Adaptive Engagement Control of a Self-Energizing Clutch Actuator System Based on Robust Position Tracking

Jinsung Kim, Member, IEEE, Seibum B. Choi, Member, IEEE, and Jiwon J. Oh

Abstract—This paper considers the problem of designing an engagement controller for a clutch actuator system having self-energizing mechanism. Since such a system includes torque amplification mechanism, parametric uncertainties in the model may lead to large erroneous results in the clutch torque controller. To compensate this undesirable effect, adaptive sliding mode control is applied based on the actuator position tracking. Estimations of the disk friction coefficient and actuator motion parameters are employed to control the engagement torque properly. The disk friction coefficient adaptation provides online stiffness inference for the engagement force while in contact. Moreover, the unstructured disturbance is also compensated by a disturbance observer. Experimental verifications show the improved performance of the developed control method.

Index Terms—automotive system, adaptive control, clutch actuator, drivetrain, torque control, positioning mechanism

I. INTRODUCTION

Recently, automotive engineering technology related with the improvement of fuel efficiency has received much attention. As a solution to this problem, advanced automotive transmissions have been developed. Due to the better efficiency, Automated Manual Transmissions (AMTs) and Dual Clutch Transmissions (DCTs) have attracted considerable attention of automotive manufactures. Unlike traditional Automatic Transmissions (ATs) which guarantees smooth transient responses for accurate torque monitoring.

In this paper, we propose the development of a clutch engagement controller for SECA system. Since such systems include a torque amplification mechanism, parametric uncertainties in the actuator model yield large erroneous results in the clutch torque feedback controller. Thus, a clutch positioning controller should be designed to be robust against torque amplification characteristics.

The main strategy is to employ adaptive sliding mode control (ASMC) with parameter adaptation of the friction coefficient on the clutch disk surface during the slip phase. In addition, a disturbance observer (DOB) is incorporated with ASMC to enhance robustness of the overall control system. The settling time can be reduced by the help of DOB.

In this paper, we consider the problem of designing an engagement controller for a clutch actuator system having self-energizing mechanism. Since such a system includes torque amplification mechanism, parametric uncertainties in the model may lead to large erroneous results in the clutch torque controller. To compensate this undesirable effect, adaptive sliding mode control is applied based on the actuator position tracking. Estimations of the disk friction coefficient and actuator motion parameters are employed to control the engagement torque properly. The disk friction coefficient adaptation provides online stiffness inference for the engagement force while in contact. Moreover, the unstructured disturbance is also compensated by a disturbance observer. Experimental verifications show the improved performance of the developed control method.

Jinsung Kim and Jiwon J. Oh are with Hyundai Motor Company, Hwasung 18280, Korea (e-mail: jsk@kaist.ac.kr; jwo@kaist.ac.kr). Seibum B. Choi are with the Department of Mechanical Engineering, KAIST (Korea Advanced Institute of Science and Technology, Daejeon 34141, Korea (e-mail: sbchoi@kaist.ac.kr).
In overall design framework, while a disturbance observer is incorporated as a mean to estimate and cancel unknown compounded disturbance at a low frequency range, ASMC compensates parametric uncertainties in high-frequencies effectively. Unlike [30], the stability of the combined system in this research is proved by Lyapunov-based analysis. The resulting sufficient condition can provide a straightforward way to select the bandwidth parameter of DOB.

This paper is organized as follows. In section II, the brief description and dynamic model of the system are introduced. In section III, a clutch torque controller based on the position description and dynamic model of the system are introduced. Experimental validations are given in section IV and conclusions can be found in section V.

II. SYSTEM AND MODEL

The schematic of the SECA system is shown in Fig. 1-2. In this paper, since we focus on the adaptive control of the clutch engagement, refer to the preliminary paper [26] for the detailed description of SECA system. In the following, the full system model and its simplified version are proposed for developing a control-oriented model.

