Adaptive control method of clutch torque during clutch slip engagement

Jinrap Park[1] and Seibum Choi[1]

***Abstract*— To precisely control clutch torque during clutch slip engagement, a clutch friction model is required. The clutch friction model refers to the actuator piston position to clutch torque relationship in the case of a dry clutch and the actuator piston pressure to clutch torque relationship in the case of a wet clutch. On the other hand, the clutch friction model is changed by uncertainties such as a clutch touch point and a clutch friction coefficient. Therefore, it is necessary to compensate the clutch friction model for precise clutch torque control. In this study, a new adaptive control method of the clutch torque is proposed and compared with a conventional method.**

I. INTRODUCTION

Clutch slip engagement means that a clutch is engaged when the clutch is slipping, and situations that require the clutch slip engagement often occur in a vehicle powertrain. However, when a clutch is slipping, clutch friction energy loss and vehicle jerk occur due to the clutch slip. This affects fuel efficiency and ride comfort of a vehicle. Here, the clutch friction energy loss and vehicle jerk are directly related to clutch transmitted torque. Therefore, the clutch torque needs to be precisely controlled when a clutch is slip-engaged.

In order to precisely control the clutch torque, it is necessary to measure the clutch torque. However, in a production vehicle, a torque sensor is not available due to cost and space issues. Therefore, the clutch torque in a production vehicle is feedforward-controlled rather than feedback-controlled. Fig. 1 shows a typical feedforward control structure of a clutch. As shown in Fig. 1, a clutch friction model is used for the feedforward control of a clutch. In the case of a dry clutch, an actuator piston position-to-clutch torque model is typically used, and in the case of a wet clutch, an actuator piston pressure-to-clutch torque model is used.

On the other hand, there are some uncertainties in the clutch friction model. In the case of a dry clutch, the parallel shift of the model in the x-axis occurs due to clutch wear and the change of clutch temperature, and additionally the change of hydraulic oil temperature if an actuator has a hydraulic line. And, in the case of a wet clutch, the parallel shift of the model in the x-axis occurs due to clutch wear. Also, no matter what type of clutch is, the model is weighted according to the change of a clutch friction coefficient. Due to the uncertainties of the clutch friction model, clutch torque error occurs during the clutch torque control. Thus, it is necessary to properly compensate for the uncertainties of the clutch friction model.

The adaptation of the clutch friction model in the feedforward control of the clutch torque can be performed by using a clutch torque estimator [1]–[4]. Thus, the estimation of the clutch torque should be dealt with together with the adaptation of the clutch friction model [5], [6].

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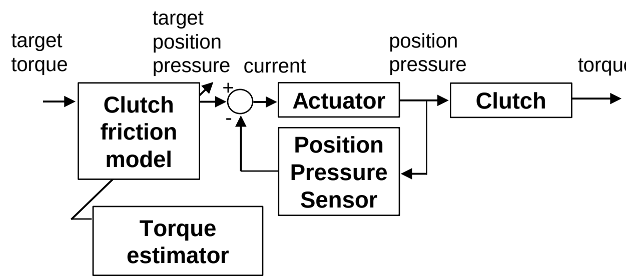

Fig. 1. Typical feedforward control structure of a clutch

In the previous study [7]–[9], the clutch friction model was represented by five parameters, and a method of estimating five parameters simultaneously using driveline speed information was proposed. The five parameters include a parameter that represents a clutch touch point and parameters that are related to the shape of the clutch friction model. Here, The touch point refers to the actuator piston position or pressure at which the clutch torque begins to be transmitted.

On the other hand, the clutch touch point has a great influence on clutch engagement feeling [10], [11]. Therefore, the estimate of the clutch touch point should be changed very slowly. However, when the clutch touch point and the shape parameters of the clutch friction model are estimated simultaneously, the change of the shape parameters affects the change of the clutch touch point. Therefore, if the estimation parameters of the clutch friction model are estimated simultaneously, the clutch touch point can be changed rapidly. In addition, physically, the shape of the clutch friction model does not affect the clutch touch point. Therefore, when the adaptation of the clutch friction model is conducted, the estimation of the clutch touch point and shape parameters needs to be performed independently.

