_2)$ is high, and the corresponding eigenvalue $A'(t = T_2)$ contain no imaginary numbers, making the trajectory converge without vibration as shown in Fig. 15b.

VII. Discussion

A. Limitation of proposed force control method

The proposed method in this study is effective in scenarios where the influence of the airflow dynamics is significant. However, limitations attributed to the low gains resulting from the presence of differential operations can hinder the performance of force control in conditions where the influence of the airflow dynamics is small, such as small pressure changes, short tubes, and small end-volume changes. In fact, when the force control task was performed under fixed conditions, as
shown in Fig. 16 the performance was not better than that of the baseline method; however, this is an unlikely situation in which PAM can be used. By adding filters to differential operations or combining the proposed pressure feedforward control with other force control methods, it is possible to achieve robust control in any scenario.

**B. Application of proposed method to proportional flow control valve**

![Diagram of transmission line](image)

Fig. 17: Two compartment model of transmission line for proportional flow control valve. Input of PAM system is mass flow rate generated by valve.

The proposed method was additionally applied to a proportional flow control valve to further evaluate whether our proposed approach is useful even with the different physical input to the system. With the proportional pressure control valves, the input to the system is pressure, whereas, with proportional flow control valves, the input to the system is mass flow. Therefore, the PAM system is represented by (2), (3) and

\[
\frac{d}{dt} P_v = \frac{RT}{SL/2} (\dot{m}_v - \dot{m}), \tag{20}
\]

where \( \dot{m}_v \) is the mass flow rate provided by the flow control valve as shown in Fig. 17. (2), (3) and (20) can be summarized as

\[
\frac{dx_f}{dt} = A_f(\epsilon, x_f) x_f + B_f \dot{m}_v \tag{21}
\]

\[
A_f(\epsilon, x_f) = \begin{bmatrix}
0 & -\frac{RT}{SL/2} \\
\frac{s}{f} & -f(x_f) & \frac{s}{f} \\
0 & \frac{k_1RT}{V_{total}} & -\frac{k_2}{V_{total}}
\end{bmatrix}
\]

\[
B_f = \begin{bmatrix}
\frac{RT}{SL/2} \\
0 \\
0
\end{bmatrix},
\]

\[
x_f = \begin{bmatrix}
P_v \\
\dot{m}_v
\end{bmatrix}
\]

The mass flow input \( \dot{m}_d \) to the system was designed as

\[
\dot{m}_d = \hat{\dot{m}} + \frac{SL/2}{RT} \left\{ \frac{dP_d}{dt} - K_f (P_v - P_d) \right\}. \tag{22}
\]

\( \hat{\dot{m}} \) is the estimated value of \( \dot{m} \) obtained from (2) and (3). The proposed method for the proportional flow control valve consists of the components represented by (8), (9), and (22). \( K_f \) is the proportional gain for the valve pressure feedback control and tuned as \( K_f = 3 \times 10^2 \) 1/s. \( \dot{m}_V = \dot{m}_d \) is achieved by the proportional flow control valve and the control method of the valve is described in the Appendix.

The baseline method that did not consider the delay due to the long transmission line used the control command in (22) directly using the reference pressure as the desired pressure \( P_d = P_r \) as explained in Section V-B.

In this experiment with the proportional flow control valve, we used the polytrope numbers \( k_1 = 1.4 \) and \( k_1 = 1.2 \) in (3) for the system with the valve (Festo MPYE-5-1/4-010-B), and re-identified the PAM volume related parameters as \( V_1 = 1.09 \times 10^{-4} \text{m}^3 \) and \( V_2 = -2.37 \times 10^{-4} \text{m}^3 \) respectively. Fig. 18 shows the RMSE result of the pressure tracking task with 7 m tube under the free condition (see also Fig. 5). Our proposed method showed better control performance than the baseline method even with a proportional flow control valve.

**VIII. CONCLUSION**

Pressure transmission through the long tube between the valve and the PAM worsens the control performance of the PAM system. We derived the airflow dynamics using a long transmission line and designed a controller using the inverse dynamics model. The proposed method improved the control performance when the effect of the pressure transmission
was significant; for example, under high frequency sinusoidal reference profile or when using a long tube.

