Adaptive slip engagement control of a wet clutch in vehicle powertrain based on transmitted torque estimation

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***Abstract*—In the slip engagement of a wet clutch, the solenoid current-to-actuator piston pressure model (CPM) and the piston pressure-to-clutch transmitted torque model (PTM) are utilized for the clutch torque control. However, CPM is changed depending on the amount of hysteresis which is influenced by hydraulic fluid temperature, and PTM is changed due to the variation of a clutch friction coefficient and touch point. Thus, the adaptation of CPM and PTM should be conducted using the measurement information of the solenoid current and piston pressure for precise control of the clutch torque. In this study, an adaptive control method of the clutch torque in the slip engagement of a wet clutch is proposed, which does not require the measurement of the piston pressure. In this method, the adaptation of the target piston pressure-to-clutch torque model (TPTM) is performed, and the clutch torque is controlled using compensated TPTM and nominal CPM. The proposed control method is verified experimentally by applying it to the slip engagement control of an engine clutch equipped in a production parallel hybrid vehicle.**

***Index Terms*—Powertrain control, Clutch control, Slip control, Adaptive control, Sensorless control, Wet clutch.**

I. Introduction

CLUTCHES are widely used in a vehicle powertrain. Fig. 1 shows the driveline structure of a parallel hybrid vehicle. In the driveline of a parallel hybrid vehicle, an engine clutch is used between an engine and driving motor to transmit and interrupt engine power to wheels, and a transmission clutch is used in a transmission.

The slip engagement method of a clutch is the method to engage the clutch when the rotational speed of both sides of the clutch is different. When the clutch is engaged by the slip engagement method, clutch friction energy loss, clutch wear and vehicle jerk occur due to the clutch slip.

On the other hand, a clutch often needs to be slip-engaged in a vehicle powertrain. For example, in the case of a parallel hybrid vehicle, an engine clutch should be slip-engaged when the vehicle launches in a steep hill, driving motor torque is insufficient for the vehicle to launch, and the rotational speed of the driving motor is lower than the idle rotational speed of an engine. Also, a transmission clutch should be slip-engaged at every shifting.

The clutch friction energy loss and vehicle jerk that occur in the clutch slip engagement are related to clutch transmitted torque. Therefore, it is necessary to precisely control the clutch torque in order to improve vehicle efficiency and ride comfort in the clutch slip engagement [1]–[3].

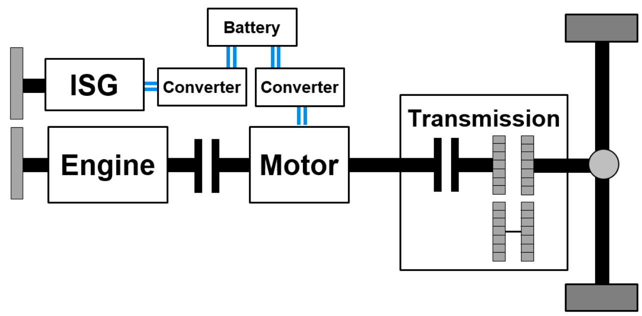

Fig. 1. Driveline structure of a parallel hybrid vehicle.

On the other hand, the measurement of the clutch torque is needed to control the clutch torque precisely. However, in production vehicles, clutch torque sensors cannot be fitted due to price and space issues [4]. Therefore, it is necessary to estimate the clutch torque using a vehicle driveline model.

There are two types of a clutch utilized in a vehicle powertrain: a dry clutch and wet clutch. Fig. 2 shows the typical actuator structure of a wet clutch which is mainly addressed in this study. Referring to Fig. 2, typically, in a wet clutch actuator, solenoid current and piston hydraulic pressure are measured. Thus, in the slip engagement of a wet clutch, the solenoid current-to-actuator piston pressure model (CPM) and the piston pressure-to-clutch torque (PTM) model are utilized for the precise control of the clutch torque.

Fig. 3 shows an example of CPM in a wet clutch [5]–[8]. Referring to Fig. 3, CPM is changed depending on the amount of hysteresis which is influenced by hydraulic fluid temperature.

Fig. 4 shows an example of PTM in a wet clutch [9]–[11]. Referring to Fig. 4, a clutch touch point is changed due to clutch wear and the x-axis intercept of PTM is changed due to the variation of the clutch touch point. In addition, a clutch friction coefficient is changed depending on clutch temperature and the slope of PTM is changed due to the variation of the clutch friction coefficient.

Thus, the adaptation of CPM and PTM should be conducted using the measurement information of the solenoid current and piston pressure for the precise control of the clutch torque.

However, if the measurement information of the piston pressure is not available, the measured piston pressure should

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Manuscript received April 19, 2005; revised August 26, 2015.

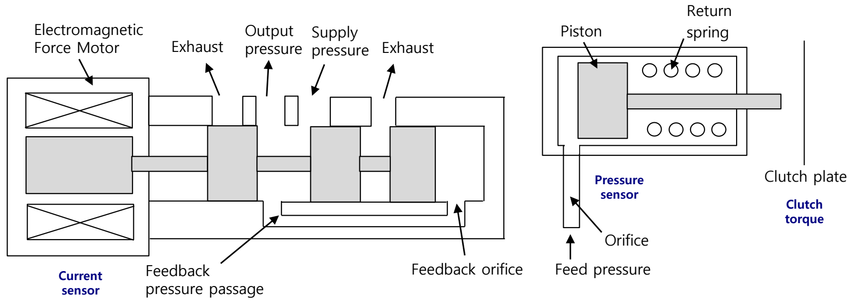

Fig. 2. Actuator structure of a wet clutch.

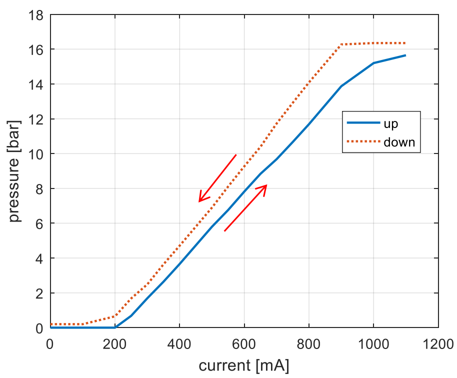

Fig. 3. Example of CPM in a wet clutch.

