Robust Flow Control of a Syringe Pump Based on Dual-Loop Disturbance Observers

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ABSTRACT Microfluidics, a field to deal with a small amount of fluid using a chip which had a lot of micro-scale channels, has applied in various fields including biology, chemistry, biomedical, and cell biology for recent few decades. One of the sub-categories of microfluidics is a droplet generation. A syringe pump is usually employed as an injection system. However, the injection system has some problems such as overshoot, steady-state error, and response delay. This paper grasped the problems of the conventional syringe pump, and then proposed a robust control method to control the flow rate precisely. Since the syringe pump includes parametric uncertainties in the system mainly due to the variation of the ball screw friction and the elasticity of the rubber, the robust control mechanism was designed with the cascade structure composed of the inner loop (i.e. disturbance observer and sliding mode control) and the outer loop (i.e. flow rate disturbance observer and 2-DOF control) against the undesired adverse effects. Finally, the proposed control scheme is evaluated through the experiment for its robustness and performance.

INDEX TERMS Microfluidics, flow rate control, disturbance observer, sliding mode control, 2-DOF control, flow rate disturbance observer, syringe pump

I. INTRODUCTION

MICROFLUIDICS is the research to deal with a small amount of fluid using a chip which had lots of micro-scale channels [1]. As the fluid passes through its channels, drops of fluid are generated by a shear force which breaks the stream of fluid. These drops undergo a series of processes, for example, mixing, separation, and analyzation [2]. The droplet-based microfluidics has attracted considerable researchers’ interests for effectively developing new technologies which satisfy diverse demands from various fields such as biology [3], chemistry [4], biomedical [5] and cell biology [6] in recent few decades because of its advantages. Firstly, it has economical efficiency since microfluidics uses very small quantities of samples and reagents in experiments. In particular fields, researchers have spent considerable money on their reagents. Thus, reducing reagents was a large economic advantage for them. Second, space and expense for experiments were dramatically reduced as the small chip replaced some large-bulk devices which have been typically used in experimental processes.

In droplet generation, two types of injection devices are typically used; pressure pumps [7] and syringe pumps [8]. The pressure pump has a fast response by feeding its pressure of the tube, and the flow rate is kept relatively constant with low fluctuation by the actuator. However, the pressure type pump is expensive compared to the syringe pump. On the other hand, the syringe pump is used as the most common injection system because it is economical and easy to install and operate. Nevertheless, the flow rate by the commercial syringe pump is still not reliable for droplet generation requiring the good quality of droplets of fluid [9] since the syringe pump operates with an open-loop mechanism. In addition, mechanical factors including a step motor, a ball screw, and gears cause vibration especially in the low speed of motor due to the cyclic variation and the friction of the ball screw. It makes the generation of droplet worse [10]. The sy-
ringe is also an important factor influencing the performance of generating precise droplet. The rubber type syringe has been usually used for its good leakage breaking capacity and low price. However, the syringe pump is challenged by its nonlinearity: although the piston of the syringe is moving, the fluid is not moved because of the elasticity of the rubber. For this problem, non-linear elements can be eliminated by using glass or stainless-steel type syringe, which are comparatively expensive.

In order to overcome problems mentioned above, many pieces of research have been conducted to explore the effectiveness of closed-loop feedback control for the precise flow rate control [11-15,34-36]. PI control [11,34] and PID control [35] method has been proposed to adjust the size of the droplet. The experiment results show that the closed-loop PID type control system reduced the effects of nonlinear factors. In [36], an iterative learning control has been used to improve precise control for flow position. It can modulate the input signal iteratively based on the experimental results. Flow pressure control using 4 bars linkage structure driven by a motor has been suggested in [12]. And, it focused on the positioning of fluid in micro size channel. However, these control methods can be affected by the effects of model uncertainties and unexpected disturbances. Heo et al. [13] analyzed the microchannel in terms of an electric circuit and controlled the flow rate by using a disturbance observer. They used a pressure type pump and considered a disturbance in the tube. The work in [14] proposed a disturbance observer-based nonlinear position tracking controller for electrohydraulic actuators. This method was adopted to estimate the disturbances caused by the biased sinusoidal disturbance and model uncertainties. For hydraulic servo systems, in addition, Yao et al. [15] presented an active disturbance rejection adaptive control strategy with extended state observer to handle parametric uncertainties and nonlinearities coming from variations in physical parameters. The developed scheme could avoid high-gain feedback by designing the adaptation law and the observer.