In this study, we focused on improving a single PAM system’s force control performance. Joint actuators with antagonistic PAMs were studied [24], [26], [27], with the force control term of a single PAM present inside the actuator control system. Therefore, it is suggested that improving the control performance of a single PAM can enhance the performance of the antagonistic joint, which can lead to performance improvements in robotic systems with PAM-based joints.

The pressure feedforward controller presented in this study can be applied to other pneumatic actuators [28], by changing the volume model to the relevant models.

APPENDIX

VALVE CONTROLLER SETTINGS

Here we explain the mathematical models of the proportional pressure control valve, the proportional flow control valve and their controller.

The proportional pressure control valve (Norgren VP5010B), is expressed as

\[ u = \frac{1}{c_p} \left( (P_V - P_{room}) + \tau_c \frac{dP_V}{dt} \right) - u_{bias}. \]  

and its controller was designed as [25]

\[ u = \frac{1}{c_p} \left( (P_d - P_{room}) + \tau_c \frac{dP_d}{dt} \right) - u_{bias}. \]

\( \tau_c \) is time constant,\( u_{bias} \) is the voltage bias, and \( c_p = 1 \times 10^{-1} \text{MPa/V} \) is the conversion parameter from the voltage to the pressure. The values identified from the prior experiment are shown in the table III.

TABLE III: Identified parameters of proportional pressure control valve

\[ \tau_c = 6.15 \times 10^{-2} \text{s} \quad u_{bias} = -3.00 \times 10^{-1} \text{V} \]

The model of the proportional flow control valve (Festo MPYE-5-1/4-010-B) was derived based on [29]. The flow generated by the valve is defined as

\[ \dot{m}_V = a_c(u)\phi(P_c, P_V) - a_r(u)\phi(P_V, P_{room}). \]  

(25)

\( a_c(u) \) and \( a_r(u) \) are the aperture areas of the supply and exhaust ports, respectively. \( \phi(P_a, P_b) \) is the thin-plate flow function between the pressure \( P_a \) and \( P_b \) (for more detail see [29] and [30]). The relationship between \( a_c(u) \), \( a_r(u) \), and the voltage input \( u \) is

\[ a_c(u) = L_c + \max \{ (u - U_c)b - L_c \} \]  

(26)

\[ a_r(u) = L_r + \max \{ (U_r - u)b - L_r \}. \]  

(27)

\( L_c \) and \( L_r \) are the minimum aperture areas of the supply and exhaust ports, respectively. \( U_c \) and \( U_r \) are the voltage values when the respective ports are sealed, and \( b \) is the conversion factor from voltage to aperture area. By using the inverse model of (25), (26), and (27), the feedforward controller of the flow control valve is designed as

\[
\begin{align*}
\dot{m}_d & = \frac{1}{bo_r(P_c, P_{room})} (U_c + L_r \phi(P_c, P_V)) \quad (\dot{m}_d \geq 0) \\
\dot{m}_d & = \frac{1}{bo_r(P_c, P_{room})} (U_r + L_c \phi(P_V, P_{room})) \quad (\dot{m}_d < 0).
\end{align*}
\]

(28)

We adopted a first-order low-pass filter with a time constant \( \tau = 0.05 \) to derive \( \dot{m}_d \) otherwise the valves tended to oscillate. The values identified from the prior experiment are shown in the table IV.

TABLE IV: Identified parameters of proportional flow control valve

\[ \begin{array}{c|c|c|c}
\text{Voltage} & \text{LC} & \text{LR} & \text{b} \\
\hline
U_{c} & 5.51 \text{V} & 2.14 \times 10^{-7} \text{m}^2 & 4.76 \times 10^{-3} \text{m}^2/\text{V} \\
U_{r} & 4.65 \text{V} & 3.65 \times 10^{-7} \text{m}^2 & \end{array} \]

Both valves are controlled by the multi-function board, where the board communicates with real-time OS Xenomai through real-time Ethernet as explained [7], [11].

REFERENCES


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