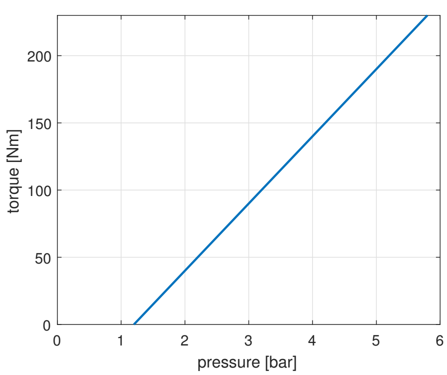

Fig. 4. Example of PTM in a wet clutch.

be replaced by the target piston pressure, and the adaptation of the target piston pressure-to-clutch torque model (TPTM) can be performed instead of the adaptation of CPM and PTM.

Here, if the amount of hysteresis is not considered perfectly in CPM, the offset between the target and measured piston pressure can occur, which can lead to the variation of the x-axis intercept of TPTM.

In previous studies, A tracking control method of actuator position was studied in [12], [13]. A tracking control method of the clutch torque considering the uncertainty on a friction coefficient in a clutch friction model in the clutch slip engagement was addressed in [14]–[18]. A method for estimating a clutch touch point under a specific situation was proposed in [19]–[22]. a physical and geometrical piston pressure-to-clutch torque models were introduced and a model adaptation method was dealt with in [9]–[11]. A vehicle driveline model was studied in [23]–[25]. A clutch torque estimation method was addressed in [4], [17], [26]–[29].

As mentioned above, some researches on an adaptive control method of the clutch torque based on one parameter adaptation such as a clutch friction coefficient, and clutch touch point have been conducted. However, few research on an adaptive control method of the clutch torque based on the adaptation of the entire uncertainty on the actuator lowest control variable(solenoid current in the case of a wet clutch)-to-clutch torque model has been performed.

In this study, an adaptive control method of the clutch torque in the slip engagement of a wet clutch is proposed, which does not require the measurement of the piston pressure. In this method, the adaptation of TPTM is performed, and the clutch torque is controlled using compensated TPTM and nominal CPM in a feedforward manner. The uncertainty on TPTM is compensated using a slope gain and the amount of parallel shift in the x-axis, and the two parameters are estimated using a clutch torque estimator. The proposed control method is verified experimentally by applying it to the slip engagement control of an engine clutch equipped in a production parallel hybrid vehicle.

This study is an extended study of an adaptive control method of the clutch torque in a dry clutch, which was previously proposed by this author [30], [31]. In this study, an adaptive control method of the clutch torque in a wet clutch without using the measurement information of piston pressure is mainly addressed.

The main contribution of this study is to propose an adaptive control method to compensate the entire uncertainty on the solenoid current-to-clutch torque model in a wet clutch, and to precisely control clutch torque without a piston pressure sensor which is mainly used for the clutch torque control of a wet clutch.

The remaining part of this paper is organized as follows. Section 2 describes an estimation method of the clutch torque. Section 3 addresses an adaptation method of TPTM of a wet clutch actuator. Section 4 deals with a feedforward control method of the clutch torque. Section 5 shows the control results of the clutch torque. And, section 6 concludes this paper.

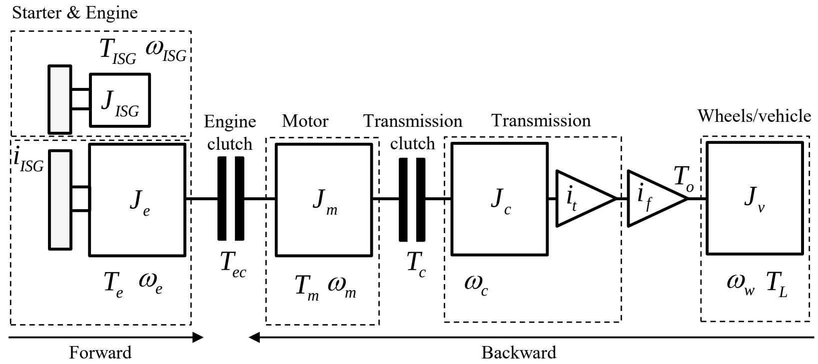

Fig. 5. Schematic diagram of estimation methods of the engine clutch torque.

II. Clutch transmitted torque estimation

As mentioned in the introduction section, in this study, an adaptive control method of the clutch torque in the slip engagement of a wet clutch is proposed, which does not require the measurement of the piston pressure. In this method, the adaptation of TPTM is performed, and the clutch torque is controlled using compensated TPTM and nominal CPM in a feedforward manner. The uncertainty on TPTM is compensated using a slope gain and the amount of parallel shift in the x-axis, and the two parameters are estimated using a clutch torque estimator.

Therefore, in this study, an estimation method of the clutch torque is introduced before an adaptive control method is dealt with.

As mentioned in the introduction section, there is no clutch torque sensor in a production vehicle. Therefore, in order to control the clutch torque precisely, it is necessary to estimate the clutch torque.

Since the control method proposed in this study is applied to the slip engagement control of a wet engine clutch, the following subsection introduces a driveline model of a parallel hybrid vehicle for estimating the engine clutch torque.

A. Driveline model

Generally, in the driveline of a parallel hybrid vehicle, since the rotational speed of an engine, motor, clutch input shaft, and wheels are measured, a lumped inertia model can be established like below, considering speed measurement positions [23]–[25].

$$J_e\dot{\omega}_e = T_e - T_{ec} \tag{1}$$

$$J_m\dot{\omega}_m = T_{ec} + T_m - T_c \tag{2}$$

$$J_c\dot{\omega}_c = T_c - \frac{1}{i_t i_f}T_o \tag{3}$$

$$J_v\dot{\omega}_w = T_o - T_r \tag{4}$$

$$T_e = f(\omega_e, \dot{m}_{ai}) \tag{5}$$

$$T_o = k_o(\frac{\theta_c}{i_t i_f} - \theta_w) + b_o(\frac{\omega_c}{i_t i_f} - \omega_w) \tag{6}$$

$$T_r = r_w\{m_v g\sin(\theta_{road}) + K_{rr}m_v g\cos(\theta_{road}) + \frac{1}{2}\rho v_x^2 C_d A\} \tag{7}$$

Where T, J, θ, ω, and i denote torque, rotational inertia, rotational angle, rotational speed, and gear ratio, respectively, and the subscripts e, ec, m, c, t, f, o, v, and r mean the engine, engine clutch, motor, transmission clutch, transmission gear, final gear, output shaft, vehicle, and road resistance, respectively. In addition, $\dot{m}_{ai}$, k_o, b_o, r_w, m_v, g, θ_{road}, , v_x, C_d, and A denote the cylinder intake air flow rate, output shaft spring coefficient, output shaft damping coefficient, wheel radius, vehicle mass, gravitational acceleration, road surface gradient, air density, vehicle longitudinal velocity, air drag coefficient, and vehicle frontal area, respectively.