In this paper, a new rubber type syringe pump system is designed, replacing the step motor by a servomotor. With the new syringe pump, it is the primary objective of this paper to precisely control the flow rate and reduce the adverse effects of the syringe characteristics or other factors. In order to achieve the objectives, we suggest a cascade control structure to increase responsiveness and robustness through the velocity control for the piston and the flow rate control. By design of dual loop, the inner velocity controller can guarantee a robust control against the cyclic variation and friction of the ball screw. Not only that, the outer flow rate controller can ensure robustness against adverse effects of model variabilities and unpredictable disturbances and tracking performance to various flow rate references. For the inner loop, we introduce disturbance observer [16]-[19] and sliding mode control [20]-[22]. These two control methods have especially implemented in the mechanical control system to eliminate the adverse effect caused by a nonlinear factor such as the friction and variation of parameters. Especially, this paper employs a disturbance observer which has been introduced by Komada et al. [17] and Umeno et al. [31] for robustness and controller design. The outer loop, the robust flow rate control loop, employs flow rate disturbance observer and 2-DOF controller with the feedback of flow rate under the condition that the velocity control for the piston is guaranteed. Flow rate disturbance observer is applied to compensate for the nonlinear characteristics of rubber type syringes based on flow rate model. Therefore, this approach offers an alternative injection system to expensive commercial equivalents. It is a more flexible scheme for syringe pump systems.

This paper is organized as follows. In Section II, the problems of a conventional syringe pump system are described. In Section III, modeling of the mechanical system and the syringe system for the syringe pump are derived respectively.
Control methods are designed and analyzed for robust and precise flow rate control in section IV. Experimental results using the proposed control method and droplet experiment are shown in Section V. Finally, Section VI summarizes this paper.

II. PROBLEMS DESCRIPTION OF CONVENTIONAL SYRINGE PUMP SYSTEM

A conventional syringe pump system is one of the popular devices as an injection source in microfluidics. It has been used for a few decades because of its conveniences and advantages in an economical aspect. With the development of microfluidics, the injection system is also being required to satisfy various requirements. However, existing syringe pumps have weaknesses including overshoot, steady-state error, and response delay in terms of flow rate. In order to explain those problems, an experiment conducted with NE-8000, commercial syringe pump, which has references of 10, 50, 100 µL/min. Therefore, this section describes the problems with the experimental results of Figure 1 as follows.

1) Syringe: In order to fulfill the high level of performance in precision and response, sealing part of the syringe is regarded as an important factor. With the most commonly used and inexpensive rubber-type syringe, however, its elasticity brings about the overshoot of transient flow rate in Fig. 1 (a).

2) Inaccuracy: The conventional syringe pump system was operated based on the pre-collected data without a feedback loop. Therefore, the system could not reflect the real-time flow rate. In Fig. 1 (b), fluctuations of fluid were frequently observed in steady state phase because of the open-loop architecture.

3) Responsiveness: Since there is no feedback loop, it is difficult to carry out fast and precise flow rate control. In Fig. 1 (c), the syringe pumps have a long response time from a few seconds to a few hours. This phenomenon causes an unnecessary waste of reagents.

4) Limitation: Since the operating range of flow rate were limited by mechanical factors (e.g., the resolution of a step motor, the gear ratio, and the diameter of a syringe), the controllable flow rate range must be extended in syringe system.

III. MODELING OF SYRINGE PUMP SYSTEM

In this section, modeling of the suggested syringe pump system is introduced. As shown in Figure 2, the system consists of the syringe system and the mechanical system. The syringe system is a relationship between the moving of piston and the flow rate coming out from the syringe. And, the mechanical system involves a motor, a ball-screw, and a piston of the syringe.

