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Proposal of Anti-Windup Method of Twin Drive Mass-Flow-Rate Control for Pneumatic Driving System

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Manufacturing equipment requires high-speed and high-precision tracking position control with a long stroke. This paper aims to utilize pneumatic cylinders for such equipment as they have many advantages. However, the pneumatic system has the challenges for achieving high accuracy positioning. One cause is that a valve has a dead zone with a variation. The dead zone, a nonlinearity, degrades the control performances of the pressure and position unless it is compensated in the mass-flow-rate control system. As a compensation method in the mass-flow-rate control system, we have proposed the mass-flow-rate twin drive system. In this method, the difference mode controls the mass-flow-rate difference that decides the pressure. On the other hand, the sum mode controls the offset of each mass flow rate. This method can avoid the effect of the dead zone by keeping the offset of each mass flow rate high. However, a high sum mode reference may cause saturation. Thus, this paper proposes an anti-windup method for a twin drive system with coordinate transformation. The originality is to introduce an anti-windup structure only in the sum mode. The experimental results show the effectiveness of the proposed method.

Keywords: pneumatic cylinder, valve, mass flow rate control, dead zone compensation, twin drive, anti-windup

1. Introduction

Manufacturing equipment such as machine tools, wafer scanners, and flat-panel scanners requires high-precision and high-speed tracking position control with a long stroke [1, 2]. As linear motors can achieve high-precision and high-speed positioning, they are implemented not only in the fine stage but also in the coarse stage. However, a massive heat generation and a lack of thrust may limit the performance [1, 3–5]. We aim to replace a linear motor with a pneumatic cylinder because it has several advantages such as high power-to-weight ratio, low-cost, and low heat generation [5–9].

A pneumatically actuated stage is shown in Fig. 1. The schematic of pneumatic cylinder is illustrated in Fig. 2. A valve-input voltage decides the mass flow rate of air passing through a valve. The mass flow rate difference between supplied mass flow rate and the exhausted mass flow rate decides the pressure. And the pressure difference between the two chambers decides the stage position. Thus, a pneumatic driving system has a position, pressure, mass flow rate, and voltage loops as shown in Fig. 3.

However, the pneumatic cylinders with cm working range cannot achieve the tracking position control with nm or um accuracy because of many challenges [5–7, 10–13]. One cause is a valve dead zone [6, 12]. As the valve dead zone degrades the control performance of the pressure loop and position loop, the dead zone compensation in the mass flow rate control system is required.

The conventional approach of dead zone compensation is

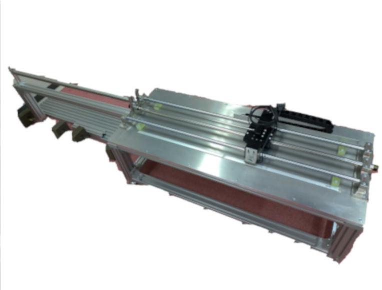

Figure 1: Pneumatically actuated stage.

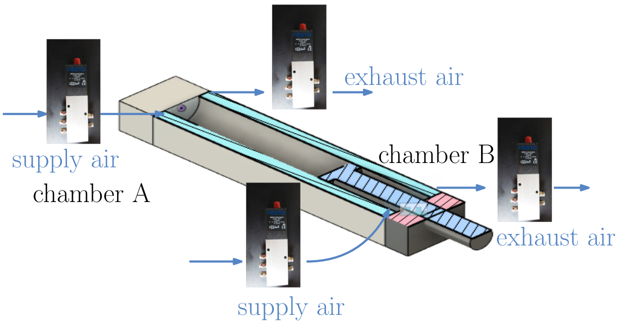

Figure 2: Schematic of pneumatic cylinder

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to assert an inverse model of input-output characteristics of a valve [6, 7, 10, 12, 14]. However, this method cannot address the variation of a valve.

To address the variation of a valve, the mass-flow-rate FB (feedback) control system with a fast-response flow meter has been proposed [15, 16]. There are two approaches. One approach is to address the nonlinearity only with a feedback controller. However, the mass-flow-rate feedback bandwidth would be low because the plant has nonlinearity. Another approach is to address the nonlinearity with an inverse model of a valve and a feedback controller. As the plant is composed of the inverse model and a valve, the plant ideally has the gain of 1 and the dead time. However, as the dead zone fluctuates, the nonlinearity remains in the plant. If the mass flow rate reference is small, the modeling error causes the nonlinearity and the mass flow rate cannot follow the reference.