A. Full System Model

1) Dynamic Model of the Electric Motor: The dynamic models of the motor are given as follows:

\[
\begin{align*}
L_m i_m + R_m i_m + k_m \omega_m &= u \quad \text{(1a)} \\
J_m \ddot{\omega}_m + T_{fm}(\omega_m) + \frac{T_m}{N_g} &= k_I i_m \quad \text{(1b)}
\end{align*}
\]

where \( L_m, R_m, k_m \) are the inductance, the resistance, the back electromotive force constant, \( i_m \) is the motor current, \( \omega_m \) is the motor angular velocity, \( k_I \) is the motor torque constant, \( N_g \) is the gear ratio between the motor rotor and the mechanical subsystem. Note that \( T_m \) is the load torque for driving the mechanical subsystem that will be introduced in (3). The friction torque \( T_{fm} \) in rotational motion of the motor is described as

\[
T_{fm}(\omega_m) = \begin{cases} 
T_{cm+} + b_m + \omega_m & \omega_m > \epsilon_m \\
T_{cm-} + b_m - \omega_m & \omega_m < -\epsilon_m \\
T_m & \text{if } |\omega_m| < \epsilon_m \text{ and } |T_{fm}| < T_{fsm} \\
T_{fm}\text{sgn}(T_m) & \text{if } |\omega_m| < \epsilon_m \text{ and } |T_{fm}| > T_{fsm}
\end{cases}
\]

where, \( T_{cm} \) is the Coulomb friction torque, \( b_m \) is the viscous friction coefficient, and \( T_{fsm} \) is the static friction torque. Note that \( \epsilon_m \) denotes a small zero velocity interval where the motor velocity is taken into account as zero. The subscripts '+' and '-' are used to represent the hysteresis phenomenon.

2) Dynamic Model of the Mechanical Subsystem: The fixed plate shown in Fig. 2 is interposed between the clutch cover and the friction disks in order to adjust the axial displacement of the actuation plate while rotating at the same time. In the clutch open phase, the rotational equation of motion for the actuation plate without the clutch engagement torque is described as

\[
J_a \ddot{\omega}_a = T_a - T_{fa}(\omega_a) \quad \text{(3)}
\]

where \( J_a \) is the moment of inertia of the actuation plate. Since the pinions are constrained by two supporting plates ((a) and (b) in Fig. 1-2) and the pinion guide, the motion of them coincides with the actuation plate. Thus, it is reasonable to assume that the inertia of the pinions is lumped into that of the actuation plate. The driving torque \( T_a \) is generated by an elastic deformation between the motor and the mechanical subsystem with the equivalent torsional stiffness \( k_a \) defined as

\[
T_a = k_a \left( \frac{\theta_m}{N_g} - \theta_a \right) \quad \text{(4)}
\]

where \( \theta_m, \theta_a \) are the motor and the actuator angular position, respectively. In (3), the frictional torque \( T_{fa} \) on the worm shaft can be represented by replacing the subscript 'm' in (2) with 'a'.

![Fig. 1. Schematic of self-energizing clutch actuator (SECA).](image)

![Fig. 2. Equivalent model for a SECA system.](image)
When the clutch is in contact with the surface for engagement operation, the clutch torque $T_c$ and the reinforcement torque from the interaction force $F_p$ on the rack and pinion surface are added in (3) as shown in Fig. 2. Therefore, the equation of motion for the actuation plate in the slip phase is represented by

$$J_a\ddot{\omega}_a = T_a + T_c - 2r_p F_p \sin \alpha - T_{fa}(\omega_a)$$  \hspace{1cm} (5)

where, $r_p$ is the radius of bevel gear position, and $\alpha$ the inclined surface angle on the actuation plate and the fixed plate, respectively. For the positive slip phase, the clutch torque $T_c$ is given as

$$T_c = \mu R_c F_n$$  \hspace{1cm} (6)

where, $\mu$ is the dry friction coefficient on the disk, $R_c$ the clutch radius, and $F_n$ the applied normal force. The interaction force $F_p$ at the contact point between the rack and pinion gear tooth meshed inside the actuation plate depends on the normal force $F_n$ with the inclined surface angle $\alpha$:

$$F_p = \frac{F_n}{\cos \alpha}.$$  \hspace{1cm} (7)

Note that the third term at the right hand side of (5) is related with a self-energizing effect. The wedge structure comes from the rack and pinion mechanism.

