Compensation of Pressure Dependent Disturbance: Poppet Position Control in a Pneumatic Valve

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Abstract—Pneumatic valves are key components for controlling mass flow rates in general industrial applications. However, they have several nonlinearities such as dead zone and airflow force, making precise control of mass flow rates challenging. Since the poppet position mostly determines the mass flow rate of a valve, this study employs a new valve with an internal position sensor. The authors propose a data-driven feedforward control method to precisely control the poppet position at arbitrary pressure differences by estimating air disturbance force including airflow force. The developed approach compensates for the air disturbance force to the poppet position and enables fast movement without overshooting. The performance improvement is experimentally validated in the poppet position tracking experiments.

Index Terms—Pneumatic valve, Mass flow rate control, Poppet position control, Disturbance estimation, Iterative learning control

I. INTRODUCTION

Pneumatic valves play a crucial role in controlling mass flow rates of gases in general industrial automation equipment such as semiconductor manufacturing equipment, industrial robots [1], and pneumatic actuators [2]. In particular, pneumatic actuators offer the advantage of a high thrust-to-weight ratio and low cost, but they exhibit the disadvantage of large nonlinearities. A major challenge for the precise control of pneumatic actuators results from the valve structure [3]. Unfortunately, pneumatic valves are subject to the following nonlinearities:

- 1) Static nonlinearity where the dead zone of input current changes due to temperature and pressure
- 2) Dynamic nonlinearity where the valving element tends to vibrate as the mass flow rate changes very fast

These nonlinearities make accurate control of the mass flow rate difficult to achieve.

As a result, to mitigate these nonlinearities, a lot of previous studies have analyzed the complex structure of a valve. The mass flow rate of a two-port valve is generally modeled by nonlinear equations using the position of poppet or spool, which are movable elements in the valve, and the pressure difference between two ports of a valve [4]. Since the position of poppet or spool is difficult to measure as an electrical signal, most studies assume a linear relationship between the input current and the position [3], [5]–[8]. However, characteristics from the input current to the mover position change depending on various parameters such as temperature and pressure. If the modeling of the current dead zone is inaccurate, significant

nonlinearities exist between input current and actual position [9]. Therefore, to compensate for the static nonlinearities caused by modeling errors of the current dead zone, a new position-controlled valve was employed to enable feedback (FB) control of the position [10]. By introducing a closed-loop control system for the position, the linearity of the valve static characteristics was improved.

In the traditional modeling of mass flow rate, the upstream pressure of a valve is often regarded as constant, neglecting the impact of pressure fluctuations. However, since the response of a pressure regulator is relatively slow, a swift adjustment in mass flow rate leads to an abrupt change and transient oscillations in the outlet pressure of the regulator, which is the upstream pressure of a valve [6], [11]. Pressure oscillations can be suppressed to some extent by installing an accumulator tank between the regulator and the valve, but the pressure fluctuates significantly even within the tank [12]. In the case of a spool valve. Mivaiima et al. [13] demonstrated that employing spool position FB control and a dead time compensator results in accurate control of the spool position dynamics. On the other hand, in the case of the poppet valve, its higher sensitivity to the pressure vibration results in the poppet oscillation with a quick alteration in mass flow rate. To suppress the fluctuation of the poppet position, it is common to reduce the airflow force by limiting a changing speed of mass flow rate, but this approach slows the valve operating speed [9], [14].

Although the static nonlinearity is alleviated by adding poppet position FB control [10], it cannot address the transient vibration of the poppet when the mass flow rate changes quickly. To mitigate the airflow force effect, this study proposes a new type of feedforward (FF) control of the poppet position. The developed approach can instantly raise the opening degree of a valve without the overshoot of the poppet position.

The main contribution of this study is an FF control design of the poppet position to enable rapid movement without transient vibrations at arbitrary pressure differences. Iterative learning control (ILC) can provide high tracking performance for one specific reference trajectory [15], [16]. The challenge in developing ILC for the poppet position is to address the variation of the airflow force associated with pressure differences. Our study estimates the air disturbance force at arbitrary pressure differences by combining ILC with linear interpolation on the initial pressure difference.

Tab. I: List of symbols in a poppet valve.