A. SYRINGE SYSTEM MODELING

The detail view of the syringe part is shown in Figure 3. The flow rate is the quantity of liquid moving through a pipe or channel within a given or standard period. It is constant in one channel if there are no energy losses like leakage, friction-head loss. In this condition, the flow rate is obtained by multiplying the internal area of syringe and velocity of the fluid.

\[ Q_{fi} = Q_{fo} \]
\[ Q_{fi} = A_p v_{fi}, \quad Q_{fo} = A_{fo} v_{fo} \]

where \( Q_{fi} \) is the flow rate near piston, \( A_p \) is the internal area of the syringe, \( v_{fi} \) is the velocity of the fluid near piston, \( Q_{fo} \) is the flow rate in channel of flow rate sensor, \( A_{fo} \) is the internal area of flow rate sensor, and \( v_{fo} \) is the velocity of the fluid in the channel of flow rate sensor. Note that, the velocity of the piston \( \dot{x}_p \) and the velocity of the fluid \( v_{fi} \) is not equal due to the rubber of the piston in transient response. Therefore, the behavior of the flow rate can be considered as a first order system,

\[ \tau_Q \dot{Q}_{fi} + v_{fi} = \dot{x}_p + d_Q \]

where \( d_Q \) is unmodeled disturbances mainly caused by the elasticity of rubber. Using Eq. (1)-(2), Eq. (3) can be rewritten with respect to flow rate as follows

\[ \frac{1}{A_p} (\tau_Q \dot{Q}_{fi} + Q_{fi}) = \dot{x}_p + d_Q \]

Accordingly, the transfer function \( P_Q \) between \( \dot{x}_p \) and the flow rate \( Q_{fo} \) is represented as

\[ P_Q = \frac{A_p}{\tau_Q s + 1} \]
where $A_p$ is the inner area of the syringe and $\tau_Q$ is the time constant of the flow rate model.

**B. MECHANICAL SYSTEM MODELING**

Figure 4 shows the detail view of the mechanical part, which is composed of the driving motor and the piston part. The dynamic model of the driving motor is considered and it is described as follows

$$J_m\ddot{\theta}_m + B_m\dot{\theta}_m + \tau_c = \tau_m$$

(6)

where $J_m$ is the moment of inertia, $B_m$ is the viscous friction coefficient of the motor, $\tau_m$ is the motor driving torque, $\tau_c$ is the coupling torque, $\dot{\theta}_m$ and $\ddot{\theta}_m$ are the angular acceleration and the angular velocity of the motor, respectively. $J_m$ includes each moment of inertia of reduction gear and the encoder. In addition, $\tau_c$ is applied to each system, the driving motor and the piston part. It plays a role as a mechanical connection and is expressed as

$$\tau_c = K_c\Delta\theta = K_c(\theta_m - \theta_p)$$

(7)

where $\Delta\theta$ is the torsional angle of the coupling generated by torsional torque between the motor shaft and the ball screw axis, $K_c$ is the constant spring stiffness of the coupling. Assuming the spring stiffness of the coupling is big enough to ignore the torsional deflection $\Delta\theta$. With this assumption, the ball screw angle is considered to be same as the motor angle and Eq.(6) can be rewritten as

$$J\ddot{\theta}_m + B\dot{\theta}_m + \tau_p = \tau_m$$

(8)

where $J$ represents the equivalent moment of inertia of the motor and the coupling, $B$ is the equivalent friction coefficient of the motor and the coupling, $\tau_p$ is the driving torque which generates the translational force to move the piston.

The dynamic equation of the piston part is subsequently considered. The piston parts include all of the devices such as the piston of the syringe, the piston holder, and the ball screw. The governing equation of the piston holder is expressed as follows

$$M_p\dddot{x}_p + C_p\ddot{x}_p = F_p, \quad x_p = \frac{L}{2\pi}\theta_m$$

(9)

where $M_p$, $C_p$, $\dddot{x}_p$, and $\ddot{x}_p$ indicate the piston parts of mass, viscous friction coefficient, acceleration, and velocity, respectively and $L$ is the ball screw pitch distance. Additionally, the position of piston holder $x_p$ can be assumed to be proportional to the ball screw angle $\theta_m$. In the mechanical system, the driving torque generated by the motor is converted by the ball screw to the force of the linear motion of the piston $F_p$. In Eq. (9), $F_p$ is the ball screw axis force caused by $\tau_p$ and their relationship is derived as.

$$F_p = K_p\tau_p$$

(10)

where the translational force constant $K_p$ is to convert the driving torque into the linear force by the ball screw. The Eq. (8) - (10) of the above are modified using Laplace transform and expressed as follows

$$Js^2\theta_m(s) + Bs\theta_m(s) + T_p(s) = T_m(s)$$

(11)

$$M_p s^2 x_p(s) + C_p s x_p(s) = K_p T_p(s) + F_d(s)$$

(12)