In this study, an adaptation method of the clutch friction model is proposed which utilizes the amount of parallel shift of the clutch friction model and a slope gain which is a weighting gain of the clutch friction model [12]. Here, the amount of parallel shift of the clutch friction model is related to the clutch touch point, and the slope gain is related to the clutch friction coefficient. Also, in this study, a method is proposed which estimates the amount of parallel shift and the slope gain independently and the proposed method is compared with a simultaneous estimation method of two parameters. The feedforward control method of the clutch torque based on the mentioned adaptation methods is verified

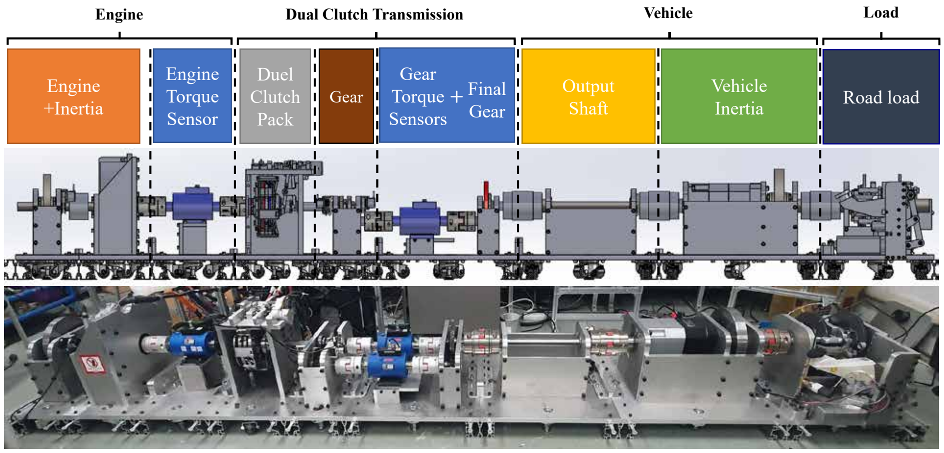

Fig. 2. Structure of a DCT test bench

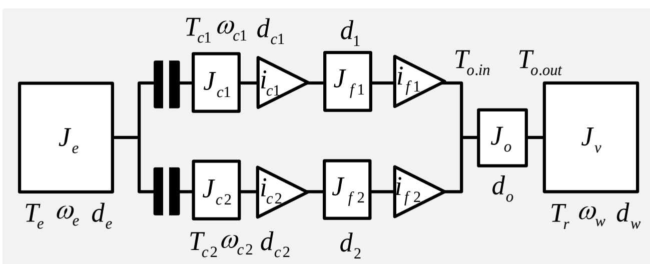

Fig. 3. Lumped inertia driveline model of the DCT test bench

and compared through clutch slip engagement experiments of the vehicle launch situation using a test bench equipped with a dry dual clutch transmission(DCT).

This paper is organized as follows. Section 2 introduces the clutch torque estimation method of DCT equipped in the test bench. Section 3 addresses the adaptation method of the clutch friction model and the feedforward control method of the clutch torque. Section 4 mentions experimental results. And, section 5 concludes this paper.

II. Clutch Torque Estimation

A. Driveline model

Fig. 2 shows the structure of a DCT test bench used in this study, and Fig. 3 shows the lumped inertia driveline model of the DCT test bench. A drive motor of a parallel hybrid vehicle is utilized as a torque source in the DCT test bench. Hereinafter, the drive motor is referred as an engine.

Referring to Fig. 3, modeling the driveline of the test bench leads to the following equations.

$$\dot{\omega}_e = -\frac{1}{J_e} d_e \omega_e + \frac{1}{J_e} T_e - \frac{1}{J_e} T_{c1} - \frac{1}{J_e} T_{c2} \tag{1}$$

$$J_{c1}\dot{\omega}_{c1} = -d_{c1}\omega_{c1} + T_{c1} - \frac{T_{f1}}{i_{c1}} \tag{2}$$

$$J_{c2}\dot{\omega}_{c2} = -d_{c2}\omega_{c2} + T_{c2} - \frac{T_{f2}}{i_{c2}} \tag{3}$$

$$J_{f1}\dot{\omega}_{f1} = -d_{f1}\omega_{f1} + T_{f1} + \frac{i_{f2}}{i_{f1}} T_{f2} - \frac{T_{o.in}}{i_{f1}} \tag{4}$$

