2019 American Control Conference (ACC)
Philadelphia, PA, USA, July 10-12, 2019

Hydraulic Clutch Fill Control using Control-oriented Model in Wet Dual Clutch Transmission

Sanghun Jung[1], Jinsung Kim[2], Hoyoung Lee[2], Youngho Ko[2], and Seibum B. Choi[1*]

Abstract— **This paper proposes a control method for filling phase of a hydraulic clutch actuation system in wet dual-clutch transmission (DCT) based on a control-oriented model. In wet DCT, a clutch-fill process cannot obtain proper fill and maintain under-fill due to the drag torque, which causes bad shift performance. In this paper, in order to compensate the filling phase, a control-oriented model considering practical issues is proposed. The proposed model is used to design a model-based controller to track a desired pressure in filling phase. The designed controller is composed of feed-forward part contains model information for fast response and convergence, and feedback part that compensates modelling errors or uncertainties. In order to verify the proposed controller, simulation based on an exact model of hydraulic clutch actuation system is constructed. Simulation results reveal that the proposed controller guarantees good pressure tracking performance in filling phase for various situations.**

I. INTRODUCTION

Recently, the importance of fuel economy is increasing in the field of automotive engineering. The dual clutch transmission (DCT) has emerged as a next generation powertrain system with high fuel efficiency on behalf of automatic transmission (AT) [1]. DCT can be classified into two types according to clutch type; dry DCT and wet DCT. In recent years, a number of vehicles equipped with dry DCT have been massively produced. However, dry DCT is not suitable for use in heavy-duty vehicles and high performance vehicles due to frictional heat and wear of friction plates. To complement this problem, research on wet DCT for high power vehicles is actively being carried out. In fact, wet DCT is difficult to control due to nonlinearity factors of hydraulic actuator and friction clutch. Therefore, precise torque and speed control of the transmission system is required for shift control of the wet DCT.

In the DCT shift process, the engine power is transferred through engaging and disengaging of the two clutches. To achieve fast and smooth shifting performance, the speed and torque of the on-coming clutch and the off-going clutch must be controlled precisely. DCT does not use the torque converter used in AT, so there is no smoothing effect and inaccurate shift control is directly related to the bad shift performance. For precise shift control, an on-coming clutch should be pre-positioned at the point where the torque begins to be transmitted, i.e. the clutch contact point. This process is called the clutch-fill process and is one of the most important technology in clutch-to-clutch shift control.

The electro-hydraulic valve (EHV) using variable force solenoid (VFS) is commonly used as a clutch actuator in many commercial vehicles due to its high energy efficiency and control convenience. The EHV has the advantage that the clutch pressure can be controlled in proportion to the input. An important state in shift process is the torque transmitted by each clutch. In the clutch engaging process, torque is not transmitted to the clutch until the clutch piston contacts the friction plate. Therefore, when EHV is used, a pre-fill control should be performed to pressurize the piston to the point before the clutch is in contact with the point to start the torque phase immediately [2]. If an under-fill or over-fill occurs due to improper pre-fill control, this will have an adverse effect on clutch shift performance immediately. Thus, it is important to control the proper clutch-fill process.

Previously, many studies have proposed the control strategy of the clutch-fill process [3]–[6]. There are hard nonlinearities and internal dynamics in the aformentioned clutch fill process. Considering these nonlinearities and dynamics, some works proposed control strategy to generate the proper filling. [4] proposed an optimized filling control strategy to minimize the energy consumption. [5] developed tracking controller to track the trajectory for smooth filling using dynamic programming. [6] analyzed the factors affecting the filling process and proposed a fuzzy control strategy. Like this, many previous studies have been conducted on how to perform proper clutch-fill, i.e. positioning the clutch piston at the touch point effectively.