Thus, we have proposed an advanced dead zone compensation method by the twin drive system [18]. The twin drive system requires two valves. The twin drive system converts the mass flow rate of two valves into sum mode and difference mode and controls the mass flow rate in the modes. While the difference mode controls the mass-flow-rate difference that decides the pressure, the sum mode controls the offset of each mass flow rate. The mass-flow-rate difference can follow the reference even though the reference is a small value for the following reasons [18].

- The dead zone of two valves cancel each other out in the difference mode.
- The sum mode keeps each mass flow rate enough high to avoid the effect of the dead zone.

The twin drive system indeed has the advantages, but under some conditions, it cannot achieve a precise following. Considering an actual situation of positioning with a pneumatic cylinder, the difference-mode reference is decided by a pressure controller. In contrast, the sum-mode reference is assigned by an experimenter. Figure.4 shows where to set a sum-mode reference. The sum-mode reference should be assigned based on three conditions in such a way that the mass-flow-rate difference follows the reference.

(1) large enough to avoid the effect of the dead zone
(2) $\dot{m}_{sum,ref} > \frac{1}{2}|\dot{m}_{dif,ref}|$
(3) avoid the effect of saturation (small value or anti-windup system in the sum mode)

Firstly, the sum-mode reference should be large in such a way that each mass flow rate is large enough to avoid the effect of the dead zone. Secondly, the sum-mode reference should be larger than the difference-mode reference. If the sum-mode reference is small, each mass flow rate is small and the mass-flow-rate difference cannot follow the reference. Therefore, the sum-mode reference should be large. However, as a large sum-mode reference gives large valve input voltages, it may cause a saturation. To avoid the effect of the saturation with a large sum-mode reference, the anti-windup (AWU) system is required.

The difficulty of the anti-windup system for the twin drive system is that the limitation on the control output should be calculated by coordinate transformation because a control output is not directly inputted to a plant. This paper proposes an anti-windup method for a twin drive system with coordinate transformation. Furthermore, this paper proposes an anti-windup structure only in the sum mode. This is because although the mass flow rate difference decides the pressure of a chamber, the mass flow rate sum decides whether a valve is influenced by the dead zone. Experimental results show the effectiveness of the proposed method.

Table 1: List of symbols

Symbols	Definition
$v_{ref,sup}$	input-voltage reference to a supply valve
$v_{ref,exh}$	input-voltage reference to an exhaust valve
v_{sup}	input voltage to a supply valve
v_{exh}	input voltage to an exhaust valve
$v_{min,sup}$	minimum voltage of supply valve (5.0 V)
$v_{min,exh}$	minimum voltage of exhaust valve (5.0 V)
$v_{max,sup}$	maximum voltage of supply valve (7.5 V)
$v_{max,exh}$	maximum voltage of exhaust valve (7.5 V)
$\dot{m}_{sup}$	mass flow rate of a supply valve
$\dot{m}_{exh}$	mass flow rate of an exhaust valve
v_{dif}	control output voltage of difference mode
v_{sum}	control output voltage of sum mode
$\dot{m}_{sum,ref}$	mass-flow-rate reference of sum mode
$\dot{m}_{dif,ref}$	mass-flow-rate reference of difference mode
$\dot{m}_{sum}$	mass flow rate of sum mode
$\dot{m}_{dif}$	mass flow rate of difference mode
$C^{\dot{m}}_{fb,sum}$	feedback controller of sum mode
$C^{\dot{m}}_{fb,dif}$	feedback controller of difference mode
$C^{\dot{m}}_{fb1,sum}$	part of anti-windup feedback controller of sum mode
$C^{\dot{m}}_{fb2,sum}$	part of anti-windup feedback controller of sum mode

2. Experimental setup

Experimental setup is shown in Fig. 5. Some valves like two-ports valves require a supply valve and an exhaust valve to control the pressure of a tank. This is because the air passes through valves one-way direction. A valve (FESTO,MPYE5-1/8-LF-010-B) is a 5-ports valve but we use it like a 2-ports valve. A supply valve passes air from a pressure regulator to a tank and an exhaust valve passes air from a tank to ambient air. Flowmeters (Keyence, FD-A100,FD-V40A) measure the mass flow rate of air passing through valves. The difference in the mass flow rate decides the pressure in a tank.

3. Conventional method: without anti-windup

The symbols are listed in Table 1.