Symbols	Definition
\dot{m}	mass flow rate of air
K_s	spring force constant
y	poppet displacement of a valve
K_{f}	electromagnetic force constant
i	valve input current
r	radius at the top of the outlet conduit
P_u	pressure upstream of the valve orifice
P_d	pressure downstream of the valve orifice
ΔP	pressure difference between two ports
	$(\Delta P = P_u - P_d)$
M	poppet mass of a valve
D	friction coefficient of a valve
ΔP_{init}	initial pressure difference during experiment
$F_{\rm flow}$	airflow force applied to the poppet
G	nominal model of poppet driving system
$C_{\rm FB}^y$	poppet position feedback controller
$C_{\rm FF}^y$	poppet position feedforward controller
$y_{ m ref}$	poppet position reference
f	feedforward input to a valve
d	input disturbance to a valve

The outline is as follows: In Section II, a classic poppet valve is described. In Section III, a new poppet valve used in this study is introduced. In Section IV, the problem that is considered in this study is formulated. In Section V, the developed approach is presented. In Section VI, the performance improvement with the developed approach is experimentally validated. In Section VII, conclusions are presented.

II. CLASSIC POPPET VALVE

A cross-sectional diagram of a classic poppet valve is shown in Fig.1 (a). The definitions of the symbols used in a poppet valve are given in Tab.I. The amount of airflow passing through the valve is adjusted by driving the poppet, shown in light gray in the diagram. The dynamics of the poppet can be expressed as follows:

$$M\ddot{y} + D\dot{y} + K_s y = K_f i + \Delta P \pi r^2 \tag{1}$$

$$= K_f i + \Delta P_{\text{init}} \pi r^2 + F_{\text{flow}} \qquad (2)$$

The position of the poppet is determined by the spring force $K_s y$, the electromagnetic force $K_f i$, and the air pressure force $\Delta P \pi r^2$ determined by the pressure difference between the two ports. The air disturbance force $\Delta P \pi r^2$ can be divided into static part $\Delta P_{\text{init}} \pi r^2$ and dynamic part F_{flow} . While static force can be easily modeled using initial pressure difference ΔP_{init} , dynamic airflow force cannot be because of nonlinear airflow dynamics. Since the poppet position is determined mechanically by a spring, there is no electrical measurement available for the position of a classic valve.

A classic valve has static nonlinearity where the dead zone of input current changes due to the pressure difference. The input-output characteristic of a classic poppet valve is displayed in Fig.2 (a). The pressure difference greatly impacts the static property, especially the current dead zone.



Fig. 1: Cross-sectional diagram of two types of poppet valves.



Fig. 2: Measurement results of the input-output characteristics of two types of poppet valves [10]. The position-controlled valve exhibits smaller variations in the dead zone.

III. POSITION CONTROLLED POPPET VALVE

In this section, a new position-controlled valve used in this experiment is introduced. This valve allows the poppet position to be measured electronically with a built-in position sensor. A cross-sectional diagram of the position-controlled valve is shown in Fig.1 (b). The dynamics of the poppet can be expressed as follows:

$$M\ddot{y} + D\dot{y} = K_f i - \Delta P \pi r^2 \tag{3}$$

$$= K_f i - \Delta P_{\text{init}} \pi r^2 - F_{\text{flow}}$$
(4)

The poppet is mainly driven by the electromagnetic force $K_f i$ and the air pressure force $\Delta P \pi r^2$. A mechanical FB control system with physical spring force $K_s y$ is removed thanks to the implementation of a poppet position sensing system that allows precise control of the poppet position.

Accurate manipulation of the poppet position in a positioncontrolled valve allows for effective compensation of the static nonlinearity. The input-output characteristic of a positioncontrolled poppet valve is displayed in Fig.2 (b). The static property from the poppet position is less dependent on the pressure difference compared to the property from the input current in a classic valve, especially the varying dead zone.

IV. PROBLEM FORMULATION

Although poppet position FB control improves the static linearity, transient vibrations of the poppet may occur because of a rapid increase in the magnitude of airflow force F_{flow} in (4). In this section, the limitations of poppet position FB control in dynamic characteristics are explained.



upstream pressure drops significantly and oscillates.

Fig. 4: Measurement results of fast-Fig. 3: Measurement results of two types of poppet position FB control response poppet position FB control. with different response speeds. As the mass flow rate increases rapidly, the An increase in the initial pressure difference makes the poppet vibrate.