However, it is difficult to apply a general clutch-fill logic for wet DCT, which is the main purpose in this paper. Since the wet clutch is used, the drag torque is generated even before the clutch contact point [7]. This causes the same effect as over-fill described above, which causes clutch tie-up and output torque reduction. Therefore, control of wet DCT will inevitably aim to under-fill in the clutch-fill process. When the torque phase is started in the under-fill situation in shifting process, a delayed pressure output occurs due to the filling phase and causes bad shift performance [8], [9]. In order to guarantee the good shift performance, it is essential to compensate the filling process in early stage of the torque phase.

In this paper, a control logic to compensate the filling phase in wet DCT is proposed. To simplify the logic, the control-oriented model is proposed. The proposed controller is constructed using the control-oriented model and obtain good pressure tracking performance in the filling phase.

[1]Sanghun Jung (shjung0225@kaist.ac.kr) and Seibum B. Choi (sbchoi@kaist.ac.kr) are with the Korea Advanced Institute of Science and Technology (KAIST), Daejeon, Republic of Korea.
[2]Jinsung Kim (jinsung.kim@hyundai.com), Hoyoung Lee (bear2501@hyundai.com), and Youngho Ko (YoungHo.Ko@hyundai.com) are with Hyundai Motor Company, Hwasung 18280, Republic of Korea.

978-1-5386-7926-5/$31.00 ©2019 AACC

In addition, it is necessary to consider the implementation aspect of the resulting controller, which allow for the use of a production type Transmission Control Unit(TCU). The proposed logic is expected to improve the shift performance equipped with wet DCT.

The remainder of this paper is organized as follow. In section II, simplified control-oriented model is described in detailed. The proposed model is verified using experiment results. In section III, a model-based controller is constructed using proposed control-oriented model. The validation of the controller is presented in section IV. Conclusion and future works are provided in section V.

II. SYSTEM MODELING

A. System description

A general hydraulic clutch actuation system is described in Fig. 1. The solenoid valve controls the output pressure P_A which is proportional to input current. In this paper, the valve is assumed to be a static system, and the valve output pressure P_A is defined as the control input u of the clutch chamber. Before shifting, the clutch piston stays at rest since it is a normally open clutch (Fig.1a). When a shift signal comes from a transmission control unit (TCU), the pressure of clutch chamber increases and the piston moves until it contacts the friction plate (Fig.1b). Through the clutch-fill process, torque is transmitted from the clutch so that the shift process is completed. A well-known dynamic model of the clutch-fill process can be derived as follows [10]:

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

where x_p is the piston displacement, 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.

The equation (1) is includes the piston and the hydraulic dynamics. However, (1) is difficult to apply to controller design since it has many unknown parameters with large uncertainties. In addition, since these equations are nonlinear, the analytic solution cannot be derived directly. Besides, the difference of dimension of each state in (1) is too large. For example, x_1 and x_2 are under 10^{-2}, but x_3 is over 10^5. This difference makes the model equation very stiff and numerically unstable to apply to the controller design. To construct a controller that can be applied in a real-time, a reduced model that is simpler than (1) is needed.

B. Control-oriented model

The clutch piston dynamics and hydraulic dynamics expressed in (1) are reconstructed considering the physical phenomena. For convenience and simplicity, the clutch-fill process is divided into three phases based on the position of the piston and the clutch chamber pressure. Phase 1 represents a pre-filling state, in which the piston remains stationary until the pressure overcomes the spring pre-load force. Phase 2 represents the filling phase, which is the main interest of this paper. In the filling phase, the piston moves to the contact point of the friction plate. Phase 3 starts after the piston contacts the plate. Intuitively, each phase can be easily distinguished by a piston pressure since the pressure can be directly measured by pressure sensor. Fig.2 depicts the clutch chamber pressure of real plant at each phase for open loop ramp up input command.