The block diagram of the mass flow rate feedback system based on a twin drive system is shown in Fig. 6. The twin drive system controls the sum mass flow rate and difference mass flow rate instead of the mass flow rate of each valve. As expressed as 1, the twin drive system converts the mass flow rate of each valve into the mass flow rate of mode with Hadamard matrix [19]. And the coordinate transformation block in Fig. 6 converts v_{dif} and v_{sum} to $v_{ref,sup}$ and $v_{ref,exh}$ based on (2).

$$\begin{pmatrix} \dot{m}_{sum} \\ \dot{m}_{dif} \end{pmatrix} = \frac{1}{2} \begin{pmatrix} 1 & 1 \\ 1 & -1 \end{pmatrix} \begin{pmatrix} \dot{m}_{sup} \\ \dot{m}_{exh} \end{pmatrix} \tag{1}$$

$$\begin{aligned} v_{ref,sup} &= v_{sum} + v_{dif} \\ v_{ref,exg} &= v_{sum} - v_{dif} \end{aligned} \tag{2}$$

As the actual valve input voltages v_{sup} and v_{exh} are limited by (3)(4), the control system without anti-windup mechanism

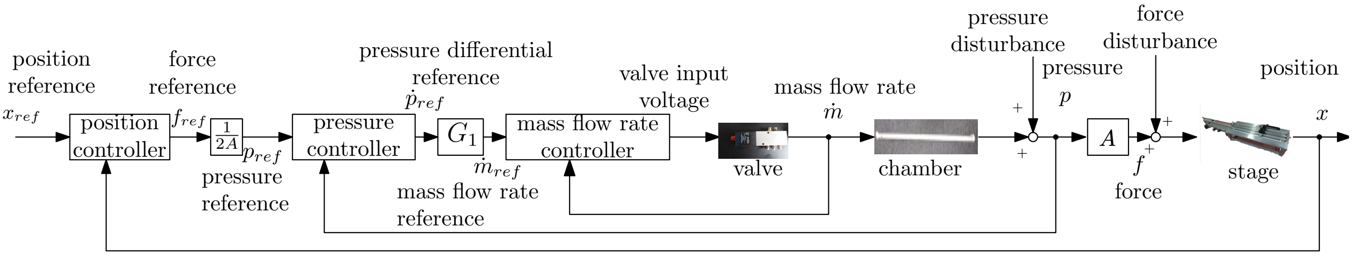


Figure 3: Block diagram of a pneumatic driving system. (G_1: $\frac{p\dot{V}+\dot{p}V}{RT} \times 22.4 \times 60$, A: piston effective area of a cylinder, x: position, f: force, p: pressure, $\dot{p}$: pressure differential, $\dot{m}$: mass flow rate)

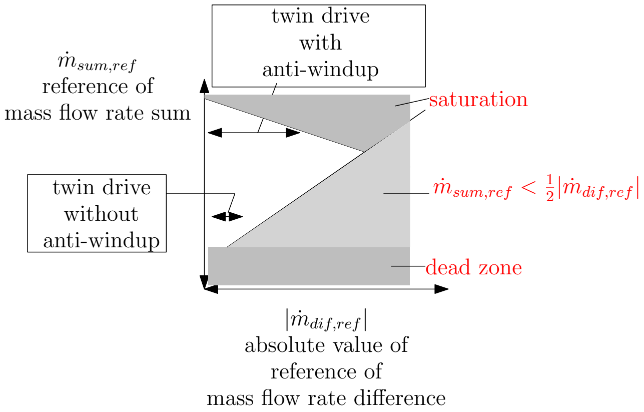


Figure 4: Conditions on a sum-mode reference that the difference mode value follows the reference.

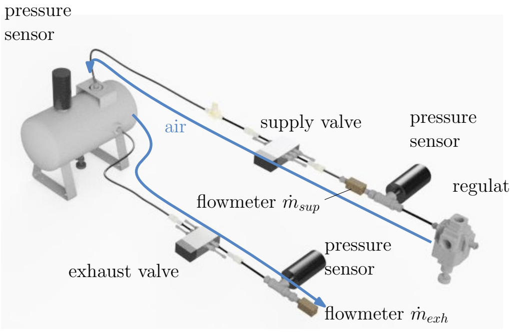


Figure 5: Hardware setup of a mass flow rate control system with two valves.

may result in a saturation.