A. Poppet vibration due to upstream pressure fluctuations

When the moving speed of the poppet position is fast, the poppet position with FB control overshoots due to an increase in the airflow force F_{flow} . Time responses of mass flow rate and upstream pressure are shown in Fig.3. The initial pressure difference of the value ΔP_{init} is set to 0.30 MPa. The red line represents responses with low bandwidth FB control, while the green line represents responses with high bandwidth FB control. When the mass flow rate increases quickly, the upstream pressure drops significantly and oscillates as shown in Fig.3 (b). Then, dynamic airflow force F_{flow} included in poppet driving force shown in (4) grows greatly, and as a result, the mass flow rate vibrates as shown in Fig.3 (a).

B. Variation of external disturbances due to airflow force

Since the response of the FB control system of the poppet position is slow, it cannot address the variation in airflow force F_{flow} , which depends on the initial pressure difference ΔP_{init} . Time responses of the poppet position are shown in Fig.4. The initial pressure differences ΔP_{init} are set to 0.00 MPa, 0.15 MPa, and 0.30 MPa. In particular, $\Delta P_{\text{init}} = 0.00 \text{ MPa}$ signifies that both ports are open, and there is no air pressure force in the valve. From Fig.4, it can be seen that as the initial pressure difference ΔP_{init} increases, the magnitude of dynamic airflow force F_{flow} also increases, causing oscillations in the poppet position response. In this study, air pressure force $\Delta P \pi r^2$ in (3) is treated as an external disturbance force d, as shown in Fig.5. These measurement results demonstrate that the disturbance rejection performance of the FB control system is not enough to suppress airflow force.

V. APPROACH

This section describes the proposed method for compensating dynamic airflow force F_{flow} using an FF controller, which accommodates variations in initial pressure differences ΔP_{init} .

A. Proposed method of poppet position feedforward control

Although accurate modeling of transient airflow force is challenging due to the compressible fluid dynamics, airflow



Fig. 5: Block diagram of a two-degree-of-freedom (2-DOF) poppet position control system.

force for one specific reference y^{ref} is highly reproducible when the initial pressure difference ΔP_{init} is constant. Consequently, this study proposes a control method aimed at constructing a lookup table for the pressure-dependent disturbance force F_{flow} using previous experimental data. This approach compensates for the disturbance at arbitrary pressure differences by incorporating interpolation for the initial pressure difference ΔP_{init} . The process of the introduced FF disturbance compensation can be explained as follows:

- 1) Experiments on tracking control for the poppet position are conducted for multiple initial pressure differences, and the FF input that minimizes the tracking error is obtained for each pressure difference ΔP_{init} .
- 2) A lookup table is created from the FF inputs obtained for each initial pressure difference ΔP_{init} .
- 3) The FF input that can compensate for the pressuredependent disturbance F_{flow} at the assumed initial pressure difference is estimated by interpolating the lookup table on the pressure difference ΔP_{init} .

As the first step, this research employs ILC to effectively obtain the FF input that minimizes the tracking error. Additionally, linear interpolation is utilized to interpolate the table of FF inputs due to its low implementation cost.

B. Variation of feedforward inputs calculated by ILC depending on the initial pressure difference

ILC is a nonparametric method of obtaining a highperformance FF input under the same experimental conditions by sequentially updating the control input after each iteration



Fig. 6: Feedforward inputs obtained through ILC for the seven types of initial pressure differences. They nonlinearly depend on the initial pressure difference.

[17]. After each task j, the ILC algorithm generates the FF input f_{j+1} to be used in the next iteration. According to the update law of (5), f_{j+1} is determined by adding a filtered version of the measured error signal e_j to the past FF input f_j . Here, L is the learning filter and Q is the robustness filter.

$$f_{j+1} = Q\left(f_j + Le_j\right) \tag{5}$$

In this paper, the initial pressure differences ΔP_{init} stored in the lookup table are configured with seven types: 0.00, 0.05, 0.10, 0.15, 0.20, 0.25, and 0.30 MPa. The FF inputs obtained through ILC for these pressure differences are illustrated in Fig.6. These results illustrate the nonlinear variation of airflow force F_{flow} at different initial pressure differences.

