Adaptive Feed-forward Control of the Clutch Filling Phase for Wet Dual Clutch Transmission

Sanghun Jung, Seibum B. Choi, Jinsung Kim, Youngho Ko, Hoyoung Lee

***Abstract*—Clutch fill control in clutch-to-clutch shift process is an essential technique to ensure smooth and precise synchronization of clutches. The filling control methods proposed in previous works are difficult to apply to production vehicles because of huge calibration effort and numerical issues of the system model. Therefore, this paper focuses on the development of clutch fill control logic considering practical issues at the production vehicle level. In order to increase the usability, a simplified reduced model that describes the filling phase is designed. The proposed model is used to construct a model-based feed-forward controller, which is not affected by a time delay in the hydraulic system. Furthermore, an adaptation algorithm of a key model parameter is proposed to cope with the model changes due to the disturbances and uncertainties of the plant. In addition, a new volume indicator is proposed to observe the status of the filling phase. Based on the proposed controller and the indicator, tracking performance of the filling phase is guaranteed without over-fill or under-fill. As a result, the proposed control logic is applied to a production type transmission control unit in a test vehicle and its performance is experimentally verified through various scenarios.**

***Index Terms*—Clutch fill control, Hydraulic clutch control, wet clutch, Wet dual clutch transmission, Adaptive feed-forward control**

I. Introduction

In vehicles using internal combustion engines or high power electric vehicles, transmission systems are essential for improving fuel economy and for efficient operation [1]–[4]. In order to improve the efficiency, various transmissions and shift logics have actively been developed [5]–[7]. Among the transmissions, the Dual Clutch Transmission (DCT) has advantages of high efficiency and fast shift performance [8], [9]. DCT can be classified into two types according to the clutch type: dry DCT and wet DCT (wDCT). In the case of dry DCT, it is difficult to use in heavy-duty vehicles due to wear and heat problems. wDCT compensates for these shortcomings and is being applied to heavy-duty vehicles for mass production. However, wDCT has difficulties in control due to the nonlinearity of fluid in friction plates. Therefore, precise torque and slip control is required during the shifting process in wDCT [10].

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Unlike the Automatic Transmission (AT), DCT has no smoothing effect on the torque converter, which has a direct adverse effect on ride quality when inappropriate shift control is applied. For smooth and precise torque transmission, the clutch pistons should be controlled to the position where the clutch torque begins to be transmitted. This is called ‘clutch filling control’, which is an important technique in the clutch-to-clutch shift process. If the filling control is not appropriate, under-fill or over-fill can occur, which results in bad shifting results such as torque deep and clutch tie-up. Especially in wDCT, a drag torque is generated even before the piston touches the friction plate [11]. This causes the same effect as over-fill. Therefore, a pre-fill control process of wet DCT will inevitably aim to under-fill, which makes a delayed pressure output at the beginning of the torque phase. In order to guarantee good shift performance, it is necessary to compensate the filing process in an early stage of the torque phase.

Electro hydraulic valve(EHV) based on a proportional solenoid valve (PSV) are common in wDCT actuator systems due to their high energy density and control convenience [12]. Actuators with a PSV have the advantage that the pressure can be controlled proportionally. However, in the case of the clutch filling process described above, nonlinearity of the pressure occurs due to the change of volume in the clutch chamber, and this phase is called the clutch filling phase [13]. This nonlinearity in the filling phase causes a lagged pressure response, which can deteriorate the shift quality. Therefore, control of filling phase is necessary to achieve the smooth shift quality. Moreover, clutch fill control is a very sensitive control since small control errors can lead to over-fill or under-fill. This means the clutch fill control requires a fast response, stability, accuracy, and robustness.

Many previous studies have been conducted on control of the clutch filling phase. In order to precisely describe the filling phase, [14] proposed a precise mathematical model of an EHV. These models are used to predict the behavior of the system and are the basis for developing control logic. [13], [15], [16] proposed an optimal clutch fill control logic based on an optimization strategy using dynamic programming and optimal theory. [17], [18] designed a sliding mode control to enhance the control robustness of an EHV. As outlined above, many methods for model-based control using exact models have been studied.

However, most of the previous studies have the shortcoming of not considering practical problems. EHVs equipped in production vehicles have some nonlinearities and hardware limitations that adversely affect control performance. Time delay is well known typical nonlinear behavior of an EHV [19]. If time delay exists, the control performance and stability will be affected by the delayed measurement. In addition, the exact models used for designing controller contain numerically stiff models such as pressure dynamics. To apply the control logic to a production vehicle, the model should be discretized at a general transmission control unit (TCU) sampling rate of 100Hz. However, it is difficult to express the stiff dynamics using a 100Hz sampling rate.

Due to these limitations, rule-based fill controls have been implemented in production vehicles. A number of studies have been done by using predefined control input and updating the input shaping parameters [20]–[24]. [22], [23] analyzed the factors affecting the filling process and constructed a fuzzy controller for pressure control. [24] proposed a logic for smooth shift control by adapting input shaping parameters in real time. However, these methods require a relatively long preparation time of a pre-fill region, which results in a longer shift time.

This paper proposes a new control logic for the clutch filling process for vehicles equipped with wDCT. The proposed controller is designed considering production vehicle application. In order to simplify the system model, a reduced order model is introduced to construct a control-oriented model. Based on the model, a model based feed-forward controller is suggested to generate fast and stable control inputs and minimize the effect of the time delay without any preparation process. In addition, an adaptive law is designed to enhance the robustness of the feed-forward controller. Overall control logic is implemented with real-time and verified through both simulations and experiments in test vehicles. The proposed logic is anticipated to improve the shift quality of wDCT.

The rest of this paper is organized as follows. In section 2, the EHV system is analyzed and a reduced model is presented. The proposed reduced model is verified using experimental results. In section 3, the design process of filling phase control is demonstrated. A feed-forward controller based on a control-oriented model is constructed. Also, an adaptive law that updates the main parameter is obtained to improve the control performance. Sections 4 and 5 verify the effectiveness of the proposed controller using simulation and experimental results, respectively. Conclusion and directions for future work are provided in section 6.

II. System Modeling

A. System description

A general illustration of a clutch actuation system using an EHV is presented in Fig.1. The EHV actuator consists of two parts: a PSV and a clutch chamber. The PSV generates pressure proportional to the input current through its own pressure feedback line. The flow and pressure generated from the PSV are connected to the clutch chamber to pressurize the clutch to engage or disengage. The system of interest in this paper is equipped with a pressure sensor to measure the pressure in the clutch chamber. The measured pressure data are used for control and analysis.