$$J_{f2}\dot{\omega}_{f2} = -d_{f2}\omega_{f2} + \frac{i_{f1}}{i_{f2}} T_{f1} + T_{f2} - \frac{T_{o.in}}{i_{f2}} \tag{5}$$

$$J_o\dot{\omega}_o = -d_o\omega_o + T_{o.in} - T_{o.out} \tag{6}$$

$$J_v\dot{\omega}_w = -d_w\omega_w + T_{o.out} - T_r \tag{7}$$

$$T_{c1} = f(p_1) \tag{8}$$

$$T_{c2} = f(p_2) \tag{9}$$

$$T_o = k_o\left(\frac{\theta_{c1}}{i_{c1}i_{f1}} - \theta_w\right) + b_o\left(\frac{\omega_{c1}}{i_{c1}i_{f1}} - \omega_w\right) \tag{10}$$

Where, T, J, θ, ω, i, d, p, k, and b represent torque, inertia moment, rotational angle, rotational speed, gear ratio, shaft damping resistance constant, actuator piston position, shaft compliance spring constant, and shaft compliance damping constant, and the subscript e, $c1$, $c2$, $f1$, $f2$, $o.in$, $o.out$, w, v, r, 1, and 2 mean the engine, odd gear clutch, even gear clutch, odd gear final shaft, even gear final shaft, output shaft input, output shaft output, wheel, vehicle, road load, odd gear clutch actuator, and even gear clutch actuator.

In a real vehicle, the engine speed, the two clutch shaft speeds of DCT, and the wheel speed are usually measurable and can be measured in the same position on the test bench. Therefore, considering the position where the speed can be measured, the above equations can be simplified as follows.

$$\dot{\omega}_e = -\frac{1}{J_e} d_e \omega_e + \frac{1}{J_e} T_e - \frac{1}{J_e} T_{c1} - \frac{1}{J_e} T_{c2} \tag{11}$$

$$J_{eq1}\dot{\omega}_{c1} = -d_{eq1}\omega_{c1} + i_{c1}i_{f1}T_{c1} + i_{c2}i_{f2}T_{c2} - T_{o.out} \tag{12}$$

$$\begin{aligned} J_{eq1} &= i_{c1}i_{f1}J_{c1} + \frac{(i_{c2}i_{f2})^2}{i_{c1}i_{f1}} J_{c2} \\ &\quad + \frac{i_{f1}}{i_{c1}} J_{f1} + \frac{i_{f2}^2}{i_{c1}i_{f1}} J_{f2} + \frac{1}{i_{c1}i_{f1}} J_o \end{aligned} \tag{13}$$

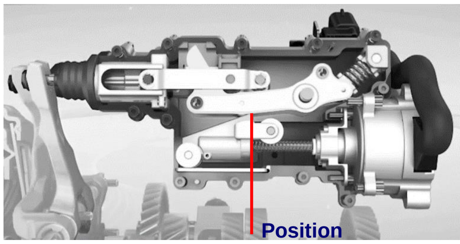

Fig. 4. Clutch actuator structure of dry DCT

$$\begin{aligned} d_{eq1} &= i_{c1}i_{f1}d_{c1} + \frac{(i_{c2}i_{f2})^2}{i_{c1}i_{f1}}d_{c2} \\ &+ \frac{i_{f1}}{i_{c1}}d_{f1} + \frac{i_{f2}^2}{i_{c1}i_{f1}}d_{f2} + \frac{1}{i_{c1}i_{f1}}d_o \end{aligned} \tag{14}$$

$$J_{eq2}\dot{\omega}_{c2} = -d_{eq2}\omega_{c2} + i_{c1}i_{f1}T_{c1} + i_{c2}i_{f2}T_{c2} - T_{o.out} \tag{15}$$

$$\begin{aligned} J_{eq2} &= \frac{(i_{c1}i_{f1})^2}{i_{c2}i_{f2}}J_{c1} + i_{c2}i_{f2}J_{c2} \\ &+ \frac{i_{f1}^2}{i_{c2}i_{f2}}J_{f1} + \frac{i_{f2}}{i_{c2}}J_{f2} + \frac{1}{i_{c2}i_{f2}}J_o \end{aligned} \tag{16}$$