B. Clutch torque estimator

Fig. 5 shows the schematic diagram of estimation methods of the engine clutch torque. There are two methods to estimate the engine clutch torque [4], [17], [26]–[29]. The first is the driveline forward estimation method in which the engine clutch torque is estimated starting from the engine torque. And, the second is the driveline backward estimation method in which the engine clutch torque is estimated starting from the output shaft torque.

In calculating the output shaft torque in the driveline backward estimation method of the engine clutch torque, the shaft compliance torque model of equation (6) or the road resistance torque model of equation (7) is utilized.

However, the shaft compliance torque model cannot describe the backlash phenomenon between axes and the output shaft torque can diverge when the transmission gear ratio is not accurate during gear shifting [32], [33]. Also, the road resistance torque model includes the uncertainty on the vehicle mass, road surface gradient, etc. Therefore, it is not preferable to utilize the driveline backward estimation method for estimating the engine clutch torque due to the large model uncertainty in a production vehicle.

However, the engine torque used in the forward estimation method of the engine clutch torque can be estimated more or less accurately based on the engine rotational speed and throttle intake air flow rate. Therefore, in this study, the driveline forward method is mainly utilized for estimating the

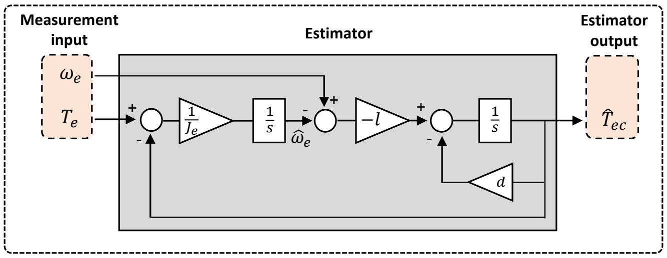

Fig. 6. Structure of the engine clutch torque estimator.

engine clutch torque, and the engine clutch torque estimator used in this study can be expressed as follows.

$$\dot{\hat{\omega}}_e = \frac{1}{J_e}(T_e - \hat{T}_{ec}) \tag{8}$$

$$\dot{\hat{T}}_{ec} = -d\hat{T}_{ec} - l(\omega_e - \hat{\omega}_e) \tag{9}$$

Where means an estimated value, d, and l are tuning parameters.

By differentiating equation (9) and substituting equation (8), it can be expressed as follows.

$$\ddot{\hat{T}}_{ec} + d\dot{\hat{T}}_{ec} + \frac{l}{J_e}\hat{T}_{ec} = \frac{l}{J_e}(T_e - J_e\dot{\omega}_e) \tag{10}$$

In the engine clutch torque estimator used in this study, the engine clutch torque is the filtered value of the value obtained by subtracting the engine inertia torque from the engine torque in the right-hand side of equation (10), and the characteristics in the frequency domain of the filter can be set by adjusting the tuning parameters d, and l.

In a special case where the following conditions are satisfied, equation (10) is modified as follows.

$$d = 2\sqrt{\frac{l}{J_e}} \tag{11}$$

$$\ddot{\hat{T}}_{ec} + 2\sqrt{\frac{l}{J_e}}\dot{\hat{T}}_{ec} + \frac{l}{J_e}\hat{T}_{ec} = \frac{l}{J_e}(T_e - J_e\dot{\omega}_e) \tag{12}$$

When equation (12) is Laplace-transformed, the following equation is derived.

$$\hat{T}_{ec}(s) = \frac{\frac{l}{J_e}}{s + \sqrt{\frac{l}{J_e}}}(T_e - J_e\dot{\omega}_e)(s) \tag{13}$$

Therefore, when the above condition is satisfied, the engine clutch torque is the low pass filtered value of the value obtained by subtracting the engine inertia torque from the engine torque, and the cut-off frequency of the low pass filter is set by adjusting the tuning parameter l.

The structure of the engine clutch torque estimator using equations (8) and (9) is shown in Fig. 6.

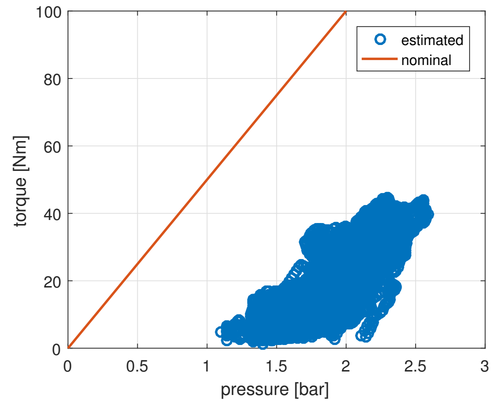

Fig. 7. Estimated engine clutch torque and nominal TPTM.

III. ADAPTATION OF TPTM

As mentioned in the introduction section, referring to Fig. 2, typically, in a wet clutch actuator, solenoid current and piston pressure are measured. Thus, in the slip engagement of a wet clutch, CPM and PTM are utilized for the precise control of the clutch torque. CPM, and PTM are shown like Fig. 3, and Fig. 4, respectively.

Referring to Fig. 3, CPM is changed depending on the amount of hysteresis which is influenced by hydraulic fluid temperature.

Referring to Fig. 4, a clutch touch point is changed due to clutch wear and the x-axis intercept of PTM is changed due to the variation of the clutch touch point. In addition, a clutch friction coefficient is changed depending on clutch temperature and the slope of PTM is changed due to the variation of the clutch friction coefficient.

Thus, the adaptation of CPM and PTM should be conducted using the measurement information of the solenoid current and piston pressure for the precise control of the clutch torque.

However, if the measurement information of the piston pressure is not available, the measured piston pressure should be replaced by the target piston pressure, and the adaptation of TPTM can be performed instead of the adaptation of CPM and PTM.

Here, if the amount of hysteresis is not considered perfectly in CPM, the offset between the target and measured piston pressure can occur, which can lead to the variation of the x-axis intercept of TPTM.

Since the control method proposed in this study is applied to the slip engagement control of a wet engine clutch, the clutch torque mentioned below means the engine clutch torque.

