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Adaptive slip engagement control of a dry clutch using a clutch torque estimator

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Abstract: A dry clutch is an important part in a vehicle powertrain system and the dry clutch is primarily utilized for small and medium sized vehicles. The slip engagement is a method of engaging a clutch in a state where the speeds of both sides of the clutch are different. When a clutch is slipping, energy loss due to clutch friction, and clutch wear occurs. Therefore, it is necessary to control a clutch precisely in slip engagement of a clutch. A clutch friction model is needed to implement feedforward control in clutch slip engagement control, but there are some uncertain parameters in a clutch friction model. In this study, a clutch friction coefficient and a clutch touch point are represented as uncertain parameters of a clutch friction model. Thus, a simultaneous adaptation method of a clutch friction coefficient and a clutch touch point in a clutch friction model, and a feedforward control method of slip engagement of a dry clutch using the adaptation method are proposed in this study. The proposed adaptive control algorithm was verified with the slip engagement simulation of a dry engine clutch in a parallel hybrid vehicle in the MATLAB/Simulink environment.

Keywords: dry clutch control, slip engagement control, clutch friction model, adaptive clutch control.

1. INTRODUCTION

As electronic systems are applied to automobiles, precise control of automobile systems is being actively carried out. As a result, vehicle performance and ride comfort have been greatly improved in recent years. On the other hand, drivers' expectations for vehicle performance and ride comfort continue to rise due to competition in the automobile market.

Slip engagement control of a clutch has a great effect on vehicle performance and ride comfort, and it is important in vehicle powertrain control. The slip engagement of a clutch always occurs when a clutch engages in situations where the speed of both sides of the clutch is different. For example, the slip engagement occurs in gear shifting of a transmission. Also, an engine clutch in the case of a parallel hybrid vehicle can be engaged while slipping. However, the slip engagement of a clutch can cause friction energy loss and vehicle jerk. Thus, precise slip engagement control should be needed to reduce the friction energy loss and vehicle jerk.

Slip engagement control is divided into two methods depending on whether a target control variable is clutch slip speed or clutch torque [1-4]. However, a clutch transmits torque, and the torque is directly related with vehicle performance and ride comfort. So, many studies have tried slip engagement control based on clutch torque.

But, a torque sensor is bulky and expensive so it cannot be installed in a production vehicle. Thus, clutch torque cannot be measured generally. For this reason, many studies have designed a clutch torque estimator and it was utilized for a torque sensor instead [5-7].

However, even if clutch torque can be estimated, it can be rather dangerous to utilize the estimated torque for feedback control of a clutch because estimation error always exists. Thus, some studies utilized the estimated torque to compensate a clutch friction model, which is the relationship between actuator input and clutch torque, and proposed an adaptive feedforward control method of a clutch [8-10].

A dry clutch, and a wet clutch are generally utilized for small or medium sized vehicles, and medium or large sized vehicles, respectively. In this study, slip engagement control of a dry clutch is addressed.

There can be two uncertain parameters in a friction model of a dry clutch: a clutch friction coefficient, and a clutch touch point. Most of previous studies have focused on the adaptation of a clutch friction coefficient only, assuming a clutch touch point does not change [1, 2, 11]. However, a clutch touch point can be changed due to clutch wear or hydraulic pressure change in a clutch

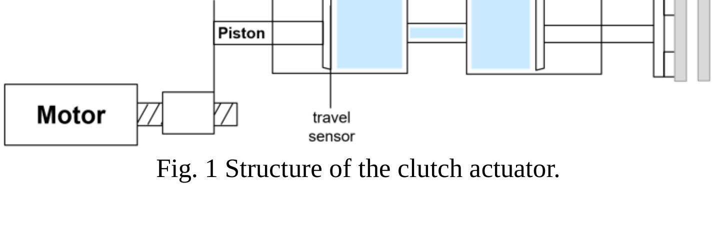

Fig. 1 Structure of the clutch actuator.