C. Effect of linear interpolation on feedforward inputs for the initial pressure difference

The approach in this study ensures flexibility of FF control by combining ILC with linear interpolation with initial pressure difference ΔP_{init} . First, accurate FF inputs $f_{\Delta P}(P_1)$ and $f_{\Delta P}(P_2)$ that will be used for linear interpolation are acquired by ILC. They correspond to the initial pressure difference of $P_1 = 0.250 \text{ MPa}$ and $P_2 = 0.300 \text{ MPa}$, respectively. Next, we consider the situation with the initial pressure difference of $P_3 = (P_1 + P_2)/2 = 0.275 \text{ MPa}$, which is not included in the lookup table. In this case, the final step calculates the FF input $f_{\Delta P}(P_3)$ using linear interpolation as follows:

$$f_{\Delta P}(P_3) = \frac{f_{\Delta P}(P_1) + f_{\Delta P}(P_2)}{2}$$
(6)

Time responses of the poppet position when the initial pressure differences ΔP_{init} are set to P_1 and P_2 are shown in Fig.7 (a) and (b), respectively. The red line represents the responses with only FB control f = 0, while the blue line depicts the responses with ILC $f = f_{\Delta P}(P_1)$ and $f = f_{\Delta P}(P_2)$. The results of the root mean square error (RMSE) are listed in Tab.II. Compared to the results of only FB control, ILC mitigates the RMSE of the poppet position by 85%.

Next, to evaluate the flexibility of FF control, time responses when the initial pressure difference ΔP_{init} is set to P_3 are

Tab. II: RMSE results of FF control with pressure dependency.

Initial pressure difference	FF controller	FF input	RMSE
$P_1 = 0.250 \mathrm{MPa}$	without FF	0	$0.0546\mathrm{mm}$
$P_1 = 0.250 \mathrm{MPa}$	lookup table	$f_{\Delta P}(P_1)$	$0.0080\mathrm{mm}$
$P_2 = 0.300 \mathrm{MPa}$	without FF	0	$0.1048\mathrm{mm}$
$P_2 = 0.300 \mathrm{MPa}$	lookup table	$f_{\Delta P}(P_2)$	$0.0072\mathrm{mm}$
$P_3 = 0.275 \mathrm{MPa}$	lookup table	$f_{\Delta P}(P_1)$	$0.0075\mathrm{mm}$
$P_3 = 0.275 \mathrm{MPa}$	lookup table	$f_{\Delta P}(P_2)$	$0.0157\mathrm{mm}$
$P_3 = 0.275 \mathrm{MPa}$	interpolated	$f_{\Delta P}(P_3)$	$0.0058\mathrm{mm}$

shown in Fig.7 (c). The red and green lines in the figures depict the responses with raw FF inputs from the table, specifically $f = f_{\Delta P}(P_1)$ and $f = f_{\Delta P}(P_2)$. In contrast, the blue line illustrates the response with the FF input obtained through interpolation, denoted as $f = f_{\Delta P}(P_3)$. Tab.II compares the RMSE results for raw FF inputs from the lookup table with the interpolated FF input. Compared to the results of raw FF inputs from the lookup table, interpolation reduces the RMSE by 23%. These results demonstrate that linear interpolation can estimate the airflow disturbance force F_{flow} at any pressure differences not stored in the lookup table.

VI. EXPERIMENTAL VALIDATION

A. Experimental setup

The experimental setup for mass flow rate control is shown in Fig.8. A buffer tank is connected to the upstream port of a valve and its pressure is determined by a pressure regulator. The upstream pressure of a valve is measured accurately by a pressure sensor. The volume of the buffer tank is 38 L. With the downstream port of a valve open to ambient air, the downstream pressure is assumed to be equal to atmospheric pressure. The sampling frequency of a controller is 16 kHz.

B. Plant model of poppet position control system

The poppet position control system in this paper is shown in Fig.5, consisting of a 2-DOF control system designed for a nominal plant G(s). The control input of the plant is the valve input current, and the output is the poppet position. The dynamic characteristics of the current control loop are sufficiently higher than those of the motion of the poppet. As described previously, the air pressure force $\Delta P \pi r^2$ in (1) is treated as an external disturbance force d. The nominal plant G(s) is identified as a combination of the second-order system and dead time, with a poppet mass $M = 2.7 \times 10^{-3}$, viscous friction coefficient D = 1.0, and dead time $\tau = 8.75 \times 10^{-4}$.

$$G(s) = \frac{1}{s} \frac{1}{Ms + D} e^{-\tau s}$$
(7)

C. Design of feedback controller and learning controller

The FB controller C_{FB}^y is designed as a proportionalintegral-derivative (PID) controller, with $K_p = 442$, $K_i = 1.00 \times 10^4$, $K_d = 0.819$, and $\tau_d = 6.60 \times 10^{-4}$. The bandwidth of the FB control system defined in [18] is 42 Hz.

$$C_{\rm FB}^y(s) = K_p + \frac{K_i}{s} + \frac{K_d s}{\tau_d s + 1}$$
 (8)



Fig. 7: Experimental results of the proposed poppet position control on various initial pressure differences. Tracking performance is improved by obtaining control inputs through ILC and interpolating them on initial pressure differences.