The axial displacement $x_p$ of the actuation plate can be calculated through the geometric relation with the angular position $\theta_a$ of that as shown in Fig. 2. It is therefore given by

$$x_p = \beta \theta_a$$

The normal force applied on the friction disk is

$$F_n = k_p x_p = k_p \beta \theta_a$$  \hspace{1cm} (9)

where $k_p$ is the stiffness of the actuation plate. It is assumed that the normal force $F_n$ is proportional to the actuator stroke $x_p$ in axial direction.

According to (7) and (9), the actuator dynamics (5) in the positive slip phase can be rewritten as

$$J_a\ddot{\omega}_a = \mu R_c F_n + T_a - \beta F_n - T_{fa}(\omega_a, z_a).$$  \hspace{1cm} (10)

Overall system dynamics described in this section is shown in Fig. 3.

### B. Simplified Model for Control

In order to facilitate a control design and implementation, the order of the system model is reduced by neglecting high frequency components. The following assumptions are made to simplify the actuator model.

A1) The electrical dynamics of the DC motor is faster than the mechanical motion. 

A2) The bandwidth of the actuation plate motion is very high, and its rotating angle is very small. 

A3) In the mechanical subsystem, frictional motion during presliding can be negligible [31].

A1 means that the inductance of the DC motor could be neglected by which $L_m$ is much smaller than $J_m$ [32]. Thus, the dynamic equation (1a) is converted into an algebraic equation as follows:

$$u = R_m i_m + k_m \omega_m$$  \hspace{1cm} (11)

Based on A2, the dynamics of the actuation plate is negligible. Consequently, (10) can be rewritten as

$$2r_p F_n \tan \alpha = T_a + \mu R_c F_n$$  \hspace{1cm} (12)

Combining (4) and (12) gives a single equation for the normal force:

$$F_n = \frac{k_a (\theta_m/N_g - \theta_a)}{2r_p \tan \alpha - \mu R_c}$$  \hspace{1cm} (13)

Using the normal force expressions (9) and (13), the relationship between the motor angular position and the angular position of the actuation plate is given by

$$\theta_a = \frac{k_a}{N_g (k_b(\mu) + k_a)} \theta_m$$  \hspace{1cm} (14)

where the auxiliary function $k_b$ is defined for notational simplicity as

$$k_b(\mu) \triangleq \xi(\mu)(2r_p k_b \tan \alpha) = \xi(\mu) k_p, \hspace{1cm} (15a)$$

$$\xi(\mu) \triangleq 2r_p \tan \alpha - \mu R_c = \beta - \mu R_c.$$  \hspace{1cm} (15b)

Note that $\beta$ defined in (8) is determined from a hardware design specification. $k_b$ and $\xi$ in the expression (15) are erroneous parameters due to the friction coefficient $\mu$ which is uncertain and time-varying. (14) shows that the positioning of the mechanical subsystem is an uncertain function of the motor position. Substituting (14) into (13) derives the relationship between the clutch normal force and the motor position

$$F_n = \Phi(\mu) \theta_m$$  \hspace{1cm} (16a)

$$\Phi(\mu) \triangleq \frac{k_a}{\xi(\mu) \left[ N_g (k_b(\mu) + k_a) \right]}$$  \hspace{1cm} (16b)

where the nonlinear function $\Phi(\mu)$ is nonsingular because all parameters in (16b) are bounded. The above equation is applied to convert the desired normal force to the desired motor position, which will be used in the subsequent control development.
From A3, the following linear-in-parameters model for motion friction depending only on the velocity is given by

\[
T_f(\omega_m) = \phi_1 \text{sgn}(\omega_m) + \phi_2 \omega_m
\]

(17)

where \( \sigma_0 \) and \( \sigma_1 \) are dominant friction parameters in the simplified model. The detailed simplification process can be found in [31]. Actual values of friction parameters are unknown but bounded.