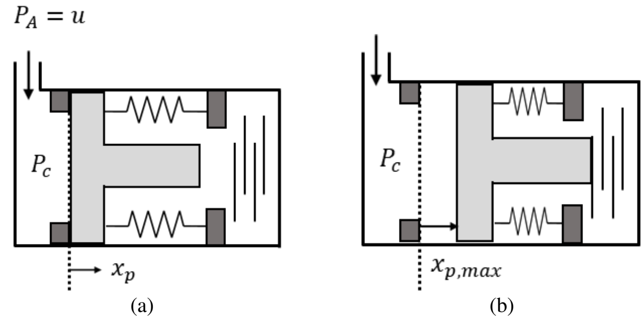


Fig. 1: Scheme of a clutch actuation system : (a) before filling, (b) after contact

The regions of pressure and piston displacement for each phase can be determined as follow:

$$\text{number of phase} = \begin{cases} 1 & \text{if} \quad P_c < P_L \\ 2 & \text{if} \quad P_L < P_c < P_H \\ 3 & \text{if} \quad P_c > P_H \end{cases} \tag{2}$$

where P_L and P_H are defined as initial and final pressure of filling phase, as depicted in Fig.2. 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 phase 1 and 3, the piston does not move and the

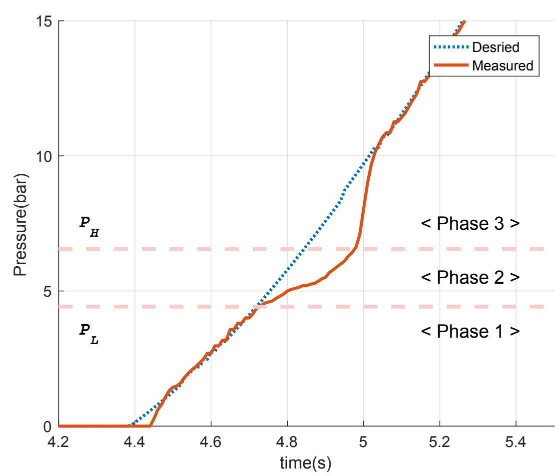


Fig. 2: Open loop response of a clutch actuation system

volume of the clutch chamber does not change, resulting in a very high pressure gradient as described in (1). This implies that pressure dynamics can be treated as a static system. Therefore, in phase 1 and 3, input and output relationship can be described as following 1st order system.

$$P_c = \frac{1}{\tau s + 1} u \tag{3}$$

where τ is time constant to be determined by experimental data.

In phase 2, the filling phase, the piston moves to the contact point and changes the volume of the clutch chamber. Because of volume expansion, pressure gradient decrease and cause the lagged response as shown in Fig.2. The equation of motion of piston in region 2 can be derived as the second equation of (1). By assuming the friction is negligible and pressurized fluid is quasi-static flow, the equation of motion of piston can be reduced as follows:

$$A_c P_c = F_{fric} + k x_p \tag{4}$$

where $\dot{x}_P$ and $\ddot{x}_P$ are negligible. In addition, the P_L and the P_H can be expressed using (4). From the definition, the P_L is the value where the piston is about to move, so the piston displacement is zero, i.e. $x_p = 0$. The P_H is the value where the piston is about to contact the friction plate, i.e. x_p=$x_{p,max}$. Therefore, the following equations can be derived:

$$P_L = \frac{1}{A_c} F_{pre} \tag{5}$$

$$P_H = \frac{1}{A_c}(F_{pre} + k x_{p,max}) \tag{6}$$

In fact, the design parameter A_c and $x_{p,max}$ are known values. If we measure the P_L and P_H as shown in Fig. 2, other parameters such as F_{pre} and k can be calculated directly.

In general, the flow rate Q filled in the clutch chamber is determined using the orifice equation as follows:

$$\Delta P = u - P_c \tag{7}$$

$$Q = C_d A_{ori} \sqrt{\frac{2}{\rho} \Delta P} \tag{8}$$

where C_d and A_{ori} are the discharge coefficient, and the area of orifice, respectively. The equation (8) comes from the assumption that turbulent flow occurs after passing the orifice. However, the clutch chamber geometrically resembles a sudden expansion pipe rather than an orifice. In quasi-static flow, a streamline can be formed after passing the pipe. This means the flow rate can be effected by laminar flow. In laminar flow, the equation (8) becomes as follows:

$$Q = \frac{\pi r^4}{8 \mu L} \Delta P \tag{9}$$

where μ, L and r represent the fluid viscosity, pipe length, and pipe radius, respectively.