**C. SIMPLIFIED MECHANICAL MODEL FOR CONTROL DESIGN**

For the sake of simplicity in control design, the mechanical system is simply modeled in regard to the translation motion because the flow rate is generated by the translation motion. The simple mechanical part dynamics is described by the following equation

$$M\dddot{x}_p + C\ddot{x}_p = K_p\tau_m + F_d + F_f$$

(13)

where $F_d$ is the unexpected external disturbances, $F_f$ is the friction force owing to the elasticity of rubber in the syringe, $M$ is the equivalent mass of the mechanical system, $C$ is also the equivalent friction coefficient from the high rate reduction gear and the ball screw as driving part. Since $M$ and $C$
include model uncertainties, these are defined as

\[ M = M_n + \Delta M \]
\[ C = C_n + \Delta C \]  \hspace{1cm} (14)

where \( M_n \) is the nominal mass of the mechanical system, \( C_n \) is the nominal friction coefficient of the mechanical system, \( \Delta M \) and \( \Delta C \) are parameter variations of \( M_n \) and \( C_n \), respectively. In order to facilitate a control design and implementation, the frequency response is processed to define a nominal model of the mechanical system of the syringe pump. In the experiment, sine sweep torque command with wide range frequency (0.1Hz to 100Hz) is applied to the driving motor. Then, the angular velocity of the driving motor is measured from the motor encoder. It is converted to the piston velocity based on the relationship in Eq. (9). With measurements, the result of the frequency response from driving torque to the velocity of the piston is shown in Figure 5. This experiment is repeated with the various magnitude of torque commands from low to high and its results are also plotted in Fig. 5. As shown in Fig. 5, the response of the mechanical system varies in low-frequency region with respect to input torque commands. This phenomenon is occurred due to the model uncertainty mainly caused by the cyclic variation and the friction of the ball screw. Consequently, the analysis of frequency response indicates that the mechanical part can be approximated by the first-order system instead of a high order system for control implementation. Here, let \( \hat{d} \) newly define as a lumped disturbance which includes parameter perturbation terms, external disturbances, and friction forces. It is represented as

\[ \hat{d} = F_d + F_f - (\Delta M \ddot{\theta}_p + \Delta C \dot{\theta}_p) \]  \hspace{1cm} (15)

where the lumped disturbance \( \hat{d} \) is assumed to be bounded and defined by

\[ \|F_d + F_f\| \leq \alpha \]
\[ \|-(\Delta M \ddot{\theta}_p + \Delta C \dot{\theta}_p)\| \leq \beta \]
\[ \|\hat{d}\| \leq \alpha + \beta \leq \eta \]  \hspace{1cm} (16)

where \( \alpha \) is the upper bound of the disturbance force and the friction force, \( \beta \) is the upper bound of model uncertainties and \( \eta \) is the upper bound of the lumped disturbance. In order to design the robust and precise controller for the mechanical system, the transfer function from the motor torque to the velocity of the piston is obtained as follows

\[ P_m = \frac{K_p}{M_s + C} \]  \hspace{1cm} (17)

IV. DESIGN ROBUST AND PRECISE FLOW RATE CONTROL

The main objectives of the proposed controller are: to realize fast and precise flow rate control; to enhance robustness against unmodeled dynamics and disturbances. Figure 6 shows the overview of the control algorithm which is divided by two subsections. One is about the inner-loop controller design for the velocity control of the piston. In the velocity control, a combination of disturbance observer (DOB) and sliding mode control (SMC) is introduced. The other is the flow rate control, the outer-loop controller. Under the precise velocity control, flow rate disturbance observer (FDB) is used to make the syringe pump system more robust against the nonlinearity of friction coming from the ball screw and the rubber.