Combining equation (1b), (11), (12), (14), and (17) yields

\[
\dot{J}_m\dot{\omega}_m + q\dot{\omega}_m + T_f(\omega_m) + r(\mu)\theta_m = p(u + d)
\]

(18)

where, \( \dot{J}_m \) in (19a) and \( r \) in (19d) are the equivalent moment of inertia, and the nonlinear function of friction coefficient, respectively. Note that \( \bar{d} \) is the unstructured disturbance from the unmodeled effect and the model reduction. It can be decomposed into \( d_h \) in high frequencies and \( d_l \) in low frequencies. They are assumed to be bounded. For simplicity, other auxiliary variables \( p, q \) and \( w \) are defined as

\[
\dot{J}_m \triangleq J_m + \frac{J_a}{N_g}, \quad w \triangleq \frac{k_a}{N_g^2},
\]

(19a)

\[
p \triangleq \frac{k_t}{R_m},
\]

(19b)

\[
q \triangleq \left( \frac{k_b k_m}{R_m} + b_m \right),
\]

(19c)

\[
r(\mu) \triangleq \frac{k_b k_b(\mu)}{N_g^2 [k_b(\mu) + k_a]} = w \left( \frac{k_b(\mu)}{k_b(\mu) + k_a} \right),
\]

(19d)

\[
d = d_h + d_l
\]

(19e)

As a result, the reduced order model can be expressed as a second-order. The fourth term implicitly includes the engagement torque when the clutch is in contact. The problem might appear to be one of the position tracking problem with uncertain reaction torque corresponding to the equivalent reaction load \( r(\mu)\theta_m \).

III. CLUTCH ENGAGEMENT CONTROL DEVELOPMENT

The overall control system for the clutch engagement has two sub-controllers in a large point of view. First, the adaptive sliding mode control (ASMC) is used to satisfy a desired performance specification and to compensate structured uncertainties for a linear-in-parameters form especially in the high frequency range. The disk friction coefficient is estimated to improve the control accuracy when the clutch comes into contact. Second, as an inner-loop compensator, the disturbance observer (DOB) compensates unmodeled effect and unstructured disturbances at low frequencies. Therefore, it can reduce uncertainties in the input-channel by rejecting the unknown disturbances.

A. Problem Formulation

The objective of the proposed strategy is to design a clutch normal force tracking controller. Due to the presence of nonlinear friction and parametric uncertainties, a robust control method is required to meet the desired specification. However, since torque transducers are very expensive, the clutch torque measurement is impractical for real environment.

On the other hand, the motor position can be measured easily using an incremental encoder. And, the clutch torque/force is converted into the actuator stroke by the relationship between the normal force and the motor position in (16a). It can be expressed as a nonlinear function of the motor position by \( \Phi(\mu) \) defined in (16b). Therefore, the motor position error will be used to define a sliding surface instead of the normal force.

Let \( e_f \triangleq F_{nd} - F_n \) be the normal force tracking error with the desired normal force \( F_{nd} \). The motor position error \( e_m \) is defined as \( e_m \triangleq \hat{\theta}_{md} - \theta_m \) and its derivative \( \dot{e}_m \) is \( \dot{\theta}_{md} - \theta_m \). By using (16), it can be also represented as

\[
e_m = \Phi^{-1}(\mu)e_f
\]

(20)

Then, the filtered tracking error \( z \) is defined as

\[
z \triangleq \varepsilon_m + \lambda \varepsilon_m
\]

(21)

where, \( \lambda \) is a constant design parameter.

The following open-loop error system is obtained by multiplying \( \dot{J}_m \) on the time derivative of (21) and combining (18):

\[
\dot{J}_m \dot{z} = \phi_1 \omega_m + \phi_2 \omega_m + r(\mu)\theta_m + \dot{J}_m \omega_{mr} - p(u + d_h + d_l)
\]

(22)

where \( \omega_{mr} := \dot{\theta}_{md} + \lambda \varepsilon_m \) and \( \phi_2 = \phi_2 + q \).

B. Adaptive Sliding Control Design

Since the self-energizing characteristics of the given system can amplify the applied normal force, small parametric uncertainty may lead to the significant tracking error that implies the mismatch between the desired force and the actual one [27].

Another drawback is mechanical design complexity induced by actual implementation of self-energizing mechanism. This problem also leads to highly nonlinear friction disturbances during operation. In order to overcome this problem, sliding mode control (SMC) is firstly introduced to make a control system robust against unmodeled dynamics and effective tracking capability. In addition, it is necessary to account for the motion friction compensation by parameter adaptation to alleviate high control action of SMC.