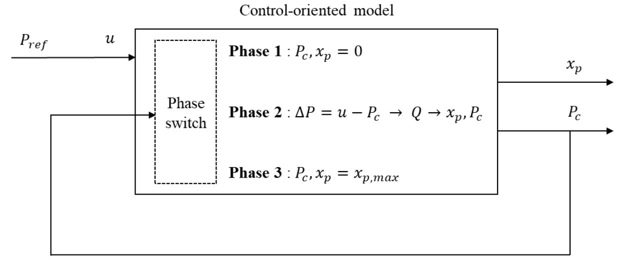


Fig. 3: Overall Scheme of control-oriented model for clutch actuation system

As mentioned above, laminar flow and turbulent flow coexist in the clutch chamber flow. Therefore, the flow equation of the chamber can be described as summation of (8) and (9) [11].

$$Q = K_1 \Delta P + K_2 \sqrt{\Delta P} \tag{10}$$

Here, K_1 and K_2 represent the coefficient of laminar flow and turbulent flow, respectively. K_1 and K_2 value should be tuned based on experiment data. The volume of fluid is integral of flow rate. Thus, the piston displacement can be calculated as follow.

$$x_p = \begin{cases} \frac{1}{A_c} \int Q dt & \text{at filling} \\ x_{p,max} - \frac{1}{A_c} \int Q dt & \text{at dumping} \end{cases} \tag{11}$$

The dumping case means that (7) is negative. In this paper, the filling phase is the only considered region because the dumping phase does not mainly affect the shift process.

Therefore, the dynamical behavior in phase 2 can be described by (4), (6), (10), and (11). As mentioned above, these models have only two tuning parameters, K_1 and K_2. Other unknown parameters can be determined according to the experimental data. In fact, the design parameters A_c and $x_{p,max}$, which are thought to be known values, are slightly different for real ones due to part-to-part variation or other uncertainties. However, the proposed model can compensate the parameter error by tuning the appropriate K_1 and K_2 values. In summary, this proposed model is quite simple compared with (1).

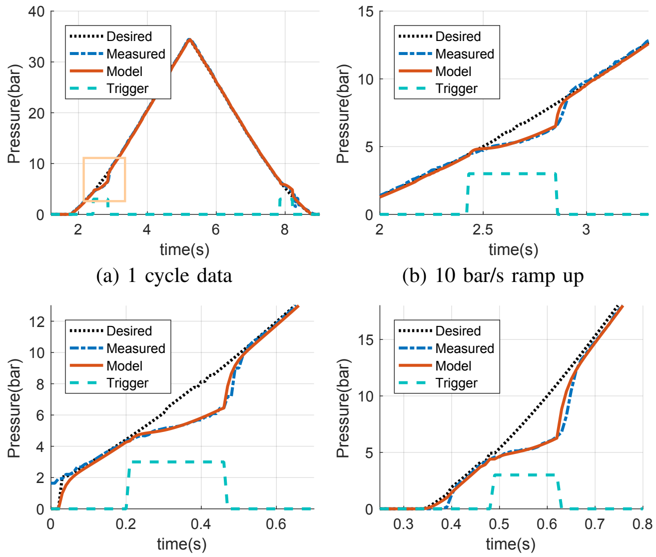
As a result, the control-oriented model is proposed in this section. The whole process of hydraulic actuation system is divided into 3 phases. In phase 1 and 3, input and output relationships are expressed as a 1st order lag model. In phase 2, the clutch actuation system is modeled using the reduced form based on physical phenomena. Each phase is determined by the calculated clutch pressure P_c. The command pressure input and model pressure are used to update the clutch pressure and phase transition. The overall structure of model is depicted in Fig.3.