A. DESIGN OF A ROBUST VELOCITY CONTROLLER

From Eq. (15), the newly defined lumped disturbance \( \hat{d} \) consists of predictable disturbances (e.g., heuristic model) and unpredictable disturbances. It means the lumped disturbance
can be decomposed into high-frequency components and low-frequency components, as shown in Figure 7. To deal with the disturbance, this paper employs sliding mode control (SMC) and disturbance observer (DOB) [24]. While SMC works to improve tracking performance and to guarantee the robustness of the proposed control method resisting to the high-frequency components of the lumped disturbance, DOB makes the mechanical part behave as a defined nominal model against model uncertainties and disturbances in low-frequency range [25], [26]. In this way, the high-frequency switching gain can be designed as a smaller value, which alleviates the chattering problem [26].

1) Disturbance observer

The disturbance observer is taken to alleviate the chattering problem of SMC and maintain the nominal performance of the proposed scheme. Additionally, the DOB-based control enhances the robustness against the lumped disturbance in low-frequency range by compensating it using a low-pass filter \( Q(s) \). Therefore, two parameters called \( Q \) filter and the nominal model become the key factors in the design of DOB.

Based on the experiment of the frequency response for the mechanical system and Eq. (17), the transfer function for the nominal velocity control model which can accurately describe the frequency response measurement (i.e., dashed red line in Fig. 5) is selected as

\[
P_{m.n} = \frac{K_p}{M_n s + C_n} \tag{18}
\]

where \( M_n \) and \( C_n \) are obtained by matching the frequency response measurement [27], [28].

Generally, since the performance of DOB depends on a \( Q \) filter bandwidth, it is critical to design an appropriate \( Q(s) \) such that \( Q(s)P_{m.n}^{-1} \) is implementable and \( Q(s) \) is stable for the overall control system. In this aspect, the order of the velocity control \( Q \) filter \( Q_v(s) \) is determined according to the order of the nominal model. A possible \( Q_v(s) \) is given by

\[
Q_v(s) = \frac{\omega_{c dob}}{s + \omega_{c dob}} \tag{19}
\]

where \( \omega_{c dob} \) is the cut-off frequency of the \( Q_v(s) \) which is determined by considering control performance requirements. Thus, through the nominal model and the \( Q \) filter, the lumped disturbance is estimated as the difference between the nominal input \( u_n \) and the actual control input \( u_{\hat{x}_p} \) in the velocity control

\[
\hat{d} = Q_v(s) \dot{x} = Q_v(s)[u_n - u_{\hat{x}_p}] = Q_v(s) \left[ P_{m.n}^{-1} \dot{x}_p - u_{\hat{x}_p} \right] \tag{20}
\]

where \( \hat{d} \) is the estimated lumped disturbance which is eliminated by DOB.

2) Sliding mode control

Sliding mode control is utterly robust to the lumped disturbance including parametric uncertainty and external disturbances in high-frequency range with switching function. From Eq. (13), the dynamic equation of the mechanical system is represented as follows

\[
M_n \ddot{x}_p + C_n \dot{x}_p = K_p \tau_m + \hat{d} \tag{21}
\]

\[
\dot{x}_p = \frac{1}{M_n} \left( K_p \tau_m - C_n \dot{x}_p + \hat{d} \right) \tag{22}
\]

The tracking error \( \epsilon \) is defined based on the motor angular velocity

\[
\epsilon = \dot{x}_{p, ref} - \dot{x}_p \tag{23}
\]

where \( \dot{x}_{p, ref} \) is the desired velocity of the piston holder and \( \dot{x}_p \) is the piston holder velocity which is converted from the motor angular velocity. In order to design an appropriate control law with a zero tracking error, the sliding surface \( \sigma \) is defined as a function of error [29]

\[
\sigma = \epsilon \tag{24}
\]

here, \( \sigma = 0 \) means \( \dot{x}_p \) accurately tracks \( \dot{x}_{p, ref} \). For asymptotic stability, the sliding surface should satisfy the reaching condition as follows

\[
\dot{\sigma} = -K \sigma - \rho \text{sgn}(\sigma) \tag{25}
\]

where \( K \) is the positive control gain which denotes the convergence rate of the tracking error and \( \rho \) is the high-frequency switching gain. The following desired control law can be constructed from (22) - (25)

\[
\tau_{m, law} = \frac{1}{K_p} \left[ M_n \ddot{x}_{p, ref} + C_n \dot{x}_p + K M_n \sigma + \rho M_n \text{sgn}(\sigma) \right] - \hat{d} \tag{26}
\]