Also, the disk friction coefficient \( \mu \) is an uncertain parameter. It varies with the operating temperature, material properties, and a slip speed of both sides of the clutch. To compensate this uncertainty, an adaptation scheme for the disk friction coefficient \( \mu \) will be designed later.

The control law (22) based on adaptive sliding mode control (ASMC) is defined as

\[
u = \frac{1}{p} (u_r + M \text{sgn}(z)) + u_d
\]

(23)

where

\[
u_r = u_0 + u_{a1} + u_{a2},\]

(24a)

\[
u_0 = K z + \dot{J}_m \omega_{mr},\]

(24b)

\[
u_{a1} = \dot{\lambda} \theta_m,\]

(24c)

\[
u_{a2} = \dot{\phi}_1 \text{sgn}(\theta_m) + \dot{\phi}_2 \dot{\theta}_m,\]

(24d)

\[
u_d = -\dot{d}_l\]

(24e)
where, \( u_0 \) is a feedback controller including feed-forward action, \( u_{a1} \) the adaptive controller for the motion tracking, and \( u_{a2} \) the adaptive disk friction compensator, respectively. In particular, \( u_q \) is the disturbance compensator to be synthesized in section III-D. \( \hat{r}, \hat{k}_b \), and \( \hat{\mu} \) are estimated parameters of \( r, k_b \), and \( \mu \) respectively. \( \hat{r}, \hat{k}_b \) are also defined as
\[
\hat{r} \triangleq r(\hat{\mu}) = \frac{\hat{k}_b(\hat{\mu})}{\hat{k}_b(\hat{\mu}) + k_n} \tag{25}
\]
\[
\hat{k}_b \triangleq \xi(\hat{\mu})k_p\beta. \tag{26}
\]

Here, \( \hat{r} \) includes an erroneous parameter \( \hat{k}_b \) which is a function of \( \hat{\mu} \) and other system parameters. The closed-loop system is rewritten by substituting equation (23) into (22) as
\[
\hat{J}_m \hat{z} = \hat{\theta}_m + \hat{\phi}_1 \text{sgn}(\hat{\theta}_m) + \hat{\phi}_2 \hat{\theta}_m + \hat{\theta}_m \tag{27}
\]
\[
- Kz - M\text{sgn}(z) - pd_l + pd_h
\]
where, \( \hat{r} = r(\mu) - \hat{r}(\mu) \), \( \hat{\phi}_1 = \phi_1 - \hat{\phi}_1 \), and \( \hat{\phi}_2 = \phi_2 - \hat{\phi}_2 \) are the parameter estimation errors. The disturbance estimation error is denoted as \( \hat{d}_l = d_l - \hat{d}_l \). Note that \( \hat{r} \) is a function of various parameters including parameter uncertainty of \( \mu \) as shown in (25).

C. Adaptation Laws

The disk friction coefficient adaptation requires additional assumption that is trivial during the clutch slip phase.

A4) The control input can be chosen such that the friction coefficient is within the set \( \Omega_\mu \):
\[
\mu \in \Omega_\mu \triangleq \{ \mu(\omega_m) \mid \omega_m < \mu < \omega_m \} \tag{28}
\]
where \( \omega_m \triangleq \min_{t \in [0,T]} \omega_m \), \( \omega_m \triangleq \max_{t \in [0,T]} \omega_m \).

A4 says that the friction coefficient \( \mu \) on the clutch disk surface is confined to the range of (28). It implies that the actuator rotational speed is not a constant ensuring that the persistence excitation (PE) condition for parameter adaptation is satisfied. (28) defines a validity domain of the estimation for the disk friction coefficient. Note that A4 is generally met when the clutch slip remains.