C. Model validation

In order to validate the proposed control-oriented model, we compared the model output data with the experimental data of EHV in test vehicle. Since the system behavior of interest is the filling phase in the torque phase, the experiment and comparisons were made on open-loop ramp up

TABLE I: SYSTEM PARAMETERS

Parameter	Value	Parameter	Value
A_c	0.0113 (m^2)	P_H	6.55 (bar)
$x_{p,max}$	0.003 (mm)	K_1	5.5e-3
P_L	4.42 (bar)	K_2	1e-3



(a) 1 cycle data (b) 10 bar/s ramp up (c) 20 bar/s ramp up (d) 40 bar/s ramp up

Fig. 4: Model validation : (a) pressure output of 1 cycle, (b) scenario 1: 10bar/s input, (c) scenario 2: 20bar/s input, (d) scenario 3: 40bar/s input

scenarios with various slopes. Comparisons were conducted with the computed model outputs and experiment data based on the same input from the experiment. The measured and identified parameter values for control-oriented model is given in TABLE I.

Note that six parameters are only used to generate the model output; two of them are known values $(A_c, x_{p,max})$, two of them are measured from pre-experiment (P_L, P_H), the others are tuning parameters. By utilizing the above parameters, the system model is no longer stiff and casn be used and run in the production type TCU. The validation results are depicted in Fig.4

Fig.4a shows the full data of one cycle of the triangle wave input. It is shown that the actual measured pressure (blue dashed-dotted line) does not follow the desired pressure (black dotted line) due to the filling and the dumping phase. Since this paper focuses on the filling phase, the model accuracy in phase 2 is very important. Fig.4b,4c,4d show the validation results of the filling phase for various slopes of the input. The cyan dashed lines indicate the trigger of phase 2, the filling phase and the dumping phase. It can be seen that the model outputs (red solid line) are generated very similar to the actual measured output regardless of the input gradient. Therefore, it can be concluded that the proposed control-oriented model well describes the behavior of actual EHV system.

III. Controller Design

A. Model-based controller design

In the case of wet DCT with a large effect of a drag torque, it is inevitably required to maintain an under-fill state during the clutch-fill process. Starting the torque phase in the under-fill state, the delayed response due to the filling phase appears as shown in the Fig.2 for the ramp up input. Therefore, pressure compensation of the filling phase is essential to generate the desired clutch torque in the torque phase since a clutch torque is proportional to a clutch pressure [12]. As a result, compensating the pressure in filling phase can be interpreted as a pressure tracking control problem.

In fact, the simplest method to solve the tracking problem is to implement a proportional-integral-derivative (PID) controller. However, there are some practical issues to apply the PID controller. In shift process, the filling phase is the short duration as shown in Fig. 4. Thus, the controller must have a fast convergence rate. In general, convergence time and system stability are in a trade-off relationship. As mentioned in the previous section, pressure overshoot in the torque phase causes the critical problem in terms of ride quality. Therefore, it is hard to match the desired performance using a simple feedback control. To obtain the fast convergence rate and stable input, the model-based controller is needed.

In order to design a controller to track a desired pressure, the model in the filling phase can be reorganized as follows.

$$P_c = \frac{1}{A_c}(F_{pre} + kx_p) \tag{12}$$
$$\dot{x}_p = \frac{1}{A_c}(K_1(u - P_c) + K_2\sqrt{u - P_c}) \tag{13}$$

By differentiating both sides of (12) and substituting it into (13), the system can be expressed as the following pressure dynamics.

$$\begin{aligned}\dot{P}_c &= \frac{k}{A_c^2}(K_1(u - P_c) + K_2\sqrt{u - P_c}) \\ &= a_1(u - P_c) + a_2\sqrt{u - P_c}\end{aligned} \tag{14}$$

where 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 coefficients of the laminar and turbulent flow, respectively. The values of a_1 and a_2 can be easily calculated by physical properties, which are denoted in TABLE I. Thus, the system equation can be simply expressed as (14).