where the high-frequency switching gain \( \rho \) should be greater than the upper bound of the lumped disturbance and it should satisfy the condition such that

\[
\frac{\eta}{M_n} \leq \rho \tag{27}
\]

here, the maximum \( \rho \) is limited by the maximum torque that the driving motor can generate. In order to prove the stability of the suggested control method, a positive-definite Lyapunov function candidate is employed as follows

\[
V = \frac{1}{2} \sigma^2, \quad V > 0 \tag{28}
\]
The derivative of (28) is expressed as follows

\[
\dot{V} = \sigma \dot{\sigma} = \sigma \left( \ddot{x}_{p,\text{ref}} - K_p \tau_m + \frac{C_n}{M_n} \dot{x}_p - \frac{1}{M_n} \ddot{d} \right)
\]

\[
= \sigma \left( -K \sigma - \rho \text{sgn}(\sigma) + \frac{1}{M_n} (\dot{d} - \dot{\tilde{d}}) \right)
\]  

(29)

Substituting the estimated lumped disturbance (20) into (29), yields

\[
\dot{V} = \sigma \left( -K \sigma - \rho \text{sgn}(\sigma) + \frac{1}{M_n} (Q_v(s) - 1) \dot{\tilde{d}} \right)
\]  

(30)

With an assumption \(Q_v(s) \approx 0\) in the high-frequency range, it is rewritten as

\[
\dot{V} = -K \sigma^2 - \rho |\sigma| \text{sgn}(\sigma) - \frac{1}{M_n} \dot{\tilde{d}}
\]

\[
= -K \sigma^2 - \rho |\sigma| \frac{1}{M_n} \dot{\tilde{d}}
\]

\[
\leq -K \sigma^2 - \rho |\sigma| \frac{\eta}{M_n} < 0
\]  

(31)

The above stable condition guarantee that SMC method tries to pull the velocity of the piston to converge into its desired velocity reference with high speed. However, since a too big \(K\) will require high control input, a compromise between the control input and the response speed should be made. Therefore, the control parameter \(K\) cannot be selected too large. Consequently, when the two conditions presented in Eq. (27) and Eq. (31) are satisfied, the tracking error of velocity asymptotically converges to zero, i.e. the system is stable.

3) Experiment of closed velocity loop

In order to validate the robustness of the velocity control performance using both disturbance observer and sliding mode control, experimental closed velocity control is shown in Figure 8. The experimental FRF result is marked as the measured velocity closed loop and modeled velocity closed loop.

![Figure 8: Experimental closed velocity control loop.](image)

![Figure 9: Experiment to define nominal flow rate model depending on the velocity of the piston. A solid line is a measured flow rate and a dotted line is a modeled flow rate; (a) 0.02 mm/sec, (b) 0.05 mm/sec, (c) 0.1 rad/sec, (d) 0.5 mm/sec](images)
black solid line. Due to SMC and DOB, the closed inner-loop is considered as the nominalized system which has a large enough control bandwidth. Therefore, it is reasonable to assume that the closed inner loop transfer function from $\dot{x}_{p,ref}$ to $\dot{x}_p$ can remain close to one with a large control bandwidth.

### B. DESIGN OF THE FLOW RATE CONTROLLER

The rubber has been used in various industries as sealing devices because of its inexpensiveness and fine seal-ability. However, especially for the nonlinear friction characteristics of the rubber, the rubber type syringe can be challenging for the injection application. The elasticity of the rubber affects dynamics between the velocity of the piston and the flow rate as seen before in section II. To overcome the adverse effect of the rubber, the proposed outer-loop controller for the flow rate is constructed with robust control methods, FOB and 2-DOF controller, in this paper.

1) Flow rate disturbance observer

With FOB, the robust flow rate control system is realized by compensating disturbances $d_Q$ such as variation of parameters and unmodeled dynamics. The principle of FOB is similar to DOB which is introduced in subsection IV.A.1. In order to define the nominal model for the flow rate system, the pulse shape of flow rate commands are applied to the syringe pump under robust and precise velocity control for the piston. Then, the flow rate is measured by the flow rate sensor with the open loop of the flow rate. The results of the experiment are shown in Figure 9. The decreasing tendency of the flow rate is matched with the first-order time delay. Therefore, we considered the dynamics of the rubber as the first-order time delay. The first-order model has a limitation to describe the increasing part of the flow rate. However, the disturbance observer-controlled systems can behave as the nominal model which is matched with the first-order model. It means that the unmodeled dynamics in the increasing parts is compensated by FOB. Based on Eq. (5) and defined nominal parameters, the transfer function for the nominal flow rate system is defined as follows,