$$\begin{aligned} d_{eq2} &= \frac{(i_{c1}i_{f1})^2}{i_{c2}i_{f2}}d_{c1} + i_{c2}i_{f2}d_{c2} \\ &+ \frac{i_{f1}^2}{i_{c2}i_{f2}}d_{f1} + \frac{i_{f2}}{i_{c2}}d_{f2} + \frac{1}{i_{c2}i_{f2}}d_o \end{aligned} \tag{17}$$

$$J_v\dot{\omega}_w = -d_w\omega_w + T_{o.out} - T_r \tag{18}$$

Where, the subscript $eq1$, and $eq2$ represent the equivalent inertia from the odd gear clutch perspective, and the equivalent inertia from the even gear clutch perspective.

B. Clutch torque estimator

Since only an odd clutch is used when a vehicle launches, the even clutch torque is zero. Therefore, an estimator for the odd clutch torque is designed as follows.

$$\dot{\hat{\omega}}_e = -\frac{1}{J_e}d_e\omega_e + \frac{1}{J_e}T_e - \frac{1}{J_e}\hat{T}_{c1} \tag{19}$$

$$\dot{\hat{\omega}}_{c1} = -\frac{1}{J_{eq1}}d_{eq1}\omega_{c1} + \frac{1}{J_{eq1}}i_{c1}i_{f1}\hat{T}_{c1} - \frac{1}{J_{eq1}}\hat{T}_{o.out} \tag{20}$$

$$\dot{\hat{\omega}}_w = -d_w\frac{1}{J_v}\omega_w + \frac{1}{J_v}\hat{T}_{o.out} - \frac{1}{J_v}\hat{T}_r \tag{21}$$

$$\dot{\hat{T}}_{c1} = -d_{e1}\hat{T}_{c1} - l_1(\omega_e - \hat{\omega}_e) + l_2(\omega_{c1} - \hat{\omega}_{c1}) \tag{22}$$

$$\begin{aligned} \dot{\hat{T}}_{o.out} &= -d_{e2}\hat{T}_{o.out} + k_o\left(\frac{\omega_{c1}}{i_{c1}i_{f1}} - \omega_w\right) \\ &-l_3(\omega_{c1} - \hat{\omega}_{c1}) + l_4(\omega_w - \hat{\omega}_w) \end{aligned} \tag{23}$$

$$\dot{\hat{T}}_r = -d_{e3}\hat{T}_r + l_5(\omega_w - \hat{\omega}_w) \tag{24}$$

Where, $\hat{}$ represent an estimated value, and l_1, l_2, l_3, l_4, l_5, d_{e1}, d_{e2}, and d_{e3} are tuning parameters

The stability check of the above estimator is omitted since it is rather long.

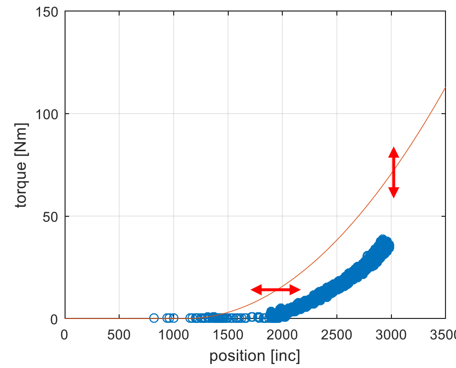

Fig. 5. Nominal clutch friction model and estimated clutch torque according to motor position

III. Clutch Friction Model Adaptation

Fig. 4 shows the clutch actuator structure of dry DCT used in this study. In this clutch actuator, rotational position of an actuator motor could be measured. Therefore, the clutch friction model in this study means the relationship between motor position and clutch torque.

Also, in this study, a method to compensate the clutch friction model is proposed using the amount of parallel shift in the x-axis related to the clutch touch point and the slope gain related to the clutch friction coefficient. In addition, the independent estimation method of these two parameters is proposed and compared with the simultaneous estimation method. Hereinafter, the amount of parallel shift in the x-axis is referred to as the amount of parallel shift.

Fig. 5 shows a nominal clutch friction model and the estimated clutch torque according to the motor position when the clutch is slip-engaged. In this study, the difference between the touch point of the nominal clutch friction model and the touch point of a real model was made to verify the adaptive control method.