The sliding control input in (23) plays a crucial role in satisfying A4. Due to the limitation of actuator bandwidth, the sign function is approximated by a saturation function that is defined as
\[
sat(z) = \begin{cases} 
\frac{z}{|z|} & \text{if } |z| \leq \sigma \\
\pm \frac{\sigma}{|z|} & \text{otherwise}
\end{cases} \tag{29}
\]
where, \( \sigma > 0 \) denotes the switching boundary, and \( \delta > 0 \) the approximation margin. A continuous slip of the clutch depends on the switching frequency of the control signal resulting from the selection of both parameters \( \sigma \) and \( \delta \). The finite switching of (29) could intentionally induce a clutch slip to identify the friction coefficient.

In the subsequent development, adaptation laws are updated depending on the tracking error so that small control error may exhibit parameter drifting. It should be noted that adaptation laws keep the estimate of the friction coefficient within the physical boundary.

To do this, the projection operator for a given vector \( \chi \) needs to be defined. Let \( \mathcal{P} := \{ r \in \mathbb{R} \mid \kappa \leq 0 \} \) be a closed convex set and \( \kappa \in \mathbb{R}^p \) a smooth function. The projection operator is defined as
\[
\text{Proj}_\mathcal{P}(\chi) = \begin{cases} 
\chi & \text{if } \chi \in \mathcal{P}^0 \text{ or if } \nabla \kappa^T \chi \geq 0 \\
(1 - \varepsilon \nabla \kappa^T \chi) & \text{if } \chi \in \partial \mathcal{P} \text{ and } \nabla \kappa^T \chi < 0
\end{cases}
\]
where, \( \varepsilon \in \mathbb{R}^{p \times p} \) is any adaptive function, \( \varepsilon \in \mathbb{R}^{p \times p} \) a positive definite adaptive gain matrix, \( \mathcal{P}^0 \) the interior of \( \mathcal{P} \), \( \partial \mathcal{P} \) the boundary of \( \mathcal{P} \), and \( \nabla \kappa = d\kappa/d\chi \) the outward unit normal vector at \( \chi \in \partial \mathcal{P} \). With the prior knowledge of the parameter variation range, a projection operator plays the role of preserving the passivity property of a self-energizing effect.

Let the admissible region \( \mathcal{P}_\mu \) be a closed set given as \( \mathcal{P}_\mu := \{ \tilde{r} \in \mathbb{R} \mid \xi(\hat{\mu}) > 0 \} \) where the clutch does not get stuck [26]. The adaptive law of \( \hat{r} \) is designed by
\[
\dot{\hat{r}} = \text{Proj}(\varepsilon \theta_m z) = \begin{cases} 
\varepsilon \theta_m z & \text{if } \hat{r} \in \mathcal{P}_\mu^0 \text{ or if } \nabla \kappa^T \theta_m z \geq 0 \\
0 & \text{otherwise}
\end{cases}
\]
(30)
where \( r_0 \) is a prescribed nominal value of \( r \). The solution trajectory of (30) is confined within \( \Omega_t = \{ \hat{r} \leq \hat{r} \leq \tilde{r} \} \subset \mathcal{P}_\mu^0 \). (30) is also valid to ensure that \( \hat{k}_b \) makes its estimate stay in the region \( \mathcal{P}_\mu \). It can be shown as follows. Since \( \hat{r} \) and \( \hat{k}_b \) are positive constants in (15), \( \hat{k}_b \) is also positive when \( \xi(\hat{\mu}) \) is a positive function. With this and the fact
\[
\frac{\partial \hat{r}}{\partial k_b} = \frac{w}{(k_b + k_n)^2} > 0, \tag{31}
\]
it implies that \( \hat{r} \) is positive when \( \xi(\hat{\mu}) > 0 \). As a result, \( \hat{r} \) is evolved inside \( \mathcal{P}_\mu \) or along the tangential plane of \( \partial \mathcal{P}_\mu \). Note that \( \hat{r} \) can be assumed as a slowly varying parameter so that the time derivative becomes \( \dot{\hat{r}} = -\hat{r} \). Adaptation laws for compensating nonlinear friction in motion in the system are given with initial conditions \( \phi_1(0) \) and \( \phi_2(0) \) as
\[
\phi_1 = \text{Proj}(\delta_1 \text{sgn}(\hat{\theta}_m)z), \quad \phi_1(0) = f_s, \quad \phi_2 = \text{Proj}(\delta_2 \hat{\theta}_m z), \quad \phi_2(0) = b_m \tag{32}
\]
(33)
where \( \delta_1 \) and \( \delta_2 \) are design parameters for determining adaptation rate.