Fig. 8: Experimental setup for valve precise control. (blue: airflow, green: signals)

The implementation of ILC requires the non-causal filters Q(s), L(s) that appear in (5). The filter Q(s) is designed as a non-causal zero-phase filter, with its pole $\omega_o = 2\pi \times 300 \text{ rad/s}$.

$$Q(s) = \left|\frac{1}{1+s/\omega_o}\right|^2 \tag{9}$$

The filter L(s) is designed as follows:

$$L(s) = \left(\frac{G(s)}{1 + C_{\rm FB}^y(s)G(s)}\right)^{-1}$$
(10)

To reduce the stick-slip friction problem of poppet position control, a certain level of chatter with a frequency of 400 Hz is added to the input current as a dither signal [19].

D. Experimental results of poppet position control

To validate the effectiveness of the proposed FF control, the performance of poppet position tracking is compared with that of the conventional FB control method. The same FB controller described in (8) and filters in (9) and (10) are used across all the experimental conditions detailed below. The FF controller $C_{\text{FF}}^{y}(s)$ is designed in the following cases:

Case1 FF input is zero and only FB control is performed.

- Case2 ILC is applied with two ports open, and the resulting FF input with pressure difference 0.000 MPa is used under any initial pressure differences ΔP_{init} .
- Case3 ILC is applied on all 7 types of initial pressure differences, and the lookup table is integrated with

Tab. III: RMSE results for 3 types of position FF controllers.

Initial pressure difference	FF controller	Situation	RMSE
$0.000\mathrm{MPa}$	without FF	Case1	$0.0263\mathrm{mm}$
$0.000\mathrm{MPa}$	ILC in no air	Case2	$0.0040\mathrm{mm}$
$0.275\mathrm{MPa}$	without FF	Case1	$0.0480\mathrm{mm}$
$0.275\mathrm{MPa}$	ILC in no air	Case2	$0.0291\mathrm{mm}$
$0.275\mathrm{MPa}$	interpolated	Case3	$0.0058\mathrm{mm}$

interpolation for the pressure differences ΔP_{init} to produce the FF input at any pressure differences.

Case1 is the conventional method explained in Section IV, and Case3 is the proposed method presented in Section V. Although Case2 implements a simple FF controller, in contrast to Case3, it does not consider the variation of airflow force F_{flow} affected by the initial pressure difference ΔP_{init} .

First, time responses of the poppet position with no air pressure force $\Delta P \pi r^2$ at the initial pressure difference $\Delta P_{\text{init}} = 0.000 \text{ MPa}$ are shown in Fig.9 (a). Tab.III shows that the RMSE of the poppet position is reduced by 85% for the 2-DOF control compared with only FB control.

Next, time responses of the poppet position and mass flow rate with the initial pressure difference $\Delta P_{\text{init}} = 0.275 \text{ MPa}$ are shown in Fig.9 (b) and (c) respectively. In Case3, where airflow disturbance force F_{flow} is taken into account by the FF controller, there is a reduction in the overshoot. Tab.III indicates that the proposed FF control achieves 80% smaller RMSE than the simple FF control that does not consider pres-



Fig. 9: Experimental results of the proposed poppet position control in the absence and presence of air pressure force. Tracking performance is improved by considering the influence of airflow disturbance force subject to the initial pressure difference.

sure dependency. Additionally, Fig.9 (c) suggests that accurate FF control of the poppet position is very effective for the precise mass flow rate control of a valve.

VII. CONCLUSION

The mass flow rate through a valve primarily depends on the poppet position, leading this study to introduce a novel valve equipped with an internal position sensor. Although it enables feedback control of the poppet position, achieving accurate tracking control of the poppet position is challenging, mainly due to dynamic variations in air disturbance force. To attenuate this disturbance before it affects the control system, the authors propose to employ a data-driven FF control strategy for the poppet position. The application of ILC allows the computation of precise control inputs for various initial pressure differences. Furthermore, linear interpolation is employed to accommodate variations in initial pressure differences. Experimental results reveal the effectiveness of the proposed FF control in compensating for pressure-dependent disturbances caused by airflow force, ensuring no overshooting in the poppet position.

Ongoing work is also aimed at developing an FF controller to accommodate changes in the target trajectory of the poppet position, which could lead to achieving precise tracking control of the mass flow rate under any pressure differences.

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