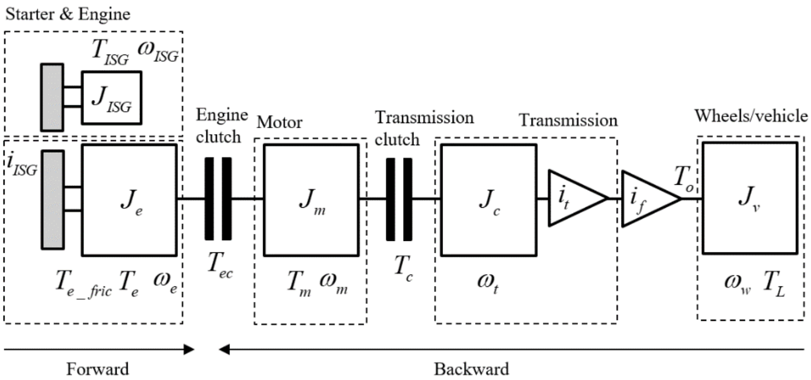

Fig. 2 Lumped inertia model of a parallel hybrid vehicle.

actuation system, in which hydraulic line exists [12-14]. Thus, a clutch friction coefficient and a clutch touch point are needed to be estimated simultaneously for precise slip control of a dry clutch.

Thus, in this paper, some simultaneous adaptation techniques of a clutch friction coefficient and a clutch touch point using a clutch torque estimator are introduced. Also, an adaptive feedforward control method is proposed using the adaptation techniques. Furthermore, the control method is applied for the slip engagement control of a dry engine clutch between an engine and a motor in a parallel hybrid vehicle. The pros and cons of each adaptation method are analyzed through the slip engagement control simulation in the MATLAB/Simulink environment.

This paper is organized as follows. Section 2 introduces adaptation techniques of a clutch friction coefficient and a clutch touch point using a clutch torque estimator, and an adaptive feedforward control method of a dry clutch. Section 3 shows the simulation results of slip engagement control of an engine clutch. Section 4 concludes this paper.

2. ADAPTATION ALGORITHM

2.1 Clutch torque estimator

The adaptation techniques of a clutch friction model introduced in this study are applied for slip engagement control of a dry engine clutch of a parallel hybrid vehicle, and the information of clutch torque and actuator input is needed for the adaptation of a clutch friction coefficient and a clutch touch point.

In this paper, the structure of an engine clutch actuator is assumed to be like Fig. 1. Also, the actuator input is considered to be measured and it is actuator piston position, which is measured by a travel sensor.

Furthermore, the transmitted torque of the engine clutch is considered not to be measured, so it should be estimated using the driveline model of a parallel hybrid vehicle. In this study, the driveline model of a parallel hybrid vehicle is modeled as follows using a lumped inertia model like Fig. 2.

$$T_e - T_{ec} = J_e \dot{\omega}_e \qquad (1)$$

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

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

$$T_o - T_L = J_v \dot{\omega}_w \qquad (4)$$

where T, J, and ω denote the torque, inertia, and angular speed, and the subscript e, ec, m, c, o, L, v, and w denote the engine, engine clutch, motor, transmission input shaft, output shaft, load resistance, vehicle, and wheel. Also, i_t, and i_f mean the transmission gear ratio, and final gear ratio

Also, a simple unknown input observer of the forward path to estimate the engine clutch torque is designed like below.

$$\dot{\hat{\omega}}_e = \frac{1}{J_e}(T_e - \hat{T}_{ec}) \qquad (5)$$

$$\dot{\hat{T}}_{ec} = l_1(\omega_e - \hat{\omega}_e) \qquad (6)$$

where l_1 denote an unknown input observer gain.

The performance of the above unknown input observer have been verified in other studies[6, 15, 16], so the process of performance verification is omitted in this paper.

Some amount of Gaussian random noise is added to engine torque, considering the uncertainty of the engine torque in a real vehicle. Then, the clutch friction model, which is the relationship between the actuator piston position and the clutch torque, can be estimated like Fig. 3. The red line is a nominal clutch friction model, and the blue dots are the estimated friction model data.