$$v_{sup} = \begin{cases} v_{min,sup} & (v_{min,sup} \leq v_{ref,sup}) \\ v_{ref,sup} & (v_{min,sup} \leq v_{ref,sup} \leq v_{max,sup}) \\ v_{max,sup} & (v_{ref,sup} \leq v_{max,sup}) \end{cases} \tag{3}$$

$$v_{exh} = \begin{cases} v_{min,exh} & (v_{min,exh} \leq v_{ref,exh}) \\ v_{ref,exh} & (v_{min,exh} \leq v_{ref,exh} \leq v_{max,exh}) \\ v_{max,exh} & (v_{ref,exh} \leq v_{max,exh}) \end{cases} \tag{4}$$

In the case of Conventional method, the controllers do not have the anti-windup mechanism. The block diagram of the sum mode controller is shown in Fig. 7. The controller of the sum mode $C_{fb,sum}$ can be implemented with (5).

$$C^{\dot{m}}_{fb,sum} = \frac{b_0 + b_1 z^{-1} + b_2 z^{-2}}{1 + a_1 z^{-1} + a_2 z^{-2}} \tag{5}$$

4. Proposed method : twin drive system with anti-windup

The block diagram of a sum-mode controller with anti-windup is shown in Fig. 8 [20]. The anti-windup controller is comprised of $C^{\dot{m}}_{fb1,sum}$, $C^{\dot{m}}_{fb2,sum}$, and AWU block. $C^{\dot{m}}_{fb1,sum}$ and $C^{\dot{m}}_{fb2,sum}$ are expressed as (6). Here, b_0, b_1, b_2, a_1, a_2 are constant values in (5). AWU block limit v_{sum} or v_{dif} so that v_{sum} or v_{dif} meets (7).

$$\begin{aligned} C^{\dot{m}}_{fb1,sum} &= b_0 \\ C^{\dot{m}}_{fb2,sum} &= \frac{C^{\dot{m}}_{fb1,sum}}{C^{\dot{m}}_{fb,sum}} - 1 \\ &= \frac{(a_1 - b_1/b_0)z^{-1} + (a_2 - b_2/b_0)z^{-2}}{1 + b_1/b_0 z^{-1} + b_2/b_0 z^{-2}} \end{aligned} \tag{6}$$

In the case of a twin drive system, a control output is not directly inputted to a plant. Therefore, we limit v_{sum} and v_{dif} so that v_{sup} and v_{exh} meet the voltage range. Figure 9(a) exhibits the area that the valve-input voltages meet the voltage range. Here, $v_{min,sup} = 5,\ v_{min,exh} = 5,\ v_{max,sup} = 7.5,\ v_{max,exh} = 7.5$. In contrast, Fig.9(b) indicates the area that the control outputs meet (7). Equation 7 is obtained by (2), (3), and (4). We use the previous v_{dif} or v_{sum} to calculate the limitation on v_{sum} or v_{dif}, respectively.

$$\begin{aligned} v_{min,sup} \leq v_{sum} + v_{dif} \leq v_{max,sup} \\ v_{min,exh} \leq v_{sum} - v_{dif} \leq v_{max,exh} \end{aligned} \tag{7}$$

4.1 Proposed method 1: anti-windup in sum mode and difference mode

Proposed method 1 has an anti-windup system in both sum mode and difference mode. We will show that the Proposed method 2 is superior to Proposed method 1 in the following ability of the difference mode.

4.2 Proposed method 2: anti-windup in the only sum mode

Proposed method 2 has an anti-windup system in only sum mode. Each valve cannot output a negative value of mass flow rate in the case of two-ports valves. Thus, valves cannot make a large mass flow rate difference with a small mass flow rate sum. However, such a limitation on the difference mode can be solved with a large mass flow rate sum. In contrast, the sum mode has a limitation of saturation. Since the saturation deteriorates the following ability of the difference mode, the anti-windup method of the sum mode is required.