Many previous efforts have been conducted to accurately model the EHV [12], [14], [25]. These papers have analyzed

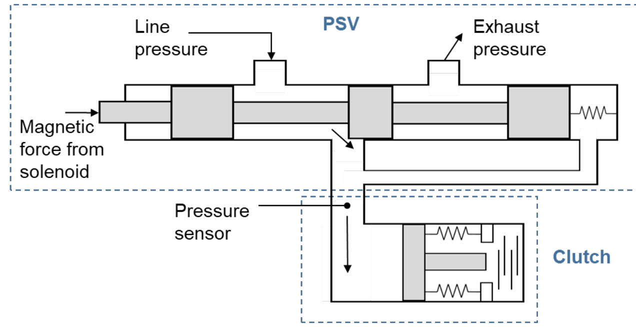


Fig. 1: Schematic diagram of the electro-hydraulic actuator in wet DCT

and modeled the effects of internal components of the valve such as spool dynamics and flow force. These makes a complex model to describe the behavior of the EHV. However, this paper focuses on and analyzes the behavior of the clutch chamber, which has a more dominant effect than the valve system. Thus, several assumptions are made to ignore the internal dynamics of the EHV system. First, the PSV can be treated as a static system because its bandwidth and response time are much faster than the control bandwidth. Second, nonlinear effects caused by the PSV, such as stiction or dead-band effects, are negligible. Finally, hysteresis behavior of the relationship between the current input and the pressure output (I-P) is not covered in this paper since it has been fully identified in other previous studies [26]. Therefore, the valve output pressure P_A is defined as the control input u of the clutch chamber in this study. The valve output pressure P_A is assumed to track the control input u directly through the identified current-pressure relationship.

The powertrain system of interest uses a normally open clutch, as illustrated in Fig.1. When a shift signal is triggered from the TCU, a pressure trajectory is generated corresponding to the shift strategy. The pressure of the clutch chamber is increased by applying current to the PSV. As the pressure increases, the clutch piston overcomes the force of the return spring and approaches the friction plates until it contacts the friction plate. The clutch filling process refers to the situation in which the above-mentioned clutch piston moves. Many previous studies have mathematically modeled the behavior of the filling phase [13], [15]. The model includes the piston dynamics, which represents the movement of the clutch piston, and the pressure dynamics, which represents the change in pressure. The representative dynamic model is expressed as follows:

$$\begin{bmatrix}\dot{x}_1\\ \dot{x}_2\\ \dot{x}_3\end{bmatrix} = \begin{bmatrix} x_2 \\ \frac{1}{M_p}[A_c x_3 - B_p x_2 - k_p x_1 - F_{fric} - F_{pre}] \\ \frac{\beta}{V_0 + A_c x_1}[sign(u-x_3) C_d A_p \sqrt{\frac{2|u-x_3|}{\rho}} - A_c x_2] \end{bmatrix}$$
$$[x_1, x_2, x_3] = [x_p, \dot{x}_p, P_c] \tag{1}$$

where x_p is the piston displacement, P_c is the pressure of the clutch chamber, M_p is the piston mass, B_p is the viscous friction coefficient, K_p is the spring stiffness, A_c is the effective area of the piston, F_{fric} is the Coulomb friction of the piston, F_{pre} is the spring pre-load force, V_0 is the initial volume of the clutch chamber, β is the bulk modulus, C_d is the discharge coefficient, A_p is the pipe area, and ρ is the fluid density.

As mentioned above, (1) describes the interaction between piston dynamics and pressure dynamics. However, since the relationship between these dynamic is nonlinear, it is difficult to apply it to controller design because the analytic solution cannot be derived directly. In addition, it can be seen that there are a number of unknown parameters. Parameter errors increase model uncertainty and cause degradation of controller performance. Furthermore, the difference in dimension of each state in (1) is too large, which causes a numerical problem. In (1), x_p and $\dot{x}_p$ are under 10^{-3}, but P_c is over 10^5. Also, the pressure dynamics is a stiff system due to the large value of the bulk modulus [27], [28]. The stiff system and difference of dimension make the model too numerically unstable to be utilized in the controller design. Therefore, a reduced model is needed to construct a real-time controller that can be applied to production vehicles.

B. Reduced model design

Here, a reduced model based on physical phenomena is provided. As mentioned above, (1) is difficult to apply to production vehicles due to practical issues such as numerical instability and parameter uncertainties. To simplify the model, the EHV system is redefined as a reduced form with some assumptions. The contents of this section are borrowed from the same authors' paper [11].

For convenience of modeling, the pressurized process of the clutch chamber is divided into three phases based on the piston position and the clutch chamber pressure. Fig.2. shows a schematic diagram of the three phases of the clutch actuation process. The change of the pressure output according to the each step can be determined using the pressure output data. Fig.3. describes the open loop response to the ramp-up input of the clutch pressure. As described in the figure, it can be seen that the system has a take off point (TOP) where the clutch piston overcomes the return spring and a volume kissing point (VKP) where the clutch piston touches the friction plates. The main focus of this paper, the filling phase, is the phase between TOP and VKP where the piston moves. It can be confirmed that the pressure gradient decreases dramatically in the filling phase because of the volume change. Based on the pressure, each phase can be defined as follows: phase 1; pre-filling phase in $0 \leq P_c < P_{TOP}$ (Fig.2a), phase 2; filling phase at $P_{TOP} \leq P_c \leq P_{VKP}$ (Fig.2b), phase 3; post-filling phase at $P_c > P_{VKP}$ (Fig.2c). Therefore, each phase can be distinguished by the clutch pressure since the pressure can be directly measured by a pressure sensor. The displacement of the piston can also be an index of phase change, but it is omitted since it cannot generally be measured in production vehicles.

In order to simplify the model, a physical phenomena-based analysis is conducted. In the pre-filling phase and the post-filling phase, there are no displacement changes of the piston, and hence the volume of the clutch chamber is fixed. Thus, the value of the pressure gradient is large due to the bulk modulus

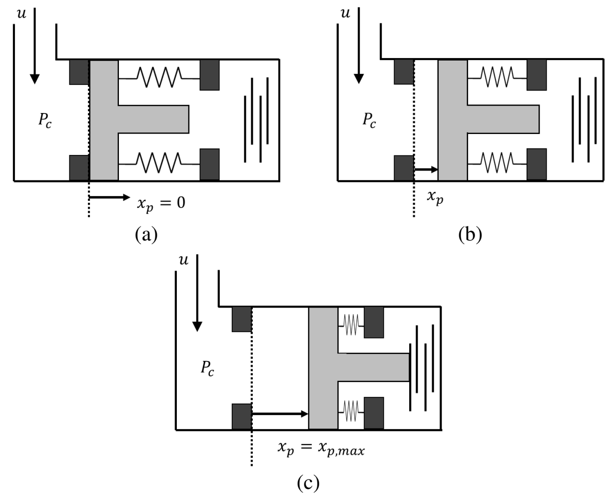


Fig. 2: Schematic diagram of clutch actuation process: (a) Pre-filling phase, (b) Filling phase, (c): Post-filling phase

in the pressure dynamics. This allows phase 1 and phase 3 to be represented by simple 1st order dynamics.

$$P_c = \frac{1}{\tau s + 1} u \tag{2}$$

where τ and u represent the time constant of the system and control input, respectively. These parameters should be obtained using experimental data.