Fig. 7 shows the points of the estimated engine clutch torque relative to the target piston pressure, and nominal TPTM. Referring to Fig. 7, in order to control the clutch torque precisely based on the target piston pressure, nominal TPTM should be fitted as much as possible to the points of the estimated engine clutch torque relative to the target piston pressure. In this study, nominal TPTM is fitted to the estimated engine clutch torque using a slope gain and the amount of parallel shift in the x-axis. And, the engine clutch torque is controlled using compensated TPTM and nominal CPM in a feedforward manner. This method proposed in this study does not require the measurement of the piston pressure.

In this study, two methods to estimate the slope gain and the amount of parallel shift in the x-axis are proposed. The first method is to estimate the slope gain and the amount of parallel shift simultaneously. In this study, a Recursive Least Square (RLS) method is used as the simultaneous estimation method.

The second method is an independent estimation method of the slope gain and the amount of parallel shift, which is newly proposed in this study, using a moving average and gradient adaptation methods. The performance of each estimation method is discussed in the later results section.

The mentioned estimation of the slope gain and the amount of parallel shift is performed under the clutch slip engagement condition, and the points of the estimated engine clutch torque relative to the target piston pressure are also collected in the clutch slip engagement condition. The clutch slip engagement is determined under the following conditions.

$$c_1 < p_t < c_2 \tag{14}$$

$$c_3 < |\omega_e - \omega_m| \tag{15}$$

$$c_4 < \hat{T}_{ec} \tag{16}$$

Where, p_t denotes the target piston pressure, and c_1, c_1, c_3, and c_4 are threshold tuning parameters.

A. Simultaneous estimation

In this study, a RLS method is used to simultaneously estimate the slope gain and the amount of parallel shift in the x-axis of nominal TPTM. Since there is no measurement of the engine clutch torque in a production parallel hybrid vehicle, the estimated engine clutch torque mentioned in the section 2 is considered as the measured engine clutch torque.

$$T_{ec} = \alpha f_n(p_t - \beta) = g(\alpha, \beta, p_t) \tag{17}$$

$$\begin{aligned} &\frac{dg}{d\alpha}(\alpha_{k-1}, \beta_{k-1}, p_t)\alpha_k + \frac{dg}{d\beta}(\alpha_{k-1}, \beta_{k-1}, p_t)\beta_k = \\ &T_{ec.k} - g(\alpha_{k-1}, \beta_{k-1}, p_t) \\ &+\frac{dg}{d\alpha}(\alpha_{k-1}, \beta_{k-1}, p_t)\alpha_{k-1} + \frac{dg}{d\beta}(\alpha_{k-1}, \beta_{k-1}, p_t)\beta_{k-1} \end{aligned} \tag{18}$$

$$\begin{aligned} z_k &= T_{ec.k} - g(\alpha_{k-1}, \beta_{k-1}, p_t) \\ &+\frac{dg}{d\alpha}(\alpha_{k-1}, \beta_{k-1}, p_t)\alpha_{k-1} + \frac{dg}{d\beta}(\alpha_{k-1}, \beta_{k-1}, p_t)\beta_{k-1} \\ &= \begin{pmatrix} \frac{dg}{d\alpha}(\alpha_{k-1}, \beta_{k-1}, p_t) & \frac{dg}{d\beta}(\alpha_{k-1}, \beta_{k-1}, p_t) \end{pmatrix} \begin{pmatrix} \alpha_k \\ \beta_k \end{pmatrix} \end{aligned} \tag{19}$$

$$x_k = \begin{pmatrix} \alpha_k \\ \beta_k \end{pmatrix} \tag{20}$$

$$H_k^T = \begin{pmatrix} \frac{dg}{d\alpha}(\alpha_{k-1}, \beta_{k-1}, p) \\ \frac{dg}{d\beta}(\alpha_{k-1}, \beta_{k-1}, p) \end{pmatrix} \tag{21}$$

$$\mathbf{K}_k = \mathbf{P}_{k-1}\mathbf{H}_k^T(\mathbf{H}_k\mathbf{P}_{k-1}\mathbf{H}_k^T + \mathbf{R}_k)^{-1} \tag{22}$$

$$\mathbf{P}_k = (\mathbf{I} - \mathbf{K}_k\mathbf{H}_k)\mathbf{P}_{k-1}(\mathbf{I} - \mathbf{K}_k\mathbf{H}_k)^T + \mathbf{K}_k\mathbf{R}_k\mathbf{K}_k^T \tag{23}$$

$$\hat{\mathbf{x}}_k = \hat{\mathbf{x}}_{k-1} + \mathbf{K}_k z_k \tag{24}$$

Where f_n denotes the nominal TPTM function, and α, and β are the slope gain and the amount parallel shift in the x-axis of nominal TPTM, and k is the time step in the clutch slip engagement.

B. Independent estimation

This subsection deals with the method of independently estimating the slope gain and the amount of parallel shift of nominal TPTM.

First, when the estimated engine clutch torque is within a specific range in the slip engagement of a wet clutch, the target piston pressure is moving-averaged to obtain an intermediate variable of the x-axis intercept of TPTM.

$$c_5 < \hat{T}_{ec} < c_6 \tag{25}$$

$$\hat{\delta}_{k+1} = \frac{n}{n+1}\hat{\delta}_k + \frac{1}{n+1}p_{t.k+1} \tag{26}$$

Where c_5, and c_6 are threshold tuning parameters, and δ, and n are the intermediate variable of the x-axis intercept of TPTM, and the number of time steps in the first moving averaging.

However, this intermediate variable can be changed greatly in each clutch slip engagement. Therefore, the final x-axis intercept of TPTM is obtained by moving-averaging the intermediate variable of the x-axis intercept at the end of each slip engagement.

$$\hat{\lambda}_{i+1} = \frac{m}{m+1}\hat{\lambda}_i + \frac{1}{m+1}\hat{\delta}_{i+1} \tag{27}$$

Where λ, and m denote the final x-axis intercept of TPTM, and the number of time steps in the second moving averaging.

Then, using the estimated and nominal x-axis intercept of TPTM, the amount of parallel shift in the x-axis of nominal TPTM can be obtained as follows.

$$\hat{\beta} = \hat{\lambda} - \lambda_n \tag{28}$$

Where λ_n means the nominal x-axis intercept of TPTM.