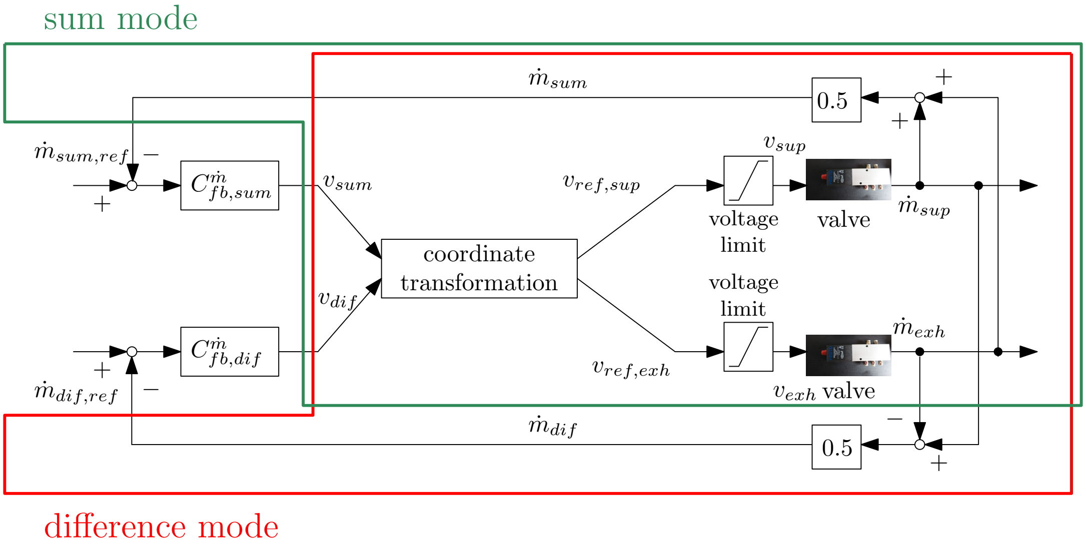


Figure 6: Block diagram of the twin drive system.

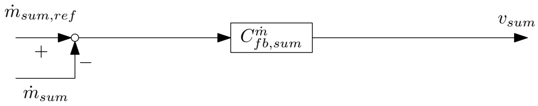


Figure 7: Block diagram of the sum mode controller (without anti-windup).

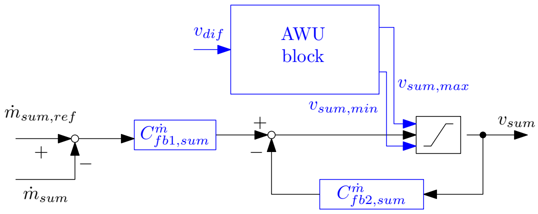


Figure 8: Block diagram of the anti-windup sum mode controller.

5. Experimental results

5.1 Input-output characteristics of a valve Figure 10 demonstrates the steady-state input-output characteristics of a valve. The input signal is the voltage and the output signal is the mass flow rate value. Δp stands for the pressure difference between 2 ports of a valve. As seen in Fig. 10, there is a dead zone from 5.2 V to 5.7 V approximately. As the valve is a spool-type valve, the input-output characteristics have a pressure dependency but the dead zone does not have a large pressure dependency. Thus, the dead zones of two valves cancel out in the difference mode.

5.2 Mass flow rate control results The conditions of the experiment are shown below.

- limitation of input voltage : 5 - 7.5 V
- sum-mode reference : 60 L/min
- difference-mode reference : 5 L/min 1 Hz sine wave
- pressure of a regulator: 0.15 MPa
- pressure of ambient air: 0 MPa

5.2.1 Conventional method The feedback controller of the mass flow rate is designed by pole placement. The closed-loop poles are set at 50 Hz. The mass-flow-rate bandwidth should be high because the mass-flow-rate control system is an inner control system in the pneumatic driving system. However, the delay makes the system unstable if the closed-loop poles are set at a high frequency.