The equation (14) shows that the filling dynamics is nonlinear non-affine system due to the square root term. Since the control object is to track a clutch pressure to a desired one, a tracking error and its dynamics can be defined as follows.

$$e \triangleq P_d - P_c \tag{15}$$
$$\dot{e} + \lambda e = 0 \tag{16}$$

where P_d and λ represent the desired pressure and the time constant, respectively. By using (16) and (14), the control input can be derived using a reverse design approach so that the error e will go to zero asymptotically. Since the system equation is nonlinear, the virtual input u^* is defined to simply solve the error dynamics. The whole right side of (14) can be defined as a virtual input.

$$u^* \triangleq a_1(u - P_c) + a_2\sqrt{u - P_c} \tag{17}$$

Therefore, the system equation can be reorganized using virtual input.

$$\dot{P}_c = u^* \tag{18}$$

By combining the (15), (16) and (18), the following virtual input value that makes the tracking error to be zero can be obtained.

$$u^* = \dot{P}_d + \lambda e \tag{19}$$

The final step is to calculate the actual input u from the obtained the virtual input u^*. The actual control input u can be calculated using (17). For convenience, the following procedure can be obtained by using new definition.

$$P_0 \triangleq \sqrt{u - P_c} \tag{20}$$

$$u^* = a_1 P_0^2 + a_2 P_0 \tag{21}$$

$$u = P_0^2 + P_c \tag{22}$$

The P_0 defined in (20) can be obtained by solving the quadratic form of (21). Since a_1, a_2, and P_0 are all positive value by definition, the larger root of quadratic equation is selected as the solution of (21). As a result, the control input u is obtained as (22), and the solution can be expressed explicitly as follows.

$$u = \left(\frac{-a_2 + \sqrt{a_2^2 + 4a_1(\dot{P}_d + \lambda e)}}{2a_1} \right)^2 + P_c \tag{23}$$

As described above, a model-based controller is constructed. The controller is composed of the feed-forward part that compensates the nominal behavior of system dynamics and the feedback part that compensates the model error or uncertainties. If pressure measurement is not available, a pure feed-forward controller can be constructed by substituting measurement value P_c for desired value P_d. The performance of the pure feed-forward controller will depend on the accuracy of the model.

B. Determination of filling time

The most important factor in the filling phase compensation is the deciding when to switch from phase 2 to phase 3. As denoted in Fig.2, for the actual plant, it is not easy to define the exact P_H. In the modeling process, an approximation of P_H was acceptable, but the error of P_H could lead to a large overshoot in control process. Also, the value of P_H can vary with fluid temperature, line pressure, time delay, and modeling uncertainties. Therefore, a switching trigger should be obtained based on a physical quantity that do not change sensitively and do not mainly affect the results. The fixed physical quantity in clutch actuation system is the piston maximum displacement, i.e. the volume of clutch piston chamber. Since most of production vehicles do not have a piston displacement sensor, the volume that can be calculated by integrating the flow rate can be used as an index. From (10) and (11), the following equation can be obtained.

$$V = A_c x_p = \int (K_1(u - P_c) + K_2\sqrt{u - P_c})dt \tag{24}$$

If the pressure tracks well by using proposed controller, i.e. $P_d \simeq P_c$, the corresponding control input can be divided into the desired pressure and additional control input.

$$\begin{aligned} u &= P_d + u_{fill} \simeq P_c + u_{fill} \\ \text{where} \quad & u_{fill} \triangleq u - P_d \end{aligned} \tag{25}$$