Then, assuming that the amount of parallel shift of nominal TPTM is known, the slope gain of nominal TPTM can be estimated through the gradient adaptation method as follow.

$$T_{ec} = \alpha f_n(p_t - \hat{\beta}), \;\; f_n(p_t - \hat{\beta}) > 0 \tag{29}$$

$$\hat{T}_{ec} = \hat{\alpha} f_n(p_t - \hat{\beta}) \tag{30}$$

$$\varepsilon = \hat{T}_{ec} - T_{ec} = (\hat{\alpha} - \alpha) f_n(p_t - \hat{\beta}) = \tilde{\alpha} f_n(p_t - \hat{\beta}) \tag{31}$$

$$\dot{\hat{\alpha}} = -\gamma_\alpha \varepsilon, \;\; \gamma_\alpha > 0 \tag{32}$$

Where λ_α denotes an adaptation gain.

To prove the stability of the above estimator, the estimation error of the slope gain is defined as follows. Then, the error dynamics can be obtained as follows using equations (29) and (30).

$$\tilde{\alpha} = \hat{\alpha} - \alpha \tag{33}$$

$$\dot{\tilde{\alpha}} = -\gamma_\alpha \tilde{\alpha} f_n(p_t - \hat{\beta}) \tag{34}$$

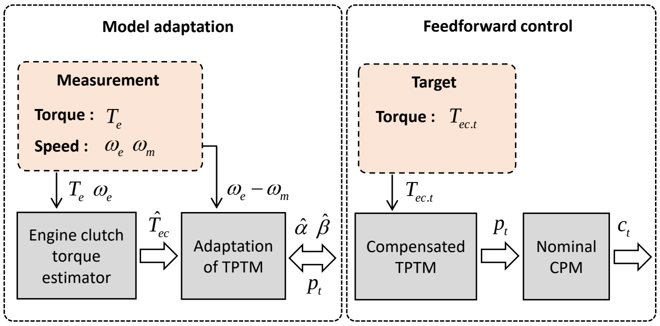

Fig. 8. Entire adaptive control algorithm of the engine clutch torque.

Finally, a Lyapunov equation is defined as follows and the stability of the slope gain estimator can be easily proved by the Lyapunove stability criterion.

$$V = \frac{1}{2\gamma_\alpha}\tilde{\alpha}^2 > 0 \tag{35}$$

$$\dot{V} = -\tilde{\alpha}^2 f_n(p_t - \hat{\beta}) < 0 \tag{36}$$

IV. Feedforward tracking control of clutch troque

In the previous section, the estimation method of the slope gain and the amount of parallel shift in the x-axis of nominal TPTM was addressed.

In this section, a feedforward tracking control method of the clutch torque using compensated TPTM and nominal CPT is introduced.

Since the control method proposed in this study is applied to the slip engagement control of a wet engine clutch, the clutch torque mentioned below means the engine clutch torque.

The following equation can be obtained by inversely transforming equation (28).

$$p_t = \hat{\alpha} f_n^{-1}(\hat{T}_{ec}) + \hat{\beta} \tag{37}$$

Here, when the estimated engine clutch torque is replaced by the target engine clutch torque, the target piston pressure can be obtained as follows.

$$p_t = \hat{\alpha} f_n^{-1}(T_{ec.t}) + \hat{\beta} \tag{38}$$

Where $T_{ec.t}$ means the target engine clutch torque.

Also, assuming that h_n is the nominal CPT function, the target solenoid current can be obtained using the inverse function of h_n as follows.

$$c_t = h_n^{-1}(p_t) \tag{39}$$

Where c_t denotes the target solenoid current.

Finally, the target solenoid current is feedback controlled using the measurement information, then the target engine clutch torque is achieved. The entire control algorithm is depicted in Fig. 8. As can be seen in Fig. 8, no measurement information is used prior to the current tracking control.

The performance of the torque tracking control is covered in the later results section.

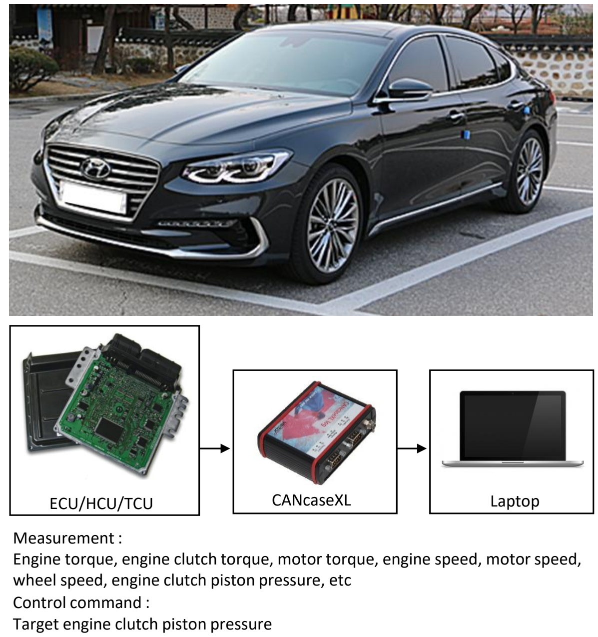

Fig. 9. Experimental vehicle and schematic diagram of measurement data acquirement.

V. Experimental results

A. Experimental environment

The adaptive control method of the clutch torque in the slip engagement of a wet clutch is verified by applying it to the slip engagement control of a wet engine clutch equipped in a production parallel hybrid vehicle.

Fig. 9 shows the experimental vehicle and schematic diagram of measurement data acquirement. The experimental vehicle was a parallel hybrid vehicle using an automatic transmission. the vehicle measurement data was acquired via CAN communication with a vehicle ECU using CANcaseXL from the Vector company.

The engine torque was estimated using the information of the engine rotational speed and throttle intake air flow rate, and the driving motor torque was estimated using the information of the motor current. Therefore, it was assumed that the engine torque and the driving motor torque are measurable.

In the following experiments, the engine torque, driving motor torque, engine rotational speed, driving motor rotational speed, wheel rotational speed, and piston pressure in the engine clutch actuator were measurable. In addition, when the target piston pressure signal is sent to an actuator solenoid driver, the pressure tracking control is performed by the solenoid driver itself in the feedforward manner. Thus, the solenoid current was not included in the measurement data.

Fig. 10 shows the installation diagram of a engine clutch torque sensor. a wireless flexplate torque sensor from the HBM company to measure the engine clutch torque was installed in the place between the flywheel and engine clutch. Thus, the engine clutch torque was additionally measurable in the following experiments.