In phase 2, the filling phase, the volume change of the clutch chamber cause a change in the pressure gradient. To express the dynamic behavior of the piston, the second equation of (1) can be used. In order to construct a simplified model, it is assumed that friction is negligible and the flow through the clutch chamber is a quasi-static flow, which means inertia force and damping force are negligible. Then, the resulting force equation of motion can be formed as follows:

$$A_c P_c = F_{fpre} + k x_p \tag{3}$$

The system of interest uses a high pressure range as the EHV equipped in the wDCT, and therefore the above assumptions can be made. In general, P_{TOP} and P_{VKP} can

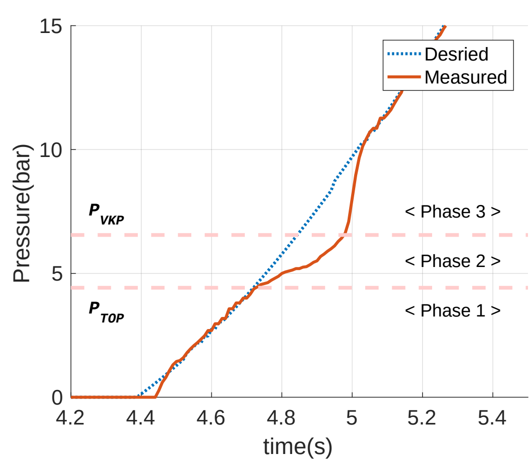


Fig. 3: Open loop response of the clutch actuation system

be identified through preliminary experiments, as shown in Fig.??. Using these values, other system parameter values can be obtained. P_{TOP} is the starting point of the filling phase, and therefore the displacement of the piston is zero; i.e. $x_p = 0$. P_{VKP} is the end point of the filling phase, and thus the displacement of the piston is the contact point, i.e. $x_p = x_{p,max}$. Therefore, the following relationships can be derived with (3).

$$P_{TOP} = \frac{1}{A_c} F_{pre} \tag{4}$$

$$P_{VKP} = \frac{1}{A_c}(F_{pre} + kx_{p,max}) \tag{5}$$

In general, the design parameters A_c and x_p are known values. Using the information of P_{TOP} and P_{VKP}, other parameters k and F_{pre} can be calculated directly. Thus, the equation of the piston can be comprehensively identified.

The pressure change inside the clutch chamber follows the pressure dynamic equation. In general, the pressure dynamics is expressed as a function of the flow in control volume. The well-known equation for the flow rate Q through the orifice is given by:

$$\Delta P = u - P_c \tag{6}$$

$$Q = C_d A_{ori} \sqrt{\frac{2}{\rho}\Delta P} \tag{7}$$

where A_{ori} denotes the area of the orifice. Equation (7) comes from an assumption of turbulent flow when the flow passes through the orifice. As a result, the flow rate Q is proportional to the $\sqrt{\Delta P}$. However, the clutch chamber geometrically resembles a sudden expansion rather than an orifice type. In this case, laminar flow dominates because streamlines are formed rather than the turbulent flow assumed in the orifice equation (7). The flow equation through the pipeline in a laminar flow is known as:

$$Q = \frac{\pi r^4}{8\mu L}\Delta P \tag{8}$$

where μ, L, and r represent the fluid viscosity, pipe length, and pipe radius, respectively. In the case of the clutch chamber, laminar flow and turbulent flow effect coexist after passing through the pipeline. Thus, the flow equation passing through the clutch chamber can be expressed as a linear sum of the effects of laminar flow and turbulent flow as follows.

$$Q = K_1 \Delta P + K_2 \sqrt{\Delta P} \tag{9}$$

Here, K_1 and K_2 are coefficients that denote the influence of laminar flow and turbulent flow, respectively. These tuning parameters need to be tuned based on experimental data. The volume of fluid entering the clutch can be calculated by integrating the flow rate using (9). Also, displacement of the clutch piston can be calculated by dividing the volume of fluid by A_c.

$$x_p = \begin{cases} \frac{1}{A_c}\int Q dt & \text{at filling} \\ x_{p,max} - \frac{1}{A_c}\int Q dt & \text{at dumping} \end{cases} \tag{10}$$

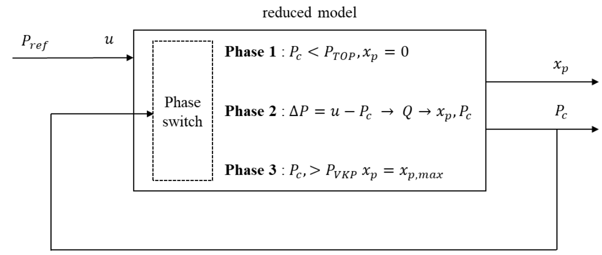


Fig. 4: Overall scheme of control-oriented model for clutch actuation system

The Dumping case refers to the case of a negative flow rate. Due to the return spring effect, the flow produced by the same pressure difference is larger than that during filling. Thus, the flow coefficients defined in (9) should be differed from the filling case. In this paper, the notation $K_{i,d}$ denotes the flow coefficients of dumping. For convenience and simplicity, $K_{i,d}$ is defined as a multiple of K_i.

$$K_{i,d} = \alpha K_i, i = 1, 2 \tag{11}$$

where α is a tuning parameter that should be tuned using experimental data.

Consequently, the dynamic behavior of the filling phase can be described by (3), (6), (9), and (10). The proposed reduced model has some advantages over (1). First, by using only three tuning parameters, one can reduce the effort of parameter tuning and effect on parameter uncertainties. In fact, design parameters A_p and $x_{p,max}$ are treated as known values, but in practice there are errors due to part-to-part variations or other uncertainties. In this work, it is assumed that the P_{TOP} and the P_{VKP} are known values through preliminary experiments. By using these system parameters, the model can be totally defined by selecting appropriate K_1 and K_2 values. The proposed model can compensate the parameter errors by tuning the appropriate K_1 and K_2 values. Second, the reduced model removes numerical instability by simplifying the stiff pressure dynamics. This increases the applicability of real-time control to production vehicles.

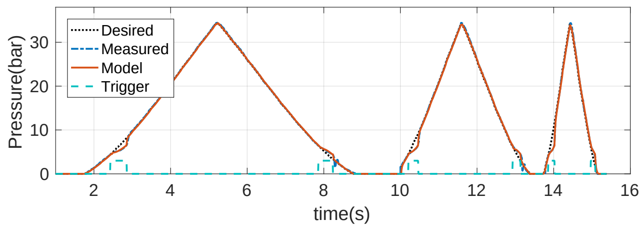
As a result, this section proposes a simplified model based on the physical phenomena of the filling phase. The model value is calculated according to the phase from the desired pressure value generated by the TCU. Each phase is determined by the calculated clutch pressure, P_c. The command input and calculated model pressure are used to update the clutch pressure and phase transition. Overall schematic of the reduced model is provided in Fig.4

C. Model Validation

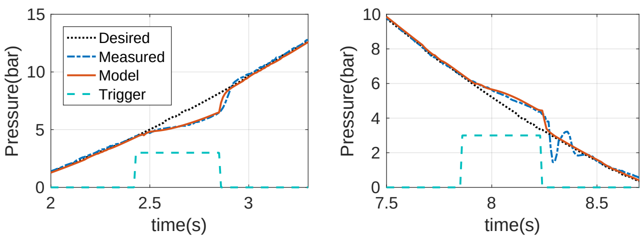
In order to validate the effectiveness of the reduced model proposed in the previous section, experiments were conducted using the EHV in the test vehicle. A detailed description of the test vehicle will be given in the results section. Since the main purpose of this section is to validate the reduced order model, experimental results and calculated model outputs are compared with same input scenario. For the scenario a ramp-up and ramp-down situation that characterizes the filling phase was selected. The tests were conducted with ramps of three