$$ P_{Q,n} = \frac{A_n}{\tau_{Q,n}s + 1} \quad (32) $$

where $P_{Q,n}$ is the nominal model between the velocity of the piston and the flow rate, $A_n$ is the normalized inner area of the syringe, and $\tau_{Q,n}$ is also the nominal time constant of the flow rate system. The order of flow rate Q filter $Q_Q(s)$ is likewise designed depending on the flow rate nominal model as follows

$$ Q_Q(s) = \frac{\omega_{Q,dob}}{s + \omega_{Q,dob}} \quad (33) $$

where $\omega_{Q,dob}$ is the cut-off frequency of $Q_Q(s)$ for FOB. The offset between the nominal model and the real flow model is compensated by using FOB. Therefore, the syringe pump becomes robust from the disturbance such as deformation of the rubber.

2) Two degrees of freedom controller

In order to improve the tracking performance for the reference flow rate, the flow rate controller also introduces the feedforward and the feedback controller, 2-DOF controller. The detail description is shown in Fig. 6. With the flow rate disturbance observer, uncertain flow rate model is modified to the nominal model for the flow rate model which we define in Eq. (32). Therefore, the 2-DOF controller is designed based on the defined nominal model [31].

The feedback controller is constructed with the conventional proportional-integration (PI) controller using pole-zero cancellation method, which reduces the flow rate tracking error [30]. Under the assumption that the closed inner loop
The system can be regarded as one in the control bandwidth of the inner loop, the transfer function from the flow rate reference \( Q^* \) to the flow rate \( Q \) is

\[
\frac{Q}{Q^*} = \frac{C_{fb} P_Q}{1 - Q_Q + C_{fb} P_Q + P_Q P_{Q,n}^{-1} Q_Q} = \frac{\omega_{fb}}{s + \omega_{fb}}
\]

(34)

where \( C_{fb} \) is the feedback controller and \( \omega_{fb} \) is the feedback control bandwidth. Here, with the assumption of \( P \simeq P_{Q,n} \) and the condition of \( Q_Q \approx 0 \), the transfer function is rewritten as

\[
\frac{Q}{Q^*} = \frac{C_{fb} P_Q}{1 + C_{fb} P_Q} = \frac{\omega_{fb}}{s + \omega_{fb}}
\]

(35)

Consequently, the feedback controller is designed as

\[
C_{fb}(s) = K_{P_Q} + \frac{K_{i_Q}}{s} = \frac{\tau_Q \omega_{fb}}{A_n} + \frac{1}{A_n s}
\]

(36)

where \( K_{P_Q} \) is the proportional gain and \( K_{i_Q} \) is the integration gain.

The closed-loop dynamics for the outer loop can limit the tracking performance of the flow rate if the output is directly used as the feedback input due to the time delay. To compensate for the delay, the desired control output needs to be compensated by the feedforward controller [32]. The feedforward controller \( C_{ff}(s) \) is designed based on the inversion of the nominal flow rate model [33] and is given by

\[
C_{ff} = P_{Q,n}^{-1} = \frac{\tau_{Q,n} s + 1}{A_n (\tau_{Qff} s + 1)}
\]

(37)

where \( \tau_{Qff} \) is the time constant of the low-pass filter which reduces the amplification of high-frequency noises due to the effect of time derivative.

V. EXPERIMENT

A. EXPERIMENTAL SETUP

The description of the whole control system is illustrated in Figure 10 and the experimental syringe pump device is shown in Figure 11. Maxon BLDC motor is used as a driving source and the driving torque is transmitted to flow rate through the coupling, the piston, and the ball screws (1mm pitch). An adopted size of syringe diameter is 12mm to verify the performance of the flow rate disturbance observer for appropriate conditions. To measure the flow rate, a product of SENSIRION called SLI-1000 was used as a flow rate sensor, and MATLAB/Simulink is used as control software. Quarc’s Quanser QPIDe was used as a DAQ device which has a real-time control with 1ms sampling time. The chips were observed with an inverted microscope (Olympus IX73). Images were obtained using a high-speed camera (Photron IDP Express) and recorded as 30fps (frame per second) using a recording program. The exposure time of each frame was...
TABLE 1: Explanation on Three Types of Flow Rate Controllers