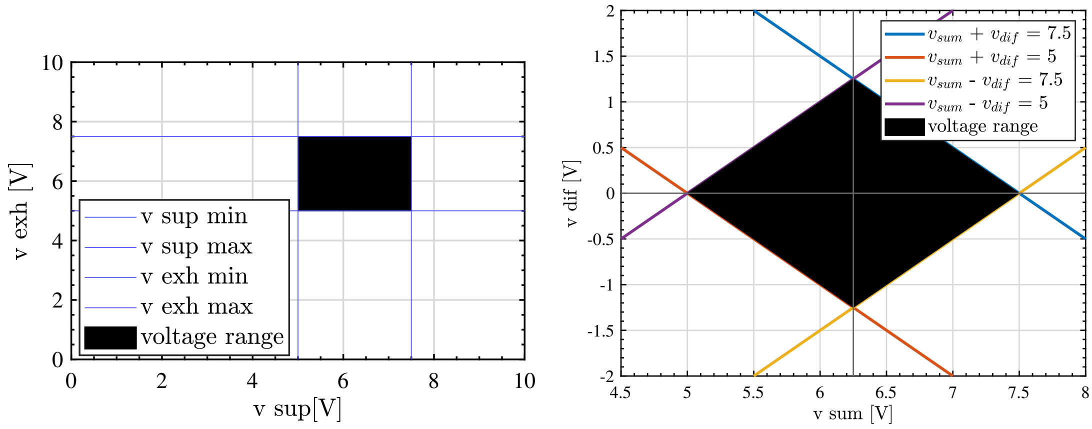
Experimental results of the conventional method are shown in Fig. 11. As the mass-flow-rate difference decides the pressure in a tank or a chamber, the mass-flow-rate difference should follow the reference. However, the mass-flow-rate difference in Fig. 11(a) does not follow the reference because of the saturation. On the other hand, the mass-flow-rate sum should be a large value but it does not have to follow the reference. Therefore, although the mass flow rate sum in Fig. 11(b) does not follow the reference, it is not a problem. Figure 11(c) displays that the reference of valve input voltage exceeds the limitation value.

5.2.2 Proposed method 1 Experimental results of Proposed method 1 (anti-windup in sum mode and difference mode) are plotted in Fig. 12. As illustrated in Fig. 12(a) and Fig. 12(b), neither the mass-flow-rate difference nor the mass-flow-rate sum follows the reference. Because of the anti-windup mechanism, the reference of the valve input voltage illustrated in Fig. 12(c) is limited by 7.5 V.

5.2.3 Proposed method 2 The experimental results of the Proposed method 2 (anti-windup in only sum mode) are shown in Fig. 13. Figure 13(a) presents that the mass-flow-rate difference follows the reference. This is because the anti-windup mechanism does not affect the difference mode. The mass-flow-rate difference follow the reference because the mass-flow-rate difference change the pressure in the tank. On the other hand, as plotted in Fig. 13(b), the mass-flow-rate sum does not follow the reference but it keeps a large value. Figure 13(c) displays that the reference of the valve input voltage is limited by 7.5 V. In short, Proposed method 2 can address the saturation and achieve the following of the difference mode.

6. Conclusion

The twin drive system is effective to compensate for the dead zone. To avoid the effect of the dead zone, the sum-mode reference must be large. However, a large sum mode may cause the windup. Thus, this paper proposes an anti-



(a) v_{sup}, v_{exh}

(b) v_{sum}, v_{dif}

Figure 9: Area that the valve input voltages/control outputs meet the limitations.

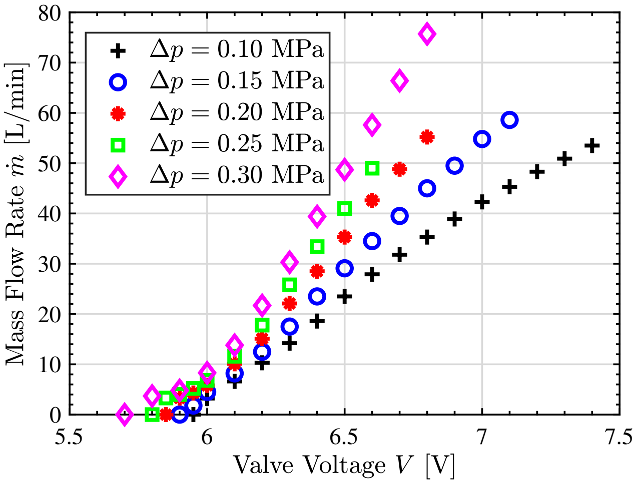


Figure 10: Steady-state input-output characteristics of a valve.

windup method for a twin drive system with coordinate transformation. The originality of the proposed method (Proposed method 2) is to introduce an anti-windup structure only in the sum mode. The experimental results show the effectiveness of the proposed method.

References

(1) T. Oomen, "Advanced Motion Control for Precision Mechatronics: Control, Identification, and Learning of Complex Systems", *IEEJ Journal of Industry Applications*, vol. 7, no. 2, pp. 127–140, 2018.

(2) Nikon Corporation, "Fx-103sh/103s", 2018.

(3) K. Sato, "High-precision and high-speed positioning of 100 G linear synchronous motor", *Precision Engineering*, vol. 39, pp. 31–37, 2015.

(4) R. H. Schmidt, "Ultra-precision engineering in lithographic exposure equipment for the semiconductor industry", *Philosophical Transactions of the Royal Society A: Mathematical, Physical and Engineering Sciences*, vol. 370, pp. 3950–3972, 2012.