TABLE I: SYSTEM PARAMETERS

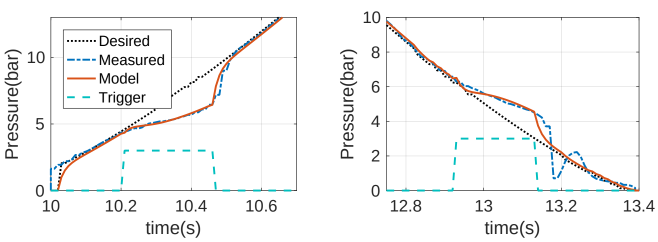
Parameter	Value	Parameter	Value
A_c	0.0113 (m^2)	$x_{p,max}$	0.003 (m)
P_{TOP}	4.42 (bar)	P_{VKP}	6.55 (bar)
K_1	5.5e-3	K_2	1e-3
α	5	T_s	0.01 (sec)



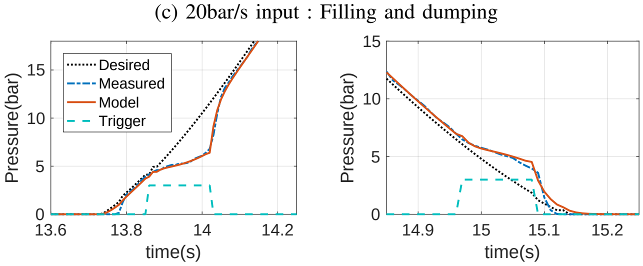
(a) Ramp input of various slopes



(b) 10bar/s input : Filling and dumping



(c) 20bar/s input : Filling and dumping



(d) 40bar/s input : Filling and dumping

Fig. 5: Model validation : (a) Pressure output of various slope input, (b) 10bar/s, (c) 20bar/s, (d) 40bar/s

different gradients. The system parameters used to calculate the model values are shown in Table I. As shown in Table I, the proposed reduced model is expressed by seven parameters. Two of them are known values (A_c, $x_{p,max}$), Two of them are measured form the preliminary experiment (P_{TOP}, P_{VKP}), and three of them are tuning parameters (K_1, K_2, α). These parameters are used to generate the model value. To verify the applicability to a production vehicle, the model calculation was conducted using the same sampling frequency as the production type TCU (100Hz). The validation results are depicted in Fig.5.

Fig.5a illustrates the full data of the various slopes of ramp-up and ramp-down input. It is shown that the actual measured pressure does not follow the desired pressure due to the filling and the dumping phase. Figs.5b, 5c, and 5d show enlarged views of phase 2 for each slope, which is the main focus of this paper. It can be seen that the model values describe the actual measured values well in all scenarios. In particular, unlike (1), it shows good results even at the sampling frequency of 100Hz, which can be applied to production vehicles. But, in the dumping phase described in Figs.5b, 5c, it can be seen that the reduced model deos not represent the transient behavior of the clutch. This is a limitation of the reduced order model, which does not represent the dynamic behavior of the clutch piston and the return spring in phase transition. However, for the pressure below P_{TOP}, the transmitted torque is zero and not the control range, so this paper focuses only on the filling phase.

The proposed model expresses the system behavior in the filling phase as the sum of the laminar flow and turbulent flow with K_1 and K_2, as described in (9). In practice, describing model values with only one of K_1 or K_2 can be tuned for a single slope, but not for another slope input. The results in Fig.5 are calculated for fixed K_1, K_2, and α. Therefore, it can be verified that the proposed reduced model closely describes the behavior of the the filling phase.

III. Controller Design

A. Model-based controller design

In this section, we design a controller based on the reduced model proposed in the previous section. Unlike Automatic Transmission (AT), DCT does not have a torque converter, and thus precise shift control is required. For example, slip control is indispensable for launch control and low speed driving with DCT due to the design characteristics. In addition, for fast shifting, the off-going clutch is positioned to stay in the filling phase where torque is not transmitted. These behaviors are determined by various shift control strategies, and the clutch actuation system requires accurate tracking performance in the filling phase in order to implement the developed modern shifting control logic.

In general, the simplest method of tracking control is feedback based proportional-integral-derivative (PID) control. The PID control is very simple and easy to implement and accordingly is used in many industry fields. However, there is a practical issue to implement a general PID controller for the clutch fill process, which is a time delay. Time delay is one of the well-known nonlinear behaviors of the EHV system. This time delay is actuation delay, which adversely affects the control for nondeterministic inputs such as shift control. The output of the system with actuation delay is generated according to the following equation:

$$P_m(t) = P_d(t - T_d) \tag{12}$$

where P_m , P_d, and T_d are the measured output, desired input, and delayed time, respectively. In an automotive control system, the time delay is known as varying parameter within a boundary [29]. The delay time of the plant is about 30ms. However, as shown in Fig.5, the filling time has a short duration of about 100 to 200ms according to the desired input gradient. The dead time of 30ms during the short duration filling time has a strong influence on the control and limits the control margin. A Smith Predictor (SP) is a well-known control technique of a time delay system. However, SP a the controller based on accurate delay time, and error of delay time increases the instability of the closed loop system [30]. Generally, convergence time and system stability are in a trade-off relationship. Therefore, it is difficult to obtain the control performance by using the conventional SP and PID control in the filling phase.

In this paper, we propose a feed-forward based controller that minimizes the influence of feedback including delayed information. A feed-forward control has the advantage of ensuring fast control input generation and stable input if the model information is accurate. In addition, since the feed-forward does not have delay information, it is not necessary to consider the effects of delay in the controller design.

In order to design a controller to track a desired pressure in the filling phase, a control-oriented model using the proposed reduced model is designed as follows:

$$P_c = \frac{1}{A_c}(F_{pre} + kx_p) \tag{13}$$

$$\dot{x}_p = \frac{1}{A_c}(K_1(u - P_c) + K_2\sqrt{u - P_c}) \tag{14}$$

The system can be expressed as pressure dynamics by combining (13) and (14) using the variable x_p.

$$\begin{aligned}\dot{P}_c &= \frac{k}{A_c^2}(K_1(u - P_c) + K_2\sqrt{u - P_c}) \\ &= a_1(u - P_c) + a_2\sqrt{u - P_c}\end{aligned} \tag{15}$$

Here, a_1 and a_2 are defined as $a_1 \triangleq \frac{k}{A_c^2}K_1$, and $a_2 \triangleq \frac{k}{A_c^2}K_2$, which correspond to the equivalent coefficient of laminar and turbulent flow. Also, flow coefficients for dumping also can be defined as $a_{i,d} = \alpha a_i$ where $i = 1, 2$. The a_1 and a_2 values can be easily calculated by known physical properties, which are given in Table I. Thus, a control-oriented model for pressure control can be constructed. Note that a_1 and a_2 are lumped system parameters, so adjusting the parameter values can handle part-to-part variation of system parameters such as k and F_{pre}.

The purpose of the feed-forward control is to calculate the control input so that the output pressure follows the desired pressure input. To calculate the appropriate control input, the desired pressure P_d is substituted into (15) instead of the actual pressure P_c.

$$\dot{P}_d = a_1(u - P_d) + a_2\sqrt{u - P_d} \tag{16}$$

The above equation predicts the behavior of the model. Thus, feed-forward input can be obtained by solving the equation for control input u to track the desired pressure. The results are calculated as follows:

$$\begin{aligned}u &= \left(\frac{-a_2 + \sqrt{a_2^2 + 4a_1\dot{P}_d}}{2a_1}\right)^2 + P_d \\ &= u_{fill} + P_d \qquad \text{for filling,} \\ &= -\left(\frac{-a_{2,d} + \sqrt{a_{2,d}^2 + 4a_{1,d}\dot{P}_d}}{2a_{1,d}}\right)^2 + P_d \\ &= u_{dump} + P_d \qquad \text{for dumping}\end{aligned} \tag{17}$$

It can be confirmed that feed-forward input u is composed of the desired pressure and an additional term for compensation of the filling phase. The compensation term consists of the flow coefficients a_1 and a_2 and the slope of the desired pressure. As a result, a feed-forward controller that contains information of the model is designed. Thus, we can design a control-oriented model based feed-forward controller that reflects the model behavior.