<table>
<thead>
<tr>
<th>Case (a)</th>
<th>Case (b)</th>
<th>Case (c)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Feedback</td>
<td>X</td>
<td>O</td>
</tr>
<tr>
<td>DOB</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>RMS error [µL/min]</td>
<td>12.08</td>
<td>3.049</td>
</tr>
</tbody>
</table>

adapted with respect to the reference flow rate.

B. EXPERIMENTAL RESULT

In order to evaluate the proposed dual-loop flow rate control method, we compared the three cases under the robust velocity control; (a) piston velocity control without flow rate control, (b) 2-DOF flow rate control only, and (c) FOB with 2-DOF controller. Figure 12 shows the experiment results with flow rate references at 10, 25, 40, 55 [µL/min]. It can be seen in Figs. 12 (a) and (d) that even though the piston driving by the motor keeps moving under the robust velocity control corresponding to the flow rate reference, the fluid is still stationary because of the elasticity of the rubber at the beginning and low flow rate. With a little more input command of the flow rate in Fig. 12 (a), the condensed fluid instantaneously starts to pour out with causing the unintended overshoot at 14 seconds as the translational force of the piston overcomes the maximum static frictional force point for the syringe. Also, the steady-state error and fluctuation are observed in the high flow rate region due to no feedback of the flow rate. Figs. 12 (b) and (d) shows the result of the flow rate feedback control via the 2-DOF controller. For high flow rate references, the 2-DOF controller allows better tracking performance with alleviating the steady-state error and the fluctuation. In addition, the response becomes much faster in the transient phase. However, the overshoot and the fluctuation are still observed in the range of low flow rate references. The result of the experiment in Figs. 12 (c) and (d) represents the effect of FOB. It implies that FOB eliminates the overshoot phenomenon by compensating the adverse effects of disturbances and model uncertainties. Moreover, the tracking performance and fast response for all the range are achieved through applying FOD. For quantitative evaluation of the proposed method, the RMS errors for each configuration are compared in Table 1. According to the RMS error, it can be said that the proposed dual-loop controller can guarantee a robust flow rate control against the effects of model variability and unpredictable disturbances, e.g. the friction of the ball screw and the nonlinear characteristic of the rubber.

Droplet generation experiment using the proposed flow rate control method is shown in Figure 13, which validates its fast responsiveness and controllability. Oil is injected into the tubes above and below the cross-shaped chips, and the water with a fluorescent material in the left tube is injected via the syringe pump. Small droplets are created by the orifice.
structure in the middle of the tube and moved to the right. Fig. 13 (a) shows precisely generated droplets according to the various flow rate references. The size of droplets and the generation frequency are quite uniform without fluctuation in the channel. Fig. 13 (b) shows the fast transient responses even though the reference decreased from 20 [µL/min] to 5 [µL/min] with the proposed control algorithm. By employing the proposed controller, therefore, the system becomes robust to unmodeled uncertainties and external disturbances. It enables high-precision and high-accuracy flow regulation with fast response.

VI. CONCLUSIONS
A new injection system to generate the precise droplet was proposed and investigated with experiments. The main contribution of this paper is the introduction of the dual loop structure to control the flow rate precisely. The robust velocity control of the piston based on disturbance observer and sliding mode control is designed as the inner loop. Its robustness against the external disturbances and the parameter perturbation is demonstrated by presenting its closed-loop frequency response. For the outer loop, we defined a relationship between the velocity of the piston and flow rate using the pulse response to design the flow rate disturbance observer and 2-DOF controller. By employing the flow rate disturbance observer and 2-DOF controller, the freedom of control design can be increased and the nonlinear characteristic of the syringe rubber can be compensated. Thus far, the conventional syringe pump system has not been able to provide accuracy and fast response with other limitations. However, the proposed dual-loop control structure guarantees the significant reduction of overshoot and the improvement of the tracking performance in the low flow rate range. Not only that, the proposed control scheme here ensures a fast response to the desired flow rate. The effectiveness was demonstrated by the experiment results.

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