Estimation of the amount of parallel shift and the slope gain of the clutch friction model mentioned below is performed under the following clutch slip condition.

$$c_1 < p < c_2 \tag{25}$$

$$c_3 < |\omega_e - \omega_m| \tag{26}$$

$$c_4 < \hat{T}_{c1} \tag{27}$$

Where, c_1, c_2, c_3, and c_4 are threshold constants.

A. Simultaneous estimations

In this subsection, the simultaneous estimation method of the amount of parallel shift and the slope gain of the clutch friction model is introduced. In this study, a recursive least square (RLS) technique is used in the simultaneous estimation method.

Clutch torque can be expressed as follows by using the nominal clutch friction model function, the amount of parallel shift, and the slope gain.

$$T_{c1} = \alpha f(p - \beta) \tag{28}$$

Linearizing the clutch torque according to the motor position, the following equations can be derived.

$$T_{c1} = \alpha f(p-\beta) = g(\alpha,\beta,p) \tag{29}$$

$$\begin{aligned}&\frac{dg}{d\alpha}(\alpha_{k-1},\beta_{k-1},p)\alpha_k + \frac{dg}{d\beta}(\alpha_{k-1},\beta_{k-1},p)\beta_k \\ &= T_{c1.k} - g(\alpha_{k-1},\beta_{k-1},p) \\ &+\frac{dg}{d\alpha}(\alpha_{k-1},\beta_{k-1},p)\alpha_{k-1} + \frac{dg}{d\beta}(\alpha_{k-1},\beta_{k-1},p)\beta_{k-1}\end{aligned} \tag{30}$$

$$\begin{aligned}&z_k = T_{c1.k} - g(\alpha_{k-1},\beta_{k-1},p) \\ &+\frac{dg}{d\alpha}(\alpha_{k-1},\beta_{k-1},p)\alpha_{k-1} + \frac{dg}{d\beta}(\alpha_{k-1},\beta_{k-1},p)\beta_{k-1} \\ &= \begin{pmatrix} \frac{dg}{d\alpha}(\alpha_{k-1},\beta_{k-1},p) & \frac{dg}{d\beta}(\alpha_{k-1},\beta_{k-1},p) \end{pmatrix} \begin{pmatrix} \alpha_k \\ \beta_k \end{pmatrix}\end{aligned} \tag{31}$$

Where, α, β, and k represent the slope gain, the amount of parallel shift, and time steps.

Then, if estimation parameters are defined as below, the slope gain and the amount of parallel shift can be estimated simultaneously as follows using the RLS technique.

$$x_k = \begin{pmatrix} \alpha_k \\ \beta_k \end{pmatrix} \tag{32}$$

$$H_k^T = \begin{pmatrix} \frac{dg}{d\alpha}(\alpha_{k-1},\beta_{k-1},p) \\ \frac{dg}{d\beta}(\alpha_{k-1},\beta_{k-1},p) \end{pmatrix} \tag{33}$$

$$K_k = P_{k-1}H_k^T(H_kP_{k-1}H_k^T + R_k)^{-1} \tag{34}$$

$$P_k = (I - K_kH_k)P_{k-1}(I-K_kH_k)^T + K_kR_kK_k^T \tag{35}$$

$$\hat{x}_k = \hat{x}_{k-1} + K_k(z_k - H_k\hat{x}_{k-1}) \tag{36}$$

B. Independent estimation

In this subsection, the independent estimation method of the amount of parallel shift and the slope gain of the nominal clutch friction model is introduced.

First, an intermediate clutch touch point is estimated by moving-averaging the estimated clutch torque which is within a specific torque band.

$$c_5 < \hat{T}_{c1} < c_6 \tag{37}$$

$$\hat{\theta}_{k+1} = \frac{n}{n+1}\hat{\theta}_k + \frac{1}{n+1}p_{k+1} \tag{38}$$

Where, c_5, and c_6 are threshold constants, and n, θ, and k represent moving average width, the intermediate clutch touch point, time steps during the clutch slip engagement.

In order to quickly estimate the touch point and reduce the change of the touch point, the final clutch touch point is estimated by moving-averaging the intermediate clutch touch point at each end of the clutch slip engagement.

$$\hat{\lambda}_{i+1} = \frac{m}{m+1}\hat{\lambda}_i + \frac{1}{m+1}\hat{\theta}_{i+1} \tag{39}$$

Where, m, λ, and i represent moving average width, final clutch touch point, and the number of clutch slip engagement.