B. Adaptive law design

The controller for the filling phase designed in the previous section is based on the control-oriented model (15). The most important model parameters in controller (17) are a_1 and a_2, which represent the amount of flow. In a physical sense, higher a_1 and a_2 values represent more flow at the same pressure difference. Thus, large flow coefficients generate a small compensation term ,u_{fill}, while small flow coefficients generate a large u_{fill}. In a real plant, the above parameters may not be fixed values in a driving situation. The parameters of EHV systems can vary due to oil temperature changes, line pressure changes, and other uncertainties. The feed-forward controller generates fast and stable inputs, but has the disadvantage of degrading performance against plant changes, model errors, and external disturbances. Therefore, real-time updates of the model are required to consider the model changes in various situations. In this section, a parameter adaptive law that compensates model error due to various factors is proposed.

The actual plant behavior can be described using the control-oriented model (15). Here, the predicted behavior of the plant by the model and the desired input is defined as follows:

$$\dot{P}_d = \hat{a}_1(u - P_d) + \hat{a}_2\sqrt{u - P_d} \tag{18}$$

where $\hat{a}_1$ and $\hat{a}_2$ denote estimated values of a_1 and a_2. In fact, (15) has two system parameters. However, the system of interest has only one measurement signal, P_c. In order to estimate two parameters, the order of Persistence of Excitation (PE) should be more than 2 [31], [32]. Since the input signal of the filling phase mostly consists of a ramp input, the control input is generated in the form of a step input according to (17). The PE order of the step input is 1, and thus it is difficult to estimate the two parameters in the filling phase based on the model (15). Thus, an additional assumption is needed to design an adaptive law. In this study, we assume that parameter a_1 is more dominant than a_2 based on the experimental data (see Table I). Therefore, a_2 is assumed to be a fixed value and only a_1 is updated; i.e. $\hat{a}_2 = a_2$. This assumption is verified through experiments in the results section. The purpose of the parameter adaptation is to converge parameter error and control error to zero simultaneously. The corresponding adaptive law for parameter $\hat{a}_1$ is then defined as follows:

$$s \triangleq P_c(t) - P_d(t - T_d) \tag{19}$$

$$\dot{\hat{a}}_1 = \gamma(u - P_d)s = \gamma u_{fill} s \tag{20}$$

where γ denotes the adaptive gain that determines the adaptation rate. It can be confirmed that the adaptive law updates $\hat{a}_1$ in real time based on the feed-forward input u_{fill} and the control error s. As described in the previous section, the EHV system has an actuation time delay. Obviously, the best control result for a system with actuation delay for unknown inputs is achieved by tracking the reference with delayed time. The control error can the be defined as (19). By combining (17) and (20), an adaptive feed-forward controller is designed.

In order to prove the stability of the proposed adaptive feed-forward controller, a Lyapunov candidate function is defined as

$$\tilde{a}_1 \triangleq \hat{a}_1 - a_1 \tag{21}$$

$$V = \frac{1}{2}s^2 + \frac{1}{2\gamma}\tilde{a}_1^2 \tag{22}$$

since the purpose of the proposed controller is to make $s \to 0$ and $\tilde{a}_1 \to 0$ as $t \to \infty$. The derivative of (22) with respect to time is described in (23) assuming the parameter is slowly varying, i.e. $\dot{a}_1 \simeq 0$.

$$\begin{aligned}\dot{V} &= s\dot{s} + \frac{1}{\gamma}\tilde{a}_1\dot{\hat{a}}_1 \\ &= -a_1 s^2 + a_2 s(\sqrt{u - (s + P_d)} - \sqrt{u - P_d})\end{aligned} \tag{23}$$

Note that the final term of (23), $a_2 s(\sqrt{u - (s + P_d)} - \sqrt{u - P_d}) < 0 \quad \forall s$ since $a_2 > 0$, and designed adaptive law (20) is used to derive (23). By selecting appropriate $\gamma > 0$, (23) can be concluded as follows:

$$\dot{V} < 0 \quad \forall s \neq 0 \tag{24}$$

By the Lyapunov stability theorem, s and $\tilde{a}_1$ converge to zero as $t \to \infty$ [33]. Note that in the case of $s = 0$, tracking performance is satisfied and the adaptive law (20) becomes zero such that $\hat{a}_1$ is not updated; i.e. $\dot{\hat{a}}_1 = 0$.

C. Determination of filling time

Another important technique in filling phase control is the determination of the filling time. In section II, measured P_{TOP} and P_{VKP} values are used to calculate the model values. But it is not easy to define the exact P_{VKP}, as shown in Fig.2. The approximation of P_{VKP} in the modeling process is acceptable, but can have a severe adverse effect in the control process. Therefore, a trigger for end of the filling phase should be obtained based on a physical quantity that is not changed sensitively and not mainly affects the results. The fixed physical quantity in a clutch actuation system is the piston maximum displacement, i.e. the volume of the clutch chamber. Therefore, we propose a novel indicator of status of the filling phase based on the volume of fluid entering the clutch chamber. The filling indicator can be used as an index to identify how much filling has been performed in the filling process. The volume of the fluid entering the clutch chamber can be calculated using (9) and (10).

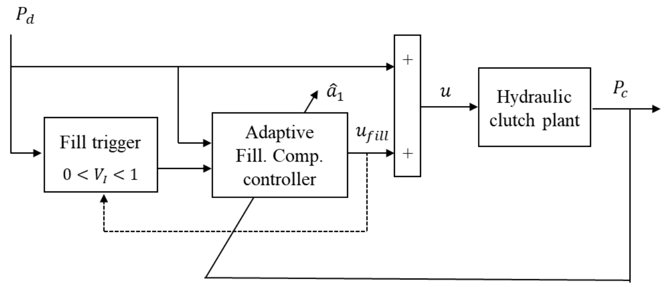


Fig. 6: The adaptive feed-forward controller scheme.

$$V = A_c x_p = \int (K_1(u - P_c) + K_2\sqrt{u - P_c})dt \tag{25}$$

If the pressure is well tracked by using the proposed controller, i.e. $P_d \simeq P_c$, the corresponding control input can be approximated as follows:

$$u = P_d + u_{fill} \simeq P_c + u_{fill} \tag{26}$$

Therefore, a volume indicator can be defined as follows:

$$\begin{aligned}V_I &= \frac{V}{V_t} \\ &= \frac{1}{V_t}\int (K_1 u_{fill} + K_2\sqrt{u_{fill}})dt\end{aligned} \tag{27}$$

where V_t is the maximum volume of the clutch chamber, and u_{fill} refers to (17). In a dumping case, K_1, K_2, u_{fill} are substituted to the form of dumping. The calculated fluid volume $V \in [0, V_t]$, i.e. $V_I \in [0, 1]$. The filling phase control ends when the value of the volume indicator V_I becomes 1. The value of V_t can be obtained by physical properties. By using V_I, accurate filling control can be performed for various desired pressure trajectories in the filling phase, which is required in wDCT. Note that V_I is defined with only the desired input information and the model parameters without feedback information. The overall scheme of the proposed control strategy is illustrated in Fig.6