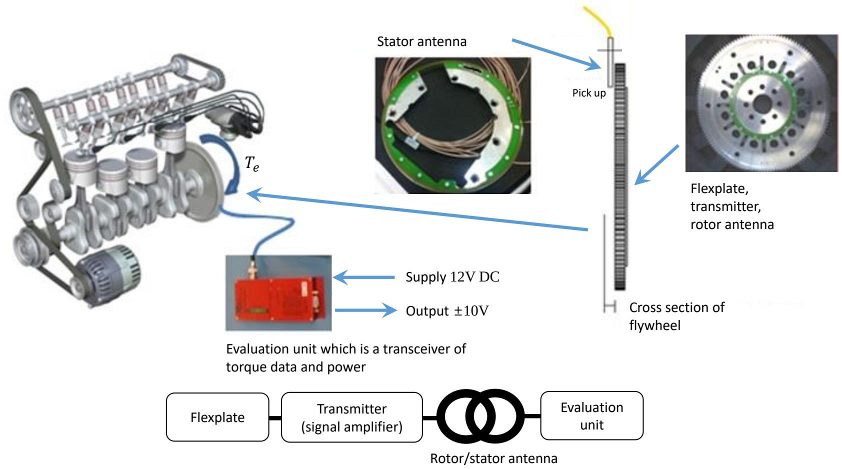

Fig. 10. Installation diagram of a engine clutch torque sensor.

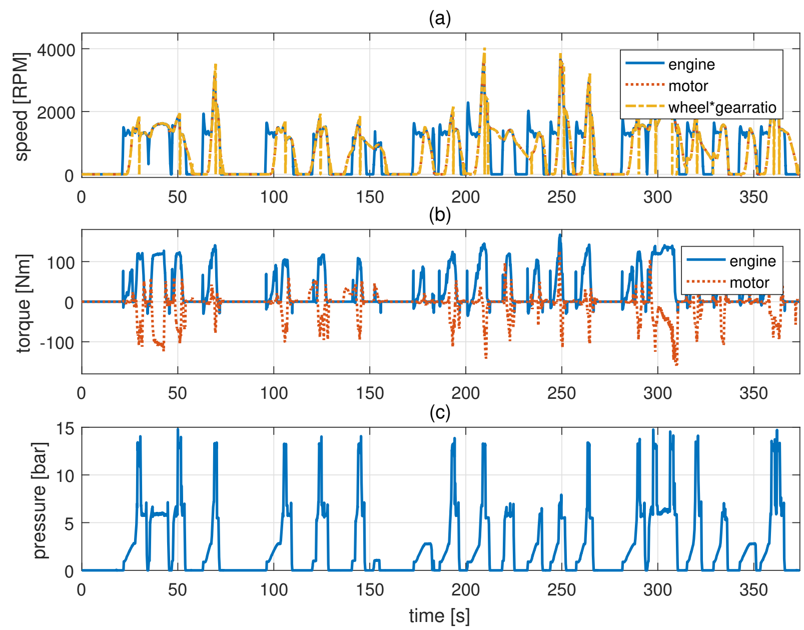

Fig. 11. Vehicle state information in the performance verification experiment of the engine clutch torque estimation: (a) driveline rotational speed, (b) driveline torque, (c) piston pressure.

B. Clutch transmitted torque estimation

In this section, the performance of the engine clutch torque estimation in the clutch slip engagement is verified. Thus, the slip engagement test of the engine clutch was conducted to verify the performance. When the vehicle was stationary and the engine was idling, the slip engagement of the engine clutch was repeated 10 times or more for about 400 seconds.

Fig. 11 shows the vehicle state information in the verification experiment of the performance of the engine clutch torque estimation. Fig. 11(a), (b), and (c) show the driveline rotational speed, driveline torque, piston pressure, respectively.

In the previous paper by the authors of this paper, the performance of various engine clutch torque estimators has been compared and it was verified that the practicality of the engine clutch torque estimator in the forward direction of the driveline in the clutch slip engagement is excellent. Therefore, in this study, the performance comparison among engine clutch torque estimators is omitted, and only the result of the engine clutch torque estimation of the forward estimator is addressed.

Fig. 12 shows the result of the engine clutch torque estimation in the performance verification experiment of the engine clutch torque estimation. Fig. 12(a) and (b) show the result over the entire experiment period, and the result over a specific experiment period.

The RMS error between the estimated and measured values of the engine clutch torque was 6.5278 Nm, and it was confirmed that the engine clutch torque did not diverge and the torque estimation was properly performed.

Also, in the previous study, the adaptive torque tracking control in the slip engagement of a dry engine clutch was performed well to the clutch torque RMS error of 10 Nm level. Thus, the performance of the clutch torque estimator of

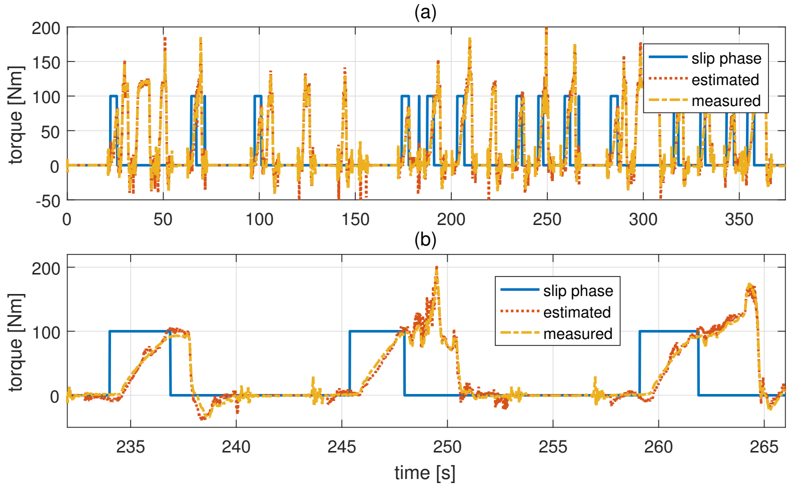

Fig. 12. Result of the engine clutch torque estimation in the performance verification experiment of the engine clutch torque estimation: (a) estimation result over the entire experiment period, (b) estimation result over a specific experiment period.

this study was expected to alright for performing the adaptive torque tracking control in the slip engagement of a wet engine clutch.

C. Adaptation of TPTM

In this section, the slip engagement test of the engine clutch was additionally performed to verify the performance of the TPTM adaptation method.

Like the performance verification experiment of the engine clutch torque estimation, when the vehicle was stationary and the engine was idling, the slip engagement of the engine clutch was repeated 60 times or more for about 1500 seconds.