Here, control input u is given in (23). The value u_{fill} is defined as additional control input of the filling phase. By substituting (25) into (24), volume of clutch chamber can be expressed as follows:

$$V = \int (K_1 u_{fill} + K_2\sqrt{u_{fill}})dt \tag{26}$$

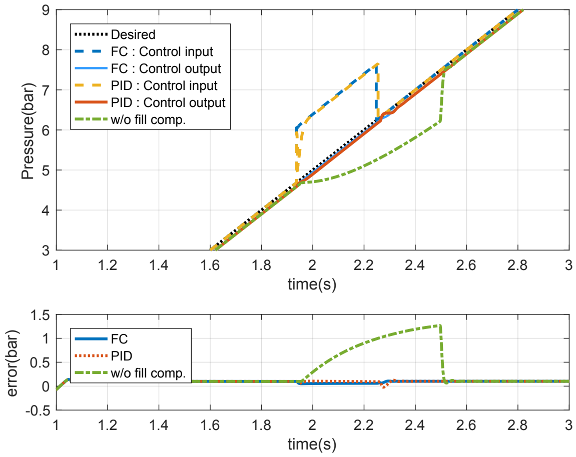
where $u - P_c = u_{fill} + P_d - P_c \simeq u_{fill}$ is utilized. Therefore, the volume of clutch chamber can be calculated by integrating additional inputs except for the desired command input. The proper volume V_{max} can be defined through pre-experiments such as K_1 and K_2.

IV. SIUMULATIONS

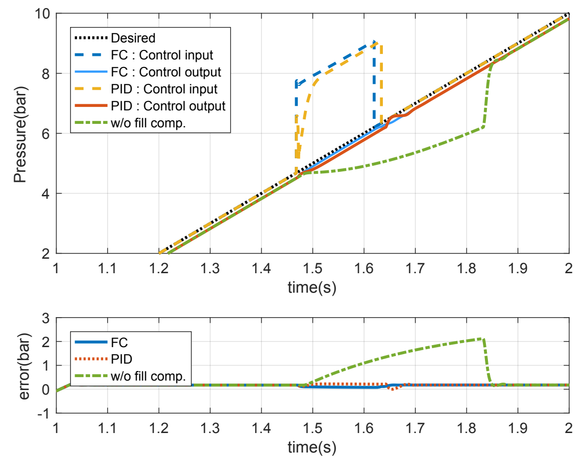
In order to verify the proposed controller, the simulations have been carried out. The actual plant used in the simulation was the model constructed by AMESim, which is a detailed description of the hydraulic system so that can be treated as the exact model of actual plant. AMESim includes the model uncertainties such as change of jet angle, friction of hydraulic line, line pressure drops, and seal drag and so on.

The ramp up input was given as a desired pressure. The shifting time is changed according to the driver's pedal input, and the gradient of the desired pressure is changed by the shifting schedule accordingly. Thus, various gradients of inputs are used to verify the controller performance. The simulation results are depicted in Fig.5.

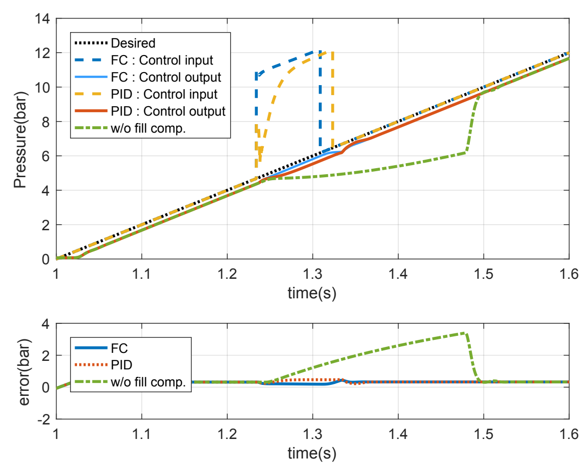
In Fig.5, the results of the proposed controller which are denoted as Fill Controller (FC) are compared with the results of conventional PID controller and open loop responses (dashed-dotted line). The corresponding control inputs and control outputs are described in dashed line and solid line, respectively. As shown in the simulation results, the proposed controller shows faster and stable control performance in short filling phase compared with the PID controller. Also, even though the proposed controller is developed based on the reduced order control-oriented model, it confirms the good tracking performances. The amount of the additional



(a) 5 bar/s ramp up



(b) 10 bar/s ramp up



(c) 20 bar/s ramp up

Fig. 5: Simulation results : comparison of the proposed filling controller (FC), PID controller, open-loop response

input is changed according to the gradient of the desired pressure since the proposed controller contains the model information. Because the volume of chamber is fixed, the total time to fill the clutch chamber also changes with the input gradient. Therefore, it can be confirmed that the proposed control logic shows good tracking performance in the filling phase regardless of the input gradient. The tracking errors are within 0.2bar for all scenarios, which are very small amount compared to open loop responses.