IV. Simulation Results

A. Simulation overview

In order to verify the effectiveness of the proposed controller, simulations were conducted. The plant was modeled using AMESim, which contains detailed dynamics of the hydraulic system so that can be treated as the exact model of the plant. The EHV was described using the VFS and

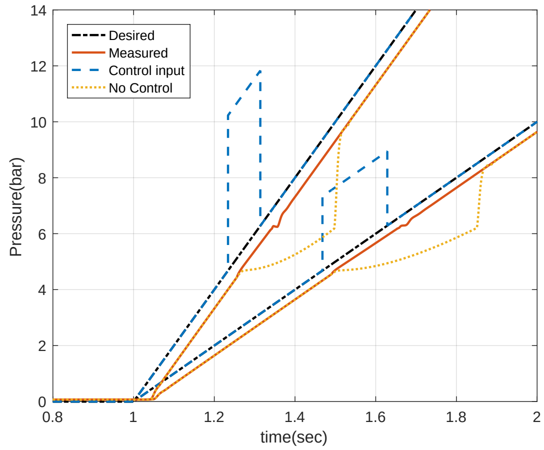
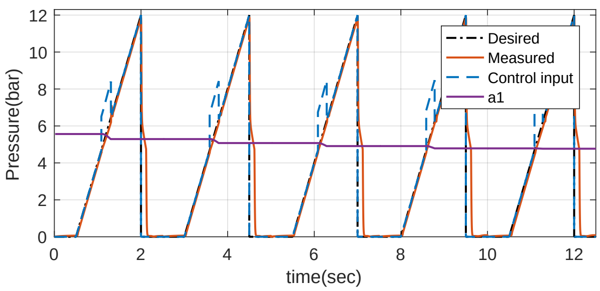
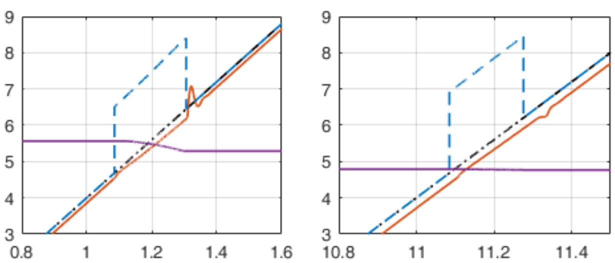


Fig. 7: Simulation results: feed-forward control with various slopes



(a) Adaptive feed-forward control results



(b) Enlarged view

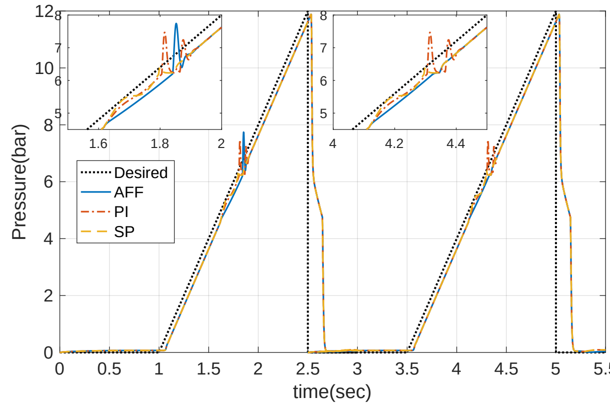
Fig. 8: Simulation results: Adaptive feed-forward control

clutch chamber as shown in Fig.1. Moreover, practical issues such as system slew rate and time delay are also included in the simulation model. The proposed control logic was implemented using Matlab Simulink, and its performance was verified by co-simulation with the constructed AMESim plant. The system parameters used in the simulation use the actual plant values obtained based on the experimental data.

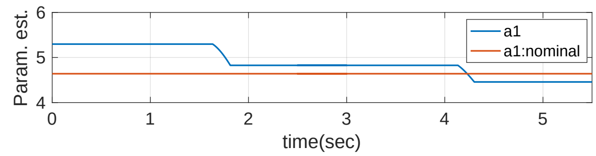
B. Results

The simulation scenario takes the ramp-up input, which is the general pressure trajectory of a torque phase. The slope of the ramp is determined by the driver's accelerator pedal input and the corresponding shift strategy. Simulations were carried out for two cases at 10bar/s and 20bar/s pressure input. The results using the proposed feed-forward controller are shown in Fig.7

As shown in Fig.7, it can be confirmed that the proposed feed-forward controller compensate stably for ramp input on various slopes. The corresponding control outputs track the



(a) Simulation results



(b) AFF : Adaptive parameter

Fig. 9: Simulation results: Comparison between AFF, PI, and SP controller

desired pressure with a time delay. The generated control inputs clearly demonstrate the advantage of fast input generation and stability, which are the advantages of feed-forward. As described in (17), u_{fill} contains the information of the model and the reference input, so it generates the appropriate control input according to the desired input slope. Further, since the filling control is triggered using the volume indicator, control is performed such that the constant volume of fluid is applied to the clutch chamber. Therefore, large control inputs are applied for a short time at a large slope input, and small control inputs are applied for a longer time at a small slope input. In this way, it can be confirmed that good control performance in the filling phase is obtained by a feed-forward controller based on desired input and model information.

Simulations were also conducted to verify the effectiveness of the adaptive law to cope with the model change, which have an adverse effect on the feed-forward controller. The simulation scenario repeats ramp-up of the same slope. In order to see the effect of the adaptive law for parameter a_1, the control was conducted using an initial value of a_1 with an arbitrary error. The adaptive gain was set to $\gamma = 5$, and the results are depicted in Fig.8.

As shown in Fig.8, it can be confirmed that the adaptation of the parameter is performed well. The parameter a_1 is updated at each filling phase and converged to the nominal value. At the beginning of the simulation, a large value of a_1 is applied to the controller, resulting in generation of small control input and under-fill pressure output. However, as the adaptation progresses, the control error converges to zero as the parameter converges to its nominal value. As described in the stability analysis of the adaptive law, (20) make both the parameter error and the control error converge to zero. The simulation results reveal the convergence of errors.

To verify the effectiveness of the proposed controller, we compare the results with Proportional-Integral (PI) control and Smith Predictor (SP), which are conventional control methods of time delay systems. The SP controller is a well-known delay compensator and its detailed description is omitted in this paper. The time delay of the simulation model is set to 30ms, but the actual delayed time of vehicle plant changes from 10 to 30ms. Therefore, the delayed time used in the controller is set to 20ms to reflect this variation. Also, model parameters a_1 and a_2 values are set by adding 15% of the nominal values. The results are illustrated in Fig.9.