Using the clutch touch point of the nominal clutch friction model, the parallel shift of the nominal clutch friction model can be estimated as follows.

$$\hat{\beta} = \hat{\lambda} - \lambda_n \tag{40}$$

Where, λ_n means the touch point of the nominal clutch friction model.

The slope gain of the nominal clutch friction model is estimated as follows using the estimated amount of parallel shift.

$$\hat{T}_{c1} = \hat{\alpha}f(p-\hat{\beta}) \tag{41}$$

$$\dot{\hat{\alpha}} = -\gamma_\alpha\varepsilon \tag{42}$$

Where, γ_α is a tuning parameter.

The stability of the above estimator is verified by the Lyapunov stability criterion as follows.

$$\varepsilon = \hat{T}_{c1} - T_{c1} = (\hat{\alpha}-\alpha)f(p-\hat{\beta}) = \tilde{\alpha}f(p-\hat{\beta}) \tag{43}$$

$$\dot{\tilde{\alpha}} = -\gamma_\alpha\tilde{\alpha}f(p-\hat{\beta}) \tag{44}$$

$$V = \frac{1}{2\gamma_a}\tilde{\alpha}^2 > 0 \ (\gamma_\alpha > 0) \tag{45}$$

$$\dot{V} = -\tilde{\alpha}^2 f(p-\hat{\beta}) \ < 0 \ \{f(p-\hat{\beta}) > 0\} \tag{46}$$

C. Clutch torque feedforward control

The feedfoward control of clutch torque is performed by utilizing the estimated amount of parallel shift and the slope gain of the nominal clutch friction model. Inverting equation (41) can be expressed as follows

$$p = f^{-1}(\frac{\hat{T}_{c1}}{\hat{\alpha}}) + \hat{\beta} \tag{47}$$

Then, if the estimated clutch torque is replaced with the target clutch torque, the target motor position can be derived as follows.

$$p_t = f^{-1}(\frac{T_{c1.t}}{\hat{\alpha}}) + \hat{\beta} \tag{48}$$

Where, the subscript t represents a target value.

IV. EXPERIMENTAL RESULTS

A. Clutch torque estimation

In order to verify the performance of the clutch torque estimator used in this study, an experiment was conducted to repeatedly engage the odd gear clutch of DCT 5 times with the engine speed maintained at 500 RPM like a vehicle launch situation.

Fig. 6 shows the driveline speed and target motor position during the verification experiment of the clutch torque estimator and Fig. 7 shows the clutch torque estimation result during the verification experiment of the clutch torque estimator.

The RMS error between the estimated clutch torque and measured clutch torque in this study was 1.0997 Nm. In the previous study [12], the RMS error of clutch torque estimation in a production vehicle was about 10 Nm and the performance of the clutch torque estimator used in this study was confirmed to be good.

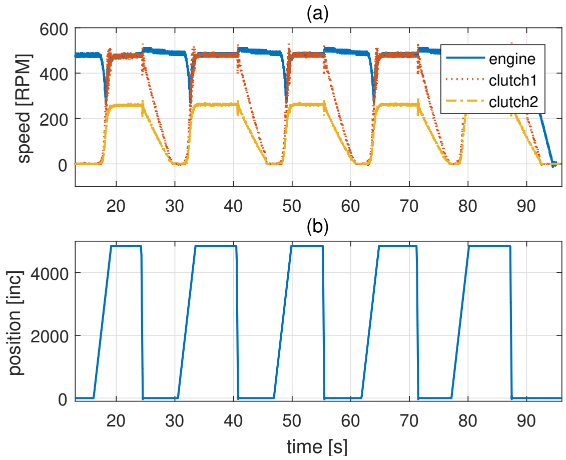

Fig. 6. Driveline speed and target motor position during the verification experiment of the clutch torque estimator