(5) W. Ohnishi, H. Fujimoto, P.-H. Yang, P.-W. Chang, B. Yuan, K. Sakata, and A. Hara, "Acoustic Wave Equation Based Modeling and Collocated Side Vibration Cancellation for Pneumatic Cylinder", *IEEJ Journal of Industry Applications*, vol. 7, no. 2, pp. 109–116, 2018.

(6) Z. Rao and G. M. Bone, "Nonlinear modeling and control of servo pneumatic actuators", *IEEE Transactions on Control Systems Technology*, vol. 16, no. 3, pp. 562–569, 2008.

(7) E. Richer and Y. Hurmuzlu, "A High Performance Pneumatic Force Actuator System Part 1 -Nonlinear Mathematical Model", *ASME Journal of Dynamic Systems Measurement and Control*, vol. 122, no. 3, pp. 416–425, 2000.

(8) D. Saravanakumar, B. Mohan, and T. Muthuramalingam, "A review on recent research trends in servo pneumatic positioning systems", *Precision Engineering*, vol. 49, pp. 481–492, 2017.

(9) P. Beater, "Pneumatic drives:system design, modelling and control", Springer, 2007.

(10) H. K. Lee, G. S. Choi, and G. H. Choi, "A study on tracking position control of pneumatic actuators", *Mechatronics*, vol. 12, no. 6, pp. 813–831, 2002.

(11) W. Ohnishi, H. Fujimoto, K. Sakata, A. Hara, and K. Saiki, "Trajectory tracking control for pneumatic actuated scan stage with time delay compensation", *IECON 2016 - 42nd Annual Conference of the IEEE Industrial Electronics Society*, pp. 5125–5130, 2016.

(12) A. C. Valdiero, C. S. Ritter, C. F. Rios, and M. Rafikov, "Nonlinear Mathematical Modeling in Pneumatic Servo Position Applications", *Mathematical Problems in Engineering*, vol. 2011, pp. 1–16, 2011.

(13) T. Miyajima, T. Fujita, K. Sakaki, K. Kawashima, and T. Kagawa, "Development of a digital control system for high-performance pneumatic servo valve", *Precision Engineering*, vol. 31, no. 2, pp. 156–161, 2007.

(14) P. Yang, Y. Choi, P. Chang, K. Sakata, and S. Lee, "Control system for controlling a fluid actuator", United States Patent, no. WO/2017/210450, 2017.

(15) Tomonori Kato, Michio Yanagisawa, Tatsuya Funaki, Kenji Kawashima, and Toshiharu Kagawa, "Pressure Control of Ultra-Precision Air-Spring Type Isolation Table", *Proceedings of JSPE Semestrial Meeting*, vol. 2004S, pp. 279-279, 2004 (in Japanese).

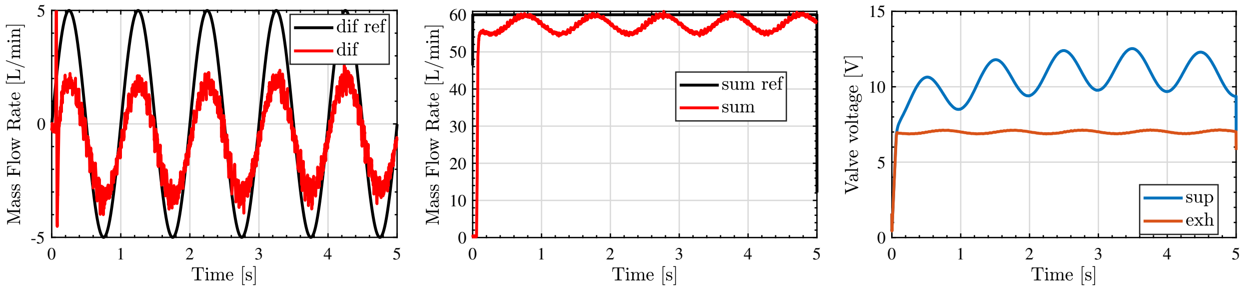
(16) Y. Shirato, W. Ohnishi, and T. Koseki, "Two-Degree-of-Freedom Flow Rate Control for Pneumatic Valves using Fast Response Flow Meter", *IEEJ Journal of Industry Applications*, vol. advpub, 2020.