**Theorem 1:** Assume that the unstructured uncertainties at a low-frequency \( d_l \) is not considered in the tracking error system (22), i.e., \( d_l = 0 \). Under the assumptions A1-A4 hold, the controller given by (23) in conjunction with the adaptation laws (30), (32) and (33) ensures the asymptotic tracking of the normal force control system in the sense that
\[
\hat{\theta}_m \to 0 \quad \text{and as } t \to \infty,
\]
and the disk friction coefficient error converged to zero in a given compact set \( \mu \in \Omega_\mu \times \mathcal{P}_\mu \) provided that \( \varepsilon > 0, \delta_1 > 0, \) and \( \delta_2 > 0 \) are properly chosen, and the desired trajectory \( \theta_{md} \) is sufficiently bounded and smooth (i.e. \( \theta_{md}, \theta_{md}, \theta_{md} \in \mathcal{L}_\infty \)).
**Proof:** Let $V(z, \hat{r}, \hat{\phi}_1, \hat{\phi}_2) \in \mathbb{R}$ denote a positive definite Lyapunov function candidate

$$V = \frac{1}{2} \dot{J}_m z^2 + \frac{1}{2\varepsilon} \hat{r}^2 + \frac{1}{2\delta_1} \hat{\phi}_1^2 + \frac{1}{2\delta_2} \hat{\phi}_2^2,$$  \hspace{1cm} (34)

and the time derivative of (34) along the trajectory of (27) with $\dot{d} = 0$ is given by

$$\dot{V} = \ddot{J}_m z^2 + \frac{1}{\varepsilon} \hat{r} \dot{r} + \frac{1}{\delta_1} \hat{\phi}_1 \dot{\hat{\phi}}_1 + \frac{1}{\delta_2} \hat{\phi}_2 \dot{\hat{\phi}}_2. \hspace{1cm} (35)$$

Using (23) and (27), it is rewritten as

$$\dot{V} = z [\hat{\phi}_1 sgn(\hat{\theta}_m) + \hat{\phi}_2 \hat{\phi}_m + \hat{r} \hat{\theta}_m - Kz - pd_h - M sgn(z)] - \frac{1}{\varepsilon} \hat{r} \dot{r} - \frac{\hat{\phi}_1 \dot{\hat{\phi}}_1 + \hat{\phi}_2 \dot{\hat{\phi}}_2}{\delta_1} \dot{z} \dot{z},$$

$$= -Kz^2 + pd_h z - M|z| + \dot{r} \left[ \hat{\theta}_m z - \frac{\dot{z}}{\varepsilon} \right],$$

$$+ \hat{\phi}_1 \left[ sgn(\hat{\theta}_m) z - \frac{\hat{\phi}_1}{\delta_1} \right] + \hat{\phi}_2 \left[ \hat{\theta}_m z - \frac{\hat{\phi}_2}{\delta_2} \right]. \hspace{1cm} (36)$$

where the design parameter $M$ is selected to satisfy the inequality $M \geq |pd_h|$. By utilizing (30), (32), and (33), the following inequality is obtained:

$$\dot{V} \leq -Kz^2. \hspace{1cm} (37)$$

Therefore, the time derivative of $V$ is negative semi-definite. Generally, the control system property of interest is asymptotic stability. Based on (34) and (37), it follows that $z, \hat{r}, \hat{\phi}_1, \hat{\phi}_2 \in \mathcal{L}_\infty$ and $z \in \mathcal{L}_2$. The definition (21) shows that $z \in \mathcal{L}_\infty$ implies $\varepsilon_m, \dot{e}_m \in \mathcal{L}_\infty$. (30), (32), and (33), and projection operators can be used to show that $\dot{r}, \hat{\phi}_1, \hat{\phi}_2 \in \mathcal{L}_\infty$. From these facts, the control inputs (24a)-(24d) are bounded. It follows from (27) that $\dot{z} \in \mathcal{L}_\infty$. Based on the fact that $z, \dot{z} \in \mathcal{