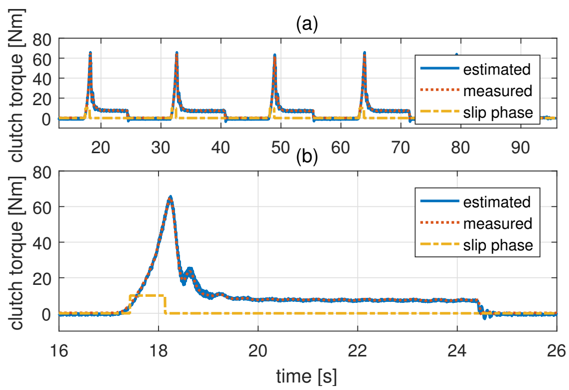

Fig. 7. Clutch torque estimation result during the verification experiment of the clutch torque estimator

B. Clutch friction model adaptation and clutch torque feed-forward control

In order to verify and compare the independent and simultaneous estimation methods of the amount of parallel shift and the slope gain of the nominal clutch friction model, the clutch torque control experiments were conducted using each method.

In the clutch torque control experiments, like the verification experiment of the clutch torque estimator, the odd gear clutch of DCT was engaged 10 times with the engine speed maintained at 500 RPM.

In addition, referring to Fig. 5, the nominal clutch friction model in the clutch torque control experiments was moved about 700 incremental positions to the left in the x-axis direction compared to the actual clutch friction model, and the slope of the nominal clutch friction model was not changed.

Fig. 8 shows measurement information in the clutch torque control experiment using the independent estimation method. Fig. 8(a), (b), (c), and (d) shows driveline speed and slip phase, driveline torque, the target clutch torque, and the target motor position, respectively. To avoid repetition, the figure

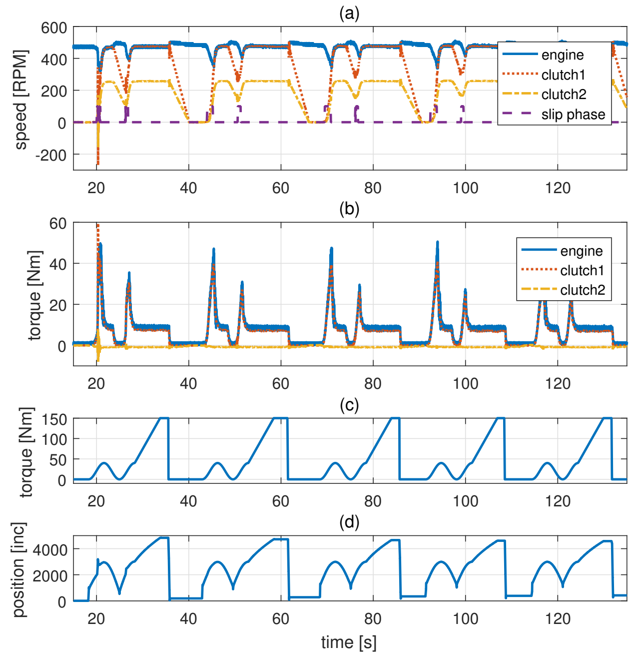

Fig. 8. Measurement information in the clutch torque control experiment using the simultaneous estimation method: (a) driveline speed and slip phase, (b) driveline torque, (c) target clutch torque, (d) target motor position

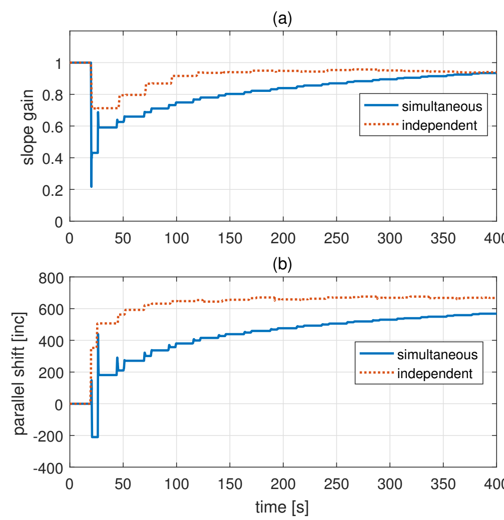

Fig. 9. Adaptation result of the clutch friction model: (a) estimation result of slope gain, (b) estimation result of the amount of parallel shift

of measurement information in the clutch torque control experiment using the simultaneous estimation method was omitted.

Fig. 9 shows the adaptation result of the clutch friction model. Fig. 9 (a), and (b) show the estimation result of slope gain, and the estimation result of the amount of parallel shift, respectively.