(17) Y. Shirato, W. Ohnishi, H. Fujimoto, and Y. Hori, "Differential 2-DOF control of mass flow rate for deadzone compensation in pneumatic driving system", *IEEJ/IAS (D) Online Technical Meetings 2020 (DOTeM 2020)*, no. IIC-20-014,MEC-20-007, pp. 37–42, 2020 (in Japanese).

(18) N. Hayashida, T. Yakoh, T. Murakami, and K. Ohnishi, "Friction compensation in twin drive system", *6th International Workshop on Advanced Motion Control. Proceedings (Cat. No.00TH8494)*, pp. 187–192, 2000.

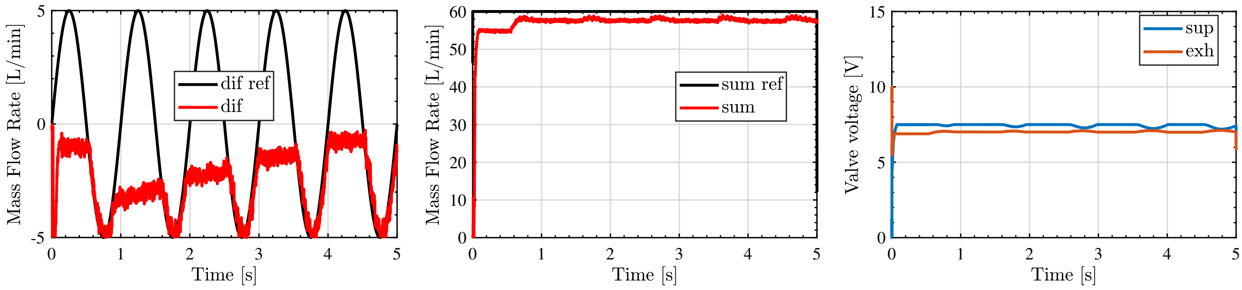
(19) S. Katsura and K. Ohnishi, "Quarry of Modal Information from Environment for Advanced Motion Control", *IEEJ Transactions on Industry Applications*, vol. 126, no. 4, pp. 372–378, 2006.

(20) S. Wakui, S. Hashimoto, H. Takanashi, and Y. Nakamura, "Fundamentals of Control Engineering Available to Industry", CORONA PUBLISHING CO., 2012 (in Japanese).



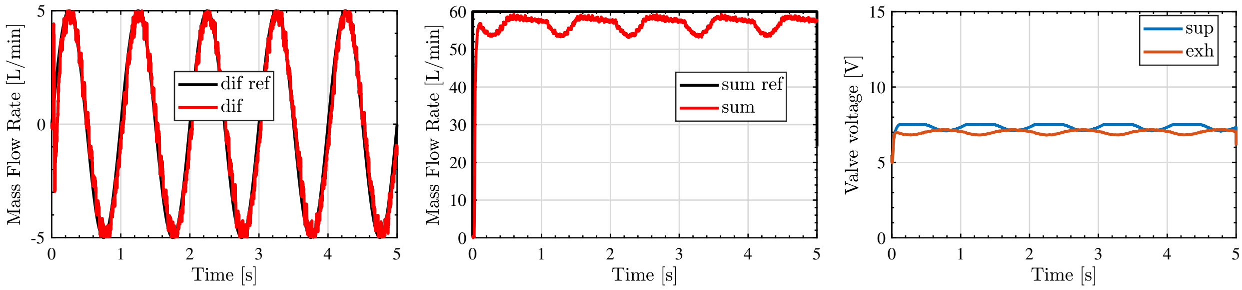
(a) $\dot{m}_{dif,ref}$ and $\dot{m}_{dif}$ (b) $\dot{m}_{sum,ref}$ and $\dot{m}_{sum}$ (c) v_{sup} and v_{exh}

Figure 11: Experimental results of the Conventional method (no anti-windup).



(a) $\dot{m}_{dif,ref}$ and $\dot{m}_{dif}$ (b) $\dot{m}_{sum,ref}$ and $\dot{m}_{sum}$ (c) v_{sup} and v_{exh}

Figure 12: Experimental results of Proposed method 1 (anti-windup in sum mode and difference mode).



(a) $\dot{m}_{dif,ref}$ and $\dot{m}_{dif}$ (b) $\dot{m}_{sum,ref}$ and $\dot{m}_{sum}$ (c) v_{sup} and v_{exh}

Figure 13: Experimental results of Proposed method 2 (anti-windup in only sum mode).