V. Conclusion

This paper presents a new control strategy for the clutch-fill process in the torque phase of the shift process in wet DCT. A control-oriented model is proposed by simplifying the well-known complex dynamics, and verified using experimental data. Based on the proposed control-oriented model, a nonlinear controller that tracks the desired pressure in the filling phase was developed. The proposed controller was verified by simulation using the exact model. Future work includes controller verification through real vehicle experiments. It is necessary to consider the tracking performance in the presence of various uncertainties through the actual vehicle test. In addition, this study is expected to be extended to other studies dealing with the existing control strategy of the filling phase.

Acknowledgement

This works was supported by Hyundai Motor Company; in parts by the BK21 Plus Program.

References

[1] S. Kim and S. Choi, "Control-oriented modeling and torque estimations for vehicle driveline with dual-clutch transmission," *Mechanism and Machine Theory*, vol. 121, pp. 633–649, 2018.

[2] H. Chen and B. Gao, *Nonlinear estimation and control of automotive drivetrains*. Springer Science & Business Media, 2013.

[3] X. Song, M. A. M. Zulkefli, and Z. Sun, "Automotive transmission clutch fill optimal control: An experimental investigation," in *Proceedings of the 2010 American Control Conference*. IEEE, 2010, pp. 2748–2753.

[4] H. Hao, T. Lu, J. Zhang, and B. Zhou, "A new control strategy of the filling phase for wet dual clutch transmission," *Proceedings of the Institution of Mechanical Engineers, Part C: Journal of Mechanical Engineering Science*, vol. 230, no. 12, pp. 2013–2027, 2016.

[5] X. Song, M. A. M. Zulkefli, Z. Sun, and H.-C. Miao, "Automotive transmission clutch fill control using a customized dynamic programming method," *Journal of Dynamic Systems, Measurement, and Control*, vol. 133, no. 5, p. 054503, 2011.

[6] W. Guo, Y. Liu, J. Zhang, and X. Xu, "Dynamic analysis and control of the clutch filling process in clutch-to-clutch transmissions," *Mathematical Problems in Engineering*, vol. 2014, 2014.

[7] S. Iqbal, F. Al-Bender, A. P. Ompusunggu, B. Pluymers, and W. Desmet, "Modeling and analysis of wet friction clutch engagement dynamics," *Mechanical Systems and Signal Processing*, vol. 60, pp. 420–436, 2015.

[8] X. Song, C.-S. Wu, and Z. Sun, "Design, modeling, and control of a novel automotive transmission clutch actuation system," *IEEE/ASME Transactions on Mechatronics*, vol. 17, no. 3, pp. 582–587, 2012.

[9] K. D. Mishra and K. Srinivasan, "Robust control and estimation of clutch-to-clutch shifts," *Control Engineering Practice*, vol. 65, pp. 100–114, 2017.

[10] X. Song and Z. Sun, "Pressure-based clutch control for automotive transmissions using a sliding-mode controller," *IEEE/ASME Transactions on mechatronics*, vol. 17, no. 3, pp. 534–546, 2012.

[11] S. Thornton, G. M. Pietron, D. Yanakiev, J. McCallum, and A. Annaswamy, "Hydraulic clutch modeling for automotive control," in *52nd IEEE Conference on Decision and Control*. IEEE, 2013, pp. 2828–2833.

[12] P. D. Walker, N. Zhang, and R. Tamba, "Control of gear shifts in dual clutch transmission powertrains," *Mechanical Systems and Signal Processing*, vol. 25, no. 6, pp. 1923–1936, 2011.