In Fig. 9, when the amount of parallel shift and the slope gain were estimated by the simultaneous estimation method using the RLS technique, the amount of parallel shift and the slope gain were greatly decreased when the clutch slip was first started. From this result, in the simultaneous

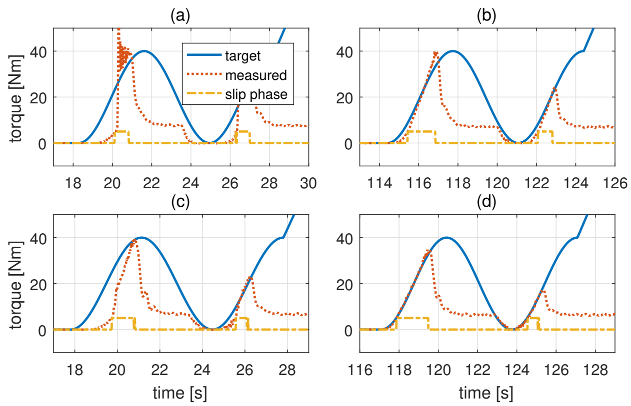

Fig. 10. Results of the clutch torque control: (a) result of the clutch torque control in the first slip engagement using the simultaneous estimation method, (b) result of the clutch torque control in the fifth slip engagement, (c) result of the clutch torque control in the first slip engagement using the in the independent estimation method, (d) result of the clutch torque control in the fifth slip engagement

estimation method, it can be seen that the estimation error of the slope gain affects the estimation error of the amount of parallel shift. This is inconsistent with the actual physical phenomenon and can adversely affect the performance of the clutch torque control. In addition, considering the actual amount of parallel shift and slope gain were 700 and 1, respectively, the two parameters were estimated rather slowly.

On the other hand, in Fig. 9, when the amount of parallel shift and the slope gain were estimated by the independent estimation method, the amount of parallel shift was not decreased as the slope gain was decreased. This means that the estimation error of the slope gain does not affect the estimation error of the amount of parallel shift. This is consistent with the actual physical phenomenon. In addition, the amount of parallel shift and the slope gain were estimated fast, compared to the simultaneous estimation.

Fig. 10 shows the results of the clutch torque control using the simultaneous and independent estimation methods. Fig. 10(a), and (b) show the result of the clutch torque control in the first slip engagement using the simultaneous estimation method, and the result of the clutch torque control in the fifth slip engagement, respectively. Fig. 10(c), and (d) show the result of the clutch torque control in the first slip engagement using the independent estimation method, and the result of the clutch torque control in the fifth slip engagement, respectively.

In the case of using the simultaneous estimation method in Fig. 10 (a), the clutch torque oscillated greatly as the amount of parallel shift and the slope gain were greatly decreased when the clutch slip was first started. And, in Fig. 10 (b), it can be seen that the performance of the clutch torque control was not good when the clutch was repeatedly engaged because the estimation speed of the amount of parallel shift and the slope gain was slow.

On the other hand, in the case of using the independent estimation method in Fig. 10 (c), the clutch torque did not oscillate greatly as the amount of parallel shift and the slope gain were estimated rapidly when the clutch slip was first started. In addition, in Fig. 10 (d), it can be seen that the performance of the clutch torque control was good when the clutch was repeatedly engaged because the estimation speed of the amount of parallel shift and the slope gain was fast.

TABLE I

RMS ERROR OF CLUTCH TORQUE

Method	RMS error (Nm)
Simultaneous	4.5340
Indepenent	3.9760

Table I shows the RMS error of the clutch torque during the first five slip engagements using each method. The RMS error of the clutch torque during the first five slip engagements was about 10% smaller when the clutch torque was controlled using the independent estimation method than when the clutch torque was controlled using the simultaneous estimation method. This means that the adaptation of the clutch friction model is performed quickly if the independent estimation method is used.

V. CONCLUSIONS

In this study, a new adaptive control method of the clutch torque was proposed and compared with a conventional method. It was confirmed that the performance of the clutch torque control is improved when using the independent estimation method of the compensation parameters of the clutch friction model, compared with when using the simultaneous estimation method. In the future, a shape estimation method of the clutch friction model will be studied.

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