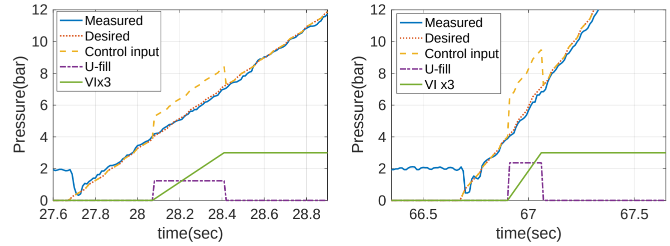
In the case of the proposed controller, which is denoted as adaptive feed-forward (AFF), control error due to the parameter error appeared in the initial stage, but it can be confirmed that stable and good tracking results are obtained due to the effect of the adaptive law. The parameter a_1 does not converge to the real value due to the error of the time delay, but it shows acceptable parameter adaptation and control results. On the other hand, control error and over-fill occur in PI control due to the delay. In the delay system, the PI control is strongly influenced by the integral term, and thus it is difficult to guarantee a fast convergence. Thus, it is difficult to utilize PI control in the short duration of the filling phase. In the case of SP, since the delay time and the model parameters used in the plant model and controller are perturbed from real values, the controller shows unstable tracking performance. The method of SP is sensitive to the accuracy of delayed time and model information. These results reveal that it is difficult to utilize the existing control method of time delay systems in actual plants that contain varying delayed time. Therefore, the proposed controller is an effective control method considering the practical issue of the plant. Since the conventional control methods are difficult to ensure stable control performance in various situations, there is a concern about the durability of hardware and instability of shift control due to erroneous control in an actual vehicle test. Therefore, only the most stable controller, the AFF, is applied to the test vehicle and experiments are conducted to verify its effectiveness. Experimental verifications are described in the subsequent section.

V. Experimental Results

A. Experiment overview

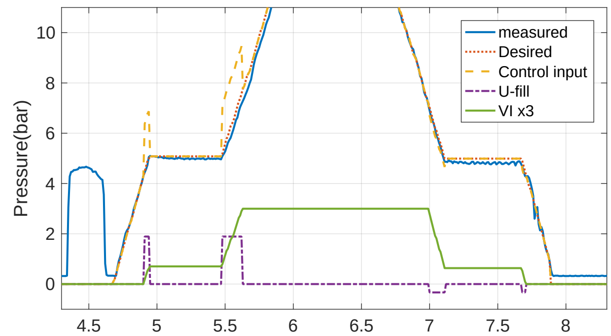
In addition to the simulation, some experiments were carried out to validate the effectiveness of the proposed controller. Experiments were conducted in a test vehicle equipped with a wDCT actuated by an EHV. The EHV used in the experiment measures the pressure inside the clutch chamber through a

Fig. 10: Experimental test vehicle

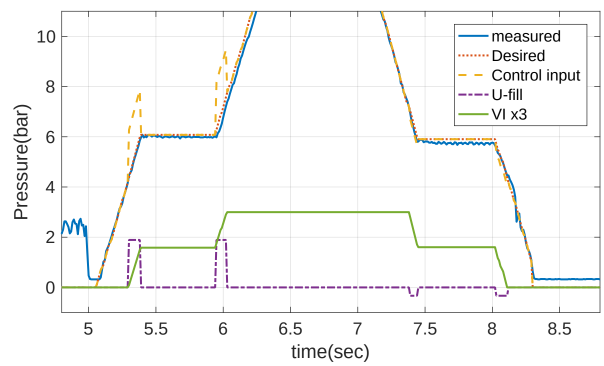


(a) Ramp up: 10bar/s (b) Ramp up: 20bar/s

Fig. 11: Experimental results: feed-forward control with various slopes



(a) Stop at 5bar



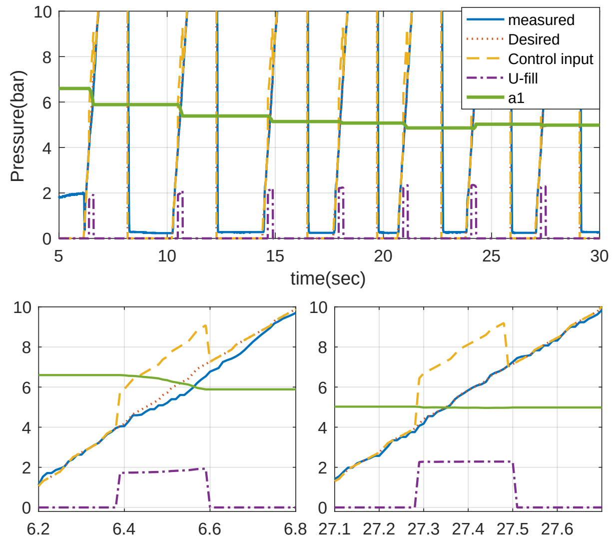
(b) Stop at 6bar

Fig. 12: Experimental results: Stop in filling region during ramp up/down

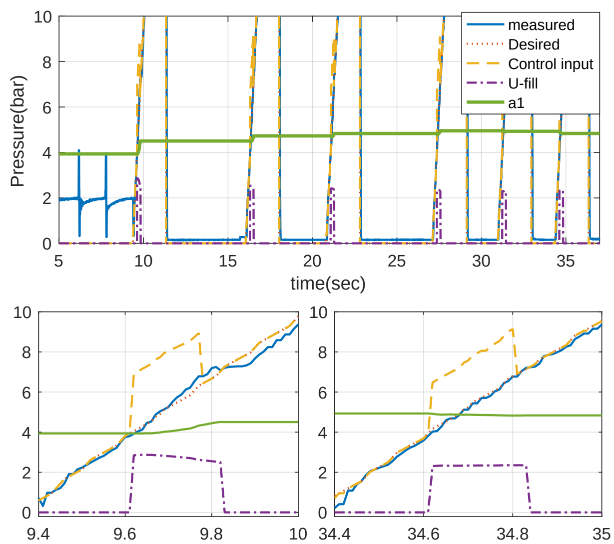
pressure sensor. In this work, it is assumed that the current control is controlled at a sufficient high speed as compared with the proposed control logic. Also, the relationship between current and pressure uses the relationship identified in previous studies. Thus, the system generates the valve output pressure immediately according to the desired pressure based on identified internal logic. The developed control logic was embedded in the production type TCU, and the data transmission and reception is obtained using the vehicle CAN network. The sampling frequency of data logging and logic implementation was set to 100Hz. The system of interest has time delay as described in (12). The best control output of a time delay system with unknown input is an output that tracks a reference profile with a delayed time. Therefore, calculation of control error and performance is obtained using the delayed reference input.

B. Results

First, some experiments were conducted to verify the performance of the model-based feed-forward controller designed in section III-A. For the experimental scenarios various slopes



(a) Larger initial value and enlarged view



(b) Smaller initial value and enlarged view

Fig. 13: Experimental results: Adaptive feed-forward control

of ramp-up inputs were selected similar to the simulation scenarios. The main model parameters were obtained from the experimental data as shown in section III, and the corresponding experimental results are illustrated in Fig.11.

As shown in the Fig.11, it can be seen that the results are similar to the simulation results. The filling phase is well compensated regardless of the input slope variation. Since the control strategy is to apply the same volume of fluid to the clutch chamber, it can be seen that a large input is applied for a short time at a large slope input and a small input is applied for a long time at a small slope input. Thus, u_{fill} is generated according to the input slope, and V_I is calculated based on u_{fill} to generate the filling trigger.