Fig. 13 shows the vehicle state information in the performance verification experiment of the TPTM adaptation method. In Figure 13, only the first 500 seconds of the 1500 second result were shown considering the visibility of the figure. Fig. 13(a), (b), (c), and (d) show the driveline rotational speed, driveline torque, target and measured piston pressure, and offset between the target and measured piston pressure, respectively.

As mentioned in the section 3, it is very difficult to precisely consider the amount of hysteresis in CPT.

For this reason, referring to Fig. 13(d), the offset between the target and measured piston pressure was slightly changed in this study, when the slip engagement of the engine clutch was repeatedly performed.

This resulted in the variation of the x-axis intercept of TPTM. Fig. 14 shows the variation of the estimated engine clutch torque relative to the target piston pressure when the slip engagement of the engine clutch was repeatedly conducted. The estimated points indicated before in the legend are the points gathered from 0 second to 250 seconds, and the estimated points indicated after are gathered from 250 seconds to 500 seconds in Fig. 13. As shown in Fig. 14, it could be seen that TPTM is considerably changed when the slip engagement of the engine clutch was repeatedly conducted.

Fig. 15 shows the estimation result of the slope gain and the amount of parallel shift in the x-axis. Fig. 15 (a), and (b) show the estimation result of the slope gain, and the amount of parallel shift, respectively.

In Fig. 15, the repetitive slip engagement of the engine clutch was started in the vicinity of 20 seconds, 650 seconds, 950 seconds, and 1350 seconds. After several dozen repetitive clutch engagement, there was some rest to prevent clutch failure. The result of the independent estimation method of the slope gain and the amount of parallel shift in Fig. 15 shows that TPTM shifted to the left of the x-axis and the slope of TPTM increased when the slip engagement of the engine clutch was repeatedly conducted. This phenomenon can be also seen in Fig. 14. The amount of parallel shift in the x-axis of nominal TPTM in the slip engagement greatly affects the clutch engagement feeling. In the result of the independent estimation method of the slope gain and the amount of parallel shift in Fig. 15, the amount of parallel shift in the beginning of the clutch slip engagement was estimated quickly, and the variation of the slope gain and the amount of parallel shift was also estimated in detail.

However, in the result of the simultaneous estimation method of the slope gain and the amount of parallel shift in Fig. 15, the amount of parallel shift was estimated slightly slow, compared with the independent estimation method in the beginning of the clutch slip engagement. Also, the variation of the slope gain and the amount of parallel shift was not estimated in detail but only tendency could be estimated.

In order to increase the estimation speed of the slope gain and the amount of parallel shift, the tuning parameters: covariance R and P of the RLS method were tuned many times. However, it was shown that the estimation result diverged if the estimation speed was increased and the detail variation of the slope gain and the amount of parallel shift could not be estimated if the estimation speed was decreased.

In the case of the simultaneous estimation method, a lot of matrix multiplication is performed. Thus, it took a lot of memory in order to match the effective number of a lot of variables having different absolute sizes and the entire logic was somewhat complicated. On the other hand, in the case of

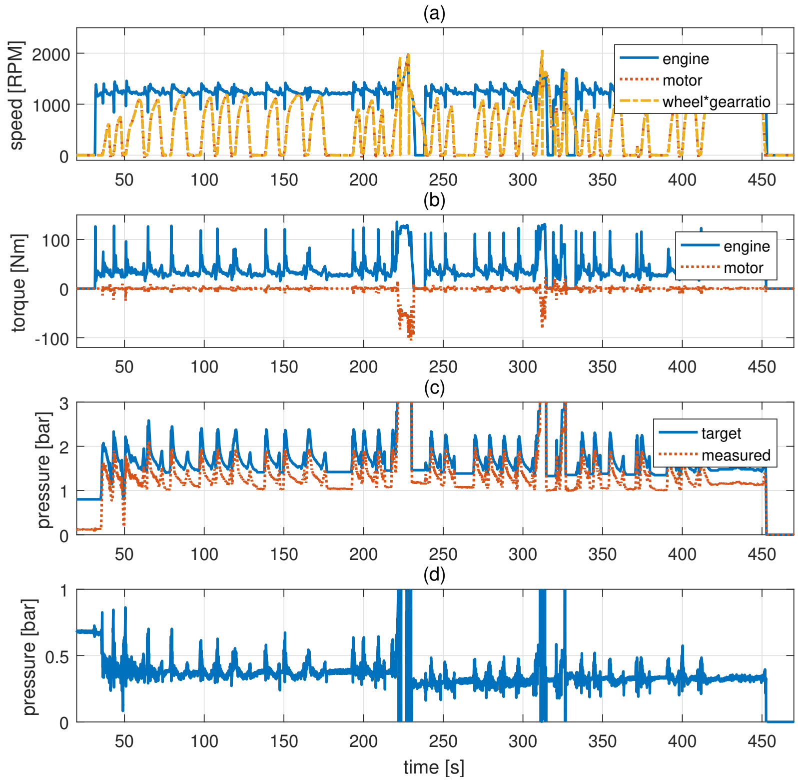

Fig. 13. Vehicle state information in the performance verification experiment of the TPTM adaptation method: (a) driveline rotational speed, (b) driveline torque, (c) piston pressure, (d) offset between the target and measured piston pressure.

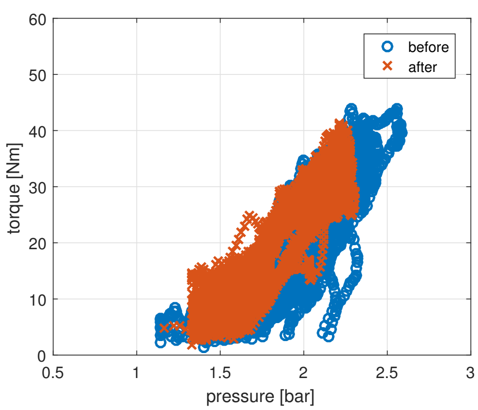

Fig. 14. Variation of the estimated engine clutch torque relative to the target piston hydraulic pressure.