The slope of a clutch friction model is directly related with the clutch friction coefficient and the clutch touch point is directly related with the amount of parallel shift of the friction model in the actuator piston position axis.

Thus, a clutch friction model is defined using a slope gain, the amount of parallel shift, and a nominal friction model as follow.

$$T_{ec} = \alpha T_{ec.n}(d - \beta) \qquad (7)$$

where $T_{ec.n}$, T_{ec}, d, α, and β denote a nominal friction model, an actual friction model, actuator piston position, a slope gain of a nominal friction model, the amount of parallel shift of a nominal friction model.

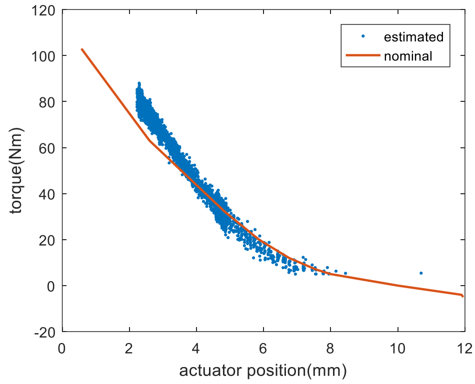

Fig. 3 Clutch friction model.

Then, the estimated clutch friction model can be expressed as follow using estimates of the slope gain and the amount of parallel shift.

$$\hat{T}_{ec} = \hat{\alpha} T_{ec.n}(d - \hat{\beta}) \qquad (8)$$

where $\hat{T}_{ec}$, $\hat{\alpha}$, and $\hat{\beta}$ denote the estimated friction model, the estimated slope gain of a nominal friction model, and the estimated amount of parallel shift of a nominal friction model.

The actual clutch friction coefficient and the clutch touch point can be expressed as follows.

$$\mu_r = \hat{\alpha}\mu_n \qquad (9)$$

$$\lambda_r = \lambda_n + \hat{\beta} \qquad (10)$$

where μ_r, μ_n, λ_r, and λ_n denote the actual friction coefficient of a clutch, nominal friction coefficient, actual touch point of a clutch, nominal touch point.

Thus, the estimation of the clutch friction coefficient and the clutch touch point can be considered as the same problem with the estimation of the slope gain and the amount of parallel shift in actuator piston position axis of a nominal friction model compared with the estimated model.

In the next subsections, the estimation techniques of the slope gain and the amount of parallel shift are introduced and compared with each other.

Also, in this study, it is assumed that the time a clutch is moved in the opening direction during clutch engagement is very small, so the hysteresis of a clutch friction model during engagement is not considered.

2.2 Recursive least square method

An adaptation method using a modified Recursive Least Square (RLS) method is introduced in this subsection. Referring to the equation (7), the relationship between the engine clutch torque, and the slope gain and the amount of parallel shift is nonlinear. Thus, the equation of the engine clutch should be linearized at an initial point of the slope gain and the amount of parallel shift.

However, the initial point of the slope gain and the amount of parallel shift for linearization affects the convergence error of estimates because the linearized model may not fit well with the original model when away from the linearized point.

Thus, a modified nonlinear algorithm which updates the linearized point at each time step is designed as follows.

$$\hat{\mathbf{x}} = \begin{pmatrix} \hat{\alpha} \\ \hat{\beta} \end{pmatrix} \qquad (10)$$

$$z_k = \hat{T}_{ec.k} - \hat{\alpha}_k T_{ec.n}(d_k - \hat{\beta}_k) \qquad (11)$$

$$\mathbf{H}_k{}^T = \begin{pmatrix} \dfrac{\partial T_{ec}}{\partial \alpha}(\hat{\mathbf{x}}_{k-1}, d_k) \\ \dfrac{\partial T_{ec}}{\partial \beta}(\hat{\mathbf{x}}_{k-1}, d_k) \end{pmatrix} = \begin{pmatrix} T_{ec.n}(d_k - \hat{\beta}_{k-1}) \\ -\hat{\alpha}_{k-1}\dfrac{\partial T_{ec.n}}{\partial d_k}(d_k - \hat{\beta}_{k-1}) \end{pmatrix} \qquad (12)$$

$$\mathbf{K}_k = \mathbf{P}_{k-1}\mathbf{H}_k{}^T(\mathbf{H}_k\mathbf{P}_{k-1}\mathbf{H}_k{}^T + R_k)^{-1} \qquad (13)$$

$$\mathbf{P}_k = (\mathbf{I} - \mathbf{K}_k\mathbf{H}_k)\mathbf{P}_{k-1}(\mathbf{I} - \mathbf{K}_k\mathbf{H}_k)^T + \mathbf{K}_k R_k \mathbf{K}_k{}^T \qquad (14)$$

$$\hat{\mathbf{x}}_k = \hat{\mathbf{x}}_{k-1} + \mathbf{K}_k z_k \qquad (15)$$

2.3 Gradient adaptation method

In this subsection, another adaptation algorithm, a gradient adaptation method is introduced. This algorithm is designed to reduce the computational load, removing complex matrix computation as follows.

$$\hat{T}_{ec.k} = T_{ec}(\hat{\mathbf{x}}_{k-1}, d_k) + \frac{\partial T_{ec}}{\partial \alpha}(\hat{\mathbf{x}}_{k-1}, d_k)\Delta\hat{\alpha}_{k-1} + \frac{\partial T_{ec}}{\partial \beta}(\mathbf{x}_{k-1}, d_k)\Delta\hat{\beta}_{k-1} \qquad (16)$$

$$\varepsilon = \begin{pmatrix} \Delta\hat{\alpha}_{k-1} & \Delta\hat{\beta}_{k-1} \end{pmatrix} \begin{pmatrix} \dfrac{\partial T_{ec}}{\partial \alpha}(\hat{\mathbf{x}}_{k-1}, d_k) \\ \dfrac{\partial T_{ec}}{\partial \beta}(\hat{\mathbf{x}}_{k-1}, d_k) \end{pmatrix} = \Delta\hat{\mathbf{x}}_{k-1}{}^T\mathbf{u} \qquad (17)$$
$$= \hat{T}_{ec.k} - T_{ec}(\hat{\mathbf{x}}_{k-1}, d_k)$$

$$\Delta\hat{\mathbf{x}}_k = -\gamma\varepsilon u \qquad (18)$$

$$\hat{\mathbf{x}}_k = \hat{\mathbf{x}}_{k-1} + \Delta\hat{\mathbf{x}}_k \qquad (19)$$

2.4 Feedforward control method

In this subsection, the feedforward control method using the estimated slope gain and the amount of parallel shift of a nominal friction model is introduced.

Referring to the equation (7), the relationship of the reference actuator piston position and clutch torque using the estimated slope gain and the amount of the parallel shift is as follow.

$$T_{ec.r} = \hat{\alpha} T_{ec.n}(d_r - \hat{\beta}) \qquad (20)$$

Then, the reference position can be calculated using the inverse function of the above equation as follow.

$$d_r = T_{ec.n}{}^{-1}\left(\frac{1}{\hat{\alpha}} T_{ec.r}\right) + \hat{\beta} \qquad (21)$$

Then, the feedback control of the actuator piston position can be performed using the above reference

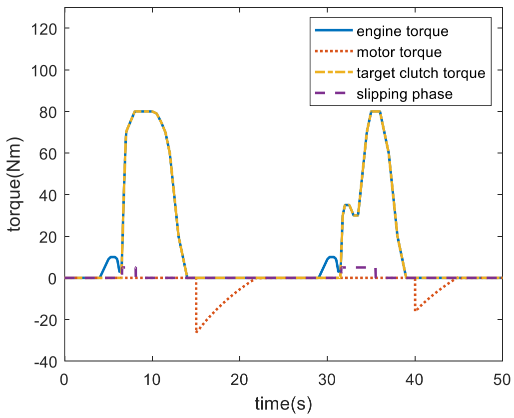

Fig. 4 Driveline torque.

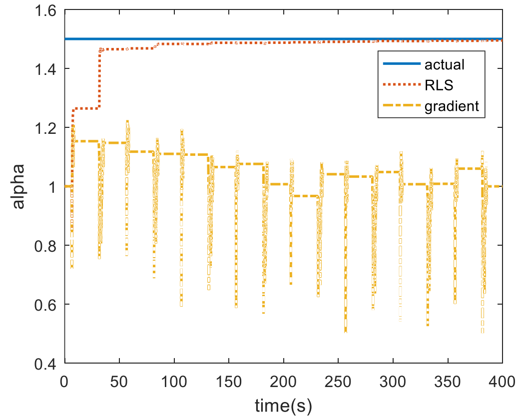

Fig. 5 Adaptation result of the slope gain.

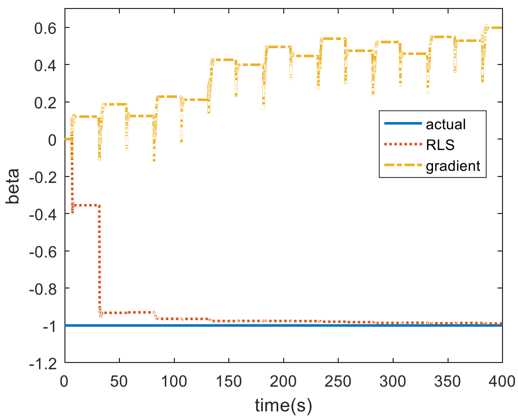

Fig. 6 Adaptation result of the amount of parallel shift.

actuator piston position value.

3. RESULTS

3.1 Adaptation methods

Firstly, the adaptation methods of a friction model using the modified RLS method and the gradient method is verified with simulation data of slip engagement control of a dry engine clutch in a parallel hybrid vehicle. The Simscape tool in MATLAB/Simulink was utilized to construct the

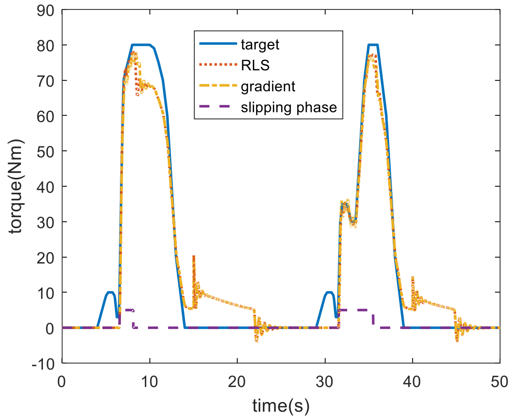

Fig. 7 Torque tracking control result.

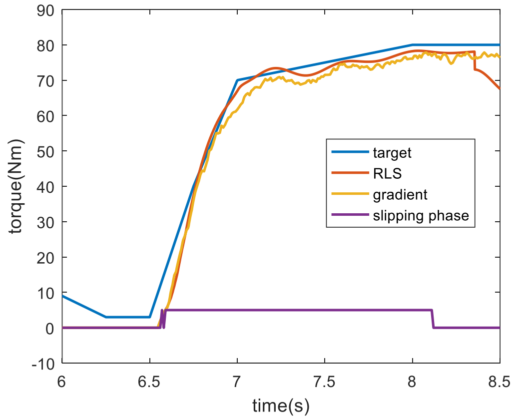

Fig. 8 First enlarged figure of the torque tracking control result.

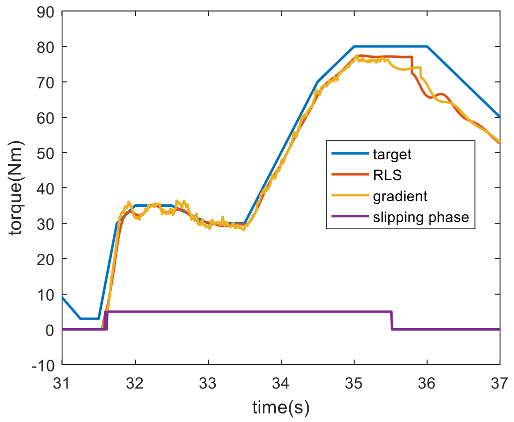

Fig. 9 Second enlarged figure of the torque tracking control result.

parallel hybrid vehicle model.

Fig. 4 shows the engine torque, motor torque, reference clutch torque, and the clutch slipping period. In the simulation, the slip engagement was performed when the engine was idling and the motor speed was zero, and repeated over ten times. Also, the actual slope gain, and the amount of parallel shift of the nominal friction model were set as 1.5, and -1 mm, and the initial values of the estimated slope gain, and the amount of parallel shift were set as 1, and 0 mm.

Fig. 5 shows the adaptation result of the slope gain of the nominal friction model using the modified RLS method and the gradient method. As can be seen in the figure, the estimated slope gain using the RLS method converged fast to the actual value but the estimated gain using the gradient method converged slowly to the actual value.

Furthermore, Fig.6 shows the adaptation result of the amount of parallel shift of the nominal friction model. Likewise the adaptation result of the slope gain, the estimated amount of parallel shift using the RLS method converged fast to the actual value but the estimated one using the gradient method converged slowly to the actual value.

Referring to Fig. 3, the RLS method estimates the slope gain and the amount of parallel shift by approximating the shape of the nominal model (red line in Fig. 3) to the cluster shape of the estimated model data (blue points in Fig. 3) obtained for several time steps. But the gradient method estimates the parameters using only one point of the estimated model data for one time step.

Therefore, the convergence of the estimated parameters to the actual value based on the RLS method is fast and the fluctuation of the estimated parameters is small. On the other hand, the convergence based on the gradient method is slow and the fluctuation of the estimated parameters is big. Also, the gradient method requires various frequency inputs to make the estimates of the slope gain and the amount of parallel shift converged to actual values. However, the computational load of the RLS method is bigger than the gradient method. Furthermore, the convergence of the gradient method is slow but it does not mean the estimated parameters does not converge to the actual value. Although not shown in this paper, the estimated parameters of the gradient method sometimes converged well to the actual value.

3.2 Feedforward control

In this subsection, it was verified whether the clutch torque tracks the reference value well in the feedforward control using the slope gain and the amount of parallel shift.

Fig. 7 shows that the results of the adaptive feedforward control. Also, Fig.8 and Fig.9 are enlarged figures of Fig. 7 during specific time.

As can be seen in Fig. 8, the torque tracking performance of the RLS method was better than that of the gradient method. Also, Fig. 9 shows that the clutch torque of the gradient method oscillates a little due to the fluctuation of the estimated slope gain and the amount of parallel shift. As mentioned in the previous subsection, this is because that the gradient method focuses on the data for one time step. The estimated slope gain and the parallel shift amount did not converge to the actual values during the simulation period for the gradient method, and this caused the clutch torque fluctuation.

4. CONCLUSION

This paper proposed an adaptive feedforward control method of a dry clutch, in which a clutch friction coefficient and a clutch touch point are simultaneously estimated using a clutch torque estimator. The adaptive algorithm was verified with slip engagement simulation of an engine clutch in parallel hybrid vehicle in the MATLAB/Simulink environment. In this paper, the adaptation of the clutch friction coefficient and clutch touch point was considered as the same problem with the adaptation of a slope gain and the amount of parallel shift in the actuator piston position axis of a clutch friction model. Two adaptation techniques were introduced: a modified RLS method and a gradient method. The pros and cons of each method were analyzed. In the future work, the adaptive control method proposed in this study will be verified through production vehicle experiments

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