By using the volume indicator V_I, information of the current filling status can be obtained. V_I provides information about how much fluid is in the clutch chamber and how much more fluid should be applied to end the filling phase. This is the essential information for wDCT where frequent slip and low-pressure range control occur. Some additional experiments were conducted to validate the effectiveness of the volume indicator. The scenarios were selected to stop between P_{TOP} and P_{VKP} during the ramp up and down input. In order to obtain good filling compensation without over-fill or under-fill, information on the exact status of filling is needed. The experimental results are depicted in Fig.12

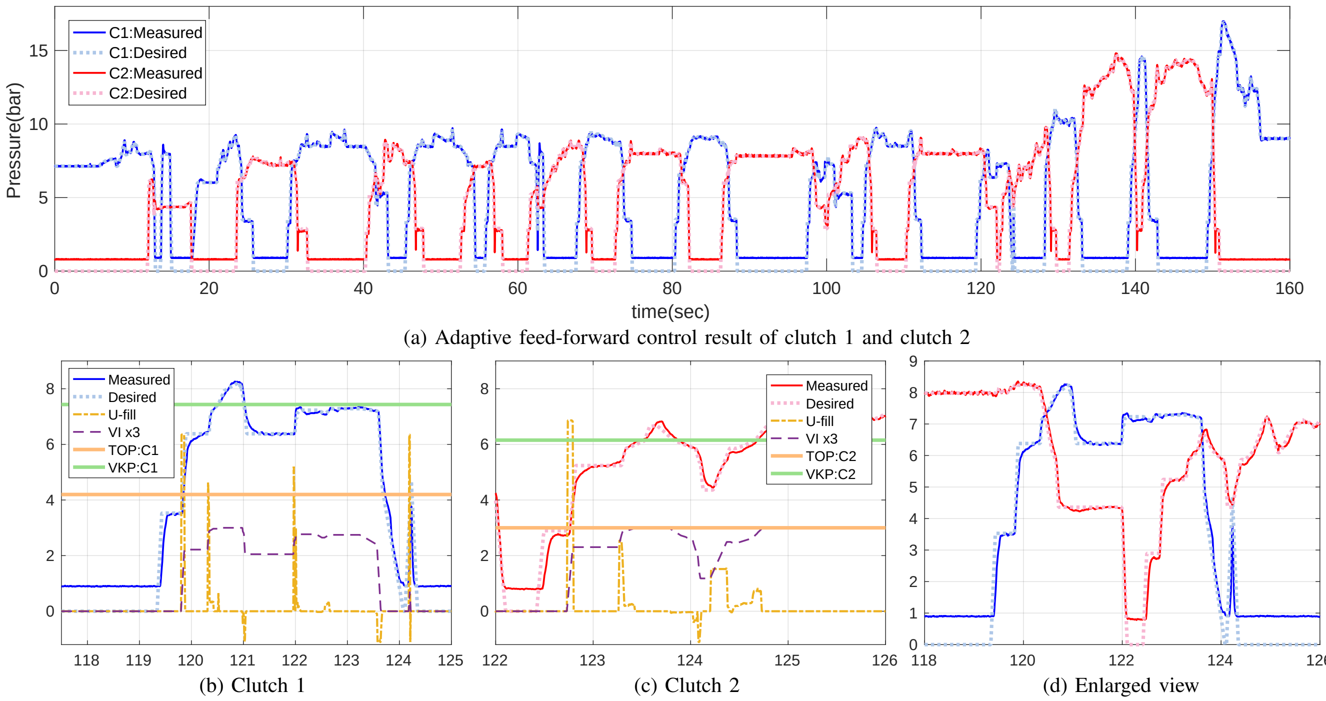
The experimental results in Fig.12 show that the filling state can be determined by calculating V_I based on the feed-forward input. This information implies the possibility of control in the filling phase range such as slip and fast launch control. Therefore, it is possible to utilize various control strategies in wDCT using control of the filling phase.

In order to verify the adaptive feed-forward controller, experiments using the perturbed model were performed. Similar to the simulation, the scenario repeats the same slope of ramp-up input using an inaccurate initial value of a_1 and a_2. In order to confirm the convergence of the parameters, the experiments were conducted in two cases with parameter values larger than and smaller than the nominal value. The results are depicted in Fig.13.

As shown in Fig.13, it can be verified that the parameter adaptation and the corresponding pressure control in the filling phase are performed well. At the beginning of the experiment, under-fill or over-fill occurs due to the error of model parameters. However, as adaptation progresses, the control error converges to zero and the parameter also converges to its nominal value, as described in adaptive law (20). Therefore, it can be confirmed that both the control error and the parameter error converge to zero simultaneously. The adaptive gain γ was set to 5. Larger adaptive gain ensures a faster convergence rate, but can cause system instability. Thus, appropriate adaptive gain should be tuned to obtain a fast and stable control process. The proposed adaptive logic can be used as an auto-tuning algorithm to find the initial value of the model parameter in production vehicles. It is expected that it will be able to cope with the part-to-part variations of model parameters by automatically finding the nominal values. In addition, it can counteract the parameter variation due to the disturbances in driving situations. The performance and effectiveness of the proposed control logic thus can be verified experimentally.

The driving test was performed to verify the performance of the proposed controller in actual driving situations. The driving test includes scenarios such as acceleration, deceleration and up/down shifts in manual mode. This driving test includes various uncertainties and disturbances that can occur in driving situations. The results are shown in Fig.14

The driving test includes many uncertainties and disturbances such as change of line pressure, variation of oil temperature, unmodeled dynamics, shift shocks, and so on. In spite of these uncertainties, it can be seen that the proposed controller provides good tracking performance in the filling phase for various situations. Fig.14b, and 14c shows the results of starting the torque phase after pre-fill in the filling phase. Although there exist pressure trajectory between P_{TOP} and P_{VKP}, it can be confirmed that the output pressure tracks the desired input without under-fill or over-fill since the volume indicator provides the exact filling status. Also, when the



(a) Adaptive feed-forward control result of clutch 1 and clutch 2

(b) Clutch 1 (c) Clutch 2 (d) Enlarged view

Fig. 14: Experiment results: Driving test

shift process is performed during driving, both clutch 1 and clutch 2 show good control results, as shown in Fig.14d. The root mean square error (RMSE) of the filling phase in the driving test is calculated as 0.2116bar and 0.1720bar in clutch 1 and 2, respectively. Thus, the proposed controller ensures the control performance in the filling phase for various situations. Moreover, by defining the filling index through the proposed V_I, it is possible to reliably respond to various control strategies in wDCT.

VI. Conclusion

This study has proposed a novel control strategy of clutch fill control in wDCT. The proposed control logic is designed considering practical issues at the production vehicle level, and thus ensures good pressure tracking performance in the filling phase. The main contribution of this work is twofold: a control-oriented model describing the filling phase and an adaptive feed-forward controller based on a volume indicator. The reduced-order control oriented model of the filling phase increases applicability to production vehicles by reducing the numerical instability and the number of unknown parameters. The adaptive feed-forward controller has the advantage of minimizing the effect of time delay, which has a strong influence on the control of the short filling phase. The controller guarantees good control performance regardless of the uncertainties and disturbances in the clutch shift process by using parameter adaptation logic. Moreover, the volume indicator based on the feed-forward input provides the exact status of filling and generates an accurate filling trigger to achieve good control results. Furthermore, the proposed strategy has a shorter pre-fill process compared to the conventional methods, so it can respond quickly to various shifting situations. These original contributions are verified through both simulations and test vehicle experiments. The proposed work is expected to improve shift performance in production vehicles using wDCT and to cope with more various advanced shift strategies.

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