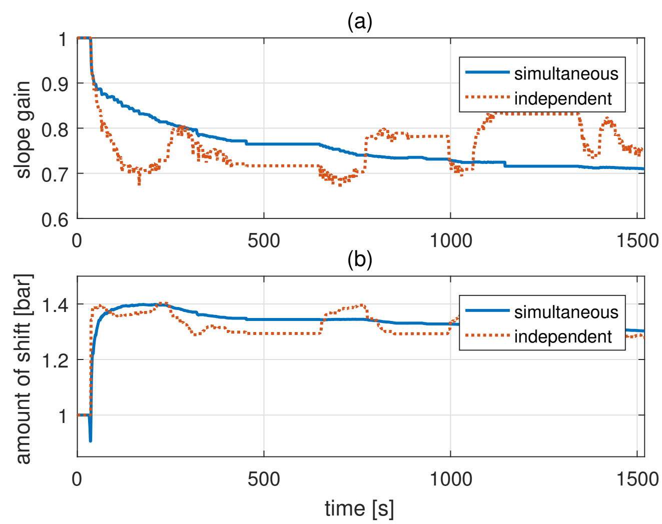

Fig. 15. Estimation result of the slope gain and the amount of parallel shift in the x-axis: (a) slope gain, (b) the amount of parallel shift.

the independent estimation method, since the scalar calculation is mainly performed, it took less memory, and the entire logic was very simple.

Considering the advantages of the independent estimation method, the torque tracking control in the slip engagement of the clutch was additionally performed using the independent estimation method of the slope gain and the amount of parallel shift in the x-axis of nominal TPTM.

D. Feedforward tracking control of clutch torque

In this section, the performance of the feedforward tracking control of the engine clutch torque is verified using the independent estimation method of the slope gain and the amount of parallel shift in the x-axis of nominal TPTM.

In the verification experiment, when the vehicle was stationary and the engine was idling, the slip engagement of the engine clutch was repeated 60 times or more for about 1500

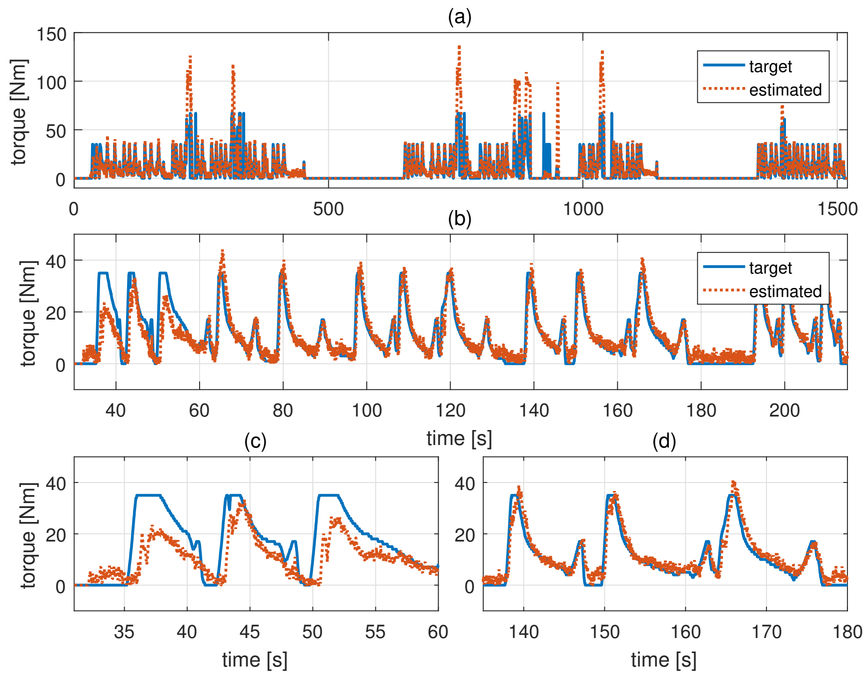

Fig. 16. Result of the feedforward tracking control of the engine clutch torque: (a) result of the torque tracking control over the entire period, (b) result of the torque tracking control over a specific experiment period, (c) result of the torque tracking control over a RMS error calculation period in the beginning of the experiment, (d) result of the torque tracking control over a RMS error calculation period in the middle of the experiment.

TABLE I
RMS TRACKING ERROR OF THE ENGINE CLUTCH TORQUE

Period	RMS error (Nm)
31-60 seconds [Fig. 15(c)]	10.2108
135-180 seconds [Fig. 15(d)]	3.745

seconds and the vehicle state information is shown in Fig. 13.

Fig. 16 shows the result of the feedforward tracking control of the engine clutch torque. Fig. 16(a), (b), (c), and (d) show the result of the torque tracking control over the entire period, result of the torque tracking control over a specific experiment period, result of the torque tracking control over a three times slip engagement period (RMS error calculation period) in the beginning of the experiment, and result of the torque tracking control over a three times slip engagement period (RMS error calculation period) in the middle of the experiment when the slope gain and the amount of parallel shift were converged and the torque tracking error was decreased, respectively.

Referring to Fig, 15 (c) and Fig. 16 (d), Table. I shows the RMS error between the target engine clutch torque and the estimated engine clutch torque over three times slip engagement periods in the beginning and middle of the experiment.

Referring to Table 1, it can be seen that the torque tracking performance was improved by more than 60% in the middle compared to in the beginning of the experiment, and the torque tracking performance was improved in a short time.

Also, referring to Fig. 15, it is confirmed that the estimated slope gain and amount of parallel shift did not diverge and showed appropriate values during dozens of times of the slip engagement.

In conclusion, the performance of the adaptive control method of the clutch torque in the slip engagement of a wet clutch was verified and it was confirmed that the clutch torque can be precisely controlled without a piston pressure sensor in a wet clutch.

VI. Conclusion

In this study, an adaptive control method of the clutch torque in the slip engagement of a wet clutch was proposed, which does not require the measurement of the piston pressure. In this method, the adaptation of TPTM was performed, and the clutch torque was controlled using compensated TPTM and nominal CPM in a feedforward manner. The uncertainty on TPTM was compensated using a slope gain and the amount of parallel shift in the x-axis, and the two parameters were estimated using a clutch torque estimator. And, simultaneous and independent estimation methods of the two parameters were introduced and compared with each other. Furthermore, it was verified that the performance of the clutch torque tracking control is considerably improved using the independent estimation method of the slope gain and the amount of parallel shift. The clutch torque tracking control using the simultaneous estimation method of the slope gain and the amount of parallel shift will be further studied in the future.

Acknowledgment

This research was partly supported by the Hyundai Motor Company, the National Research Foundation of Korea(NRF) grant funded by the Korea government(MSIP) (No. 2017R1A2B4004116), and the BK21+ program through the NRF funded by the Ministry of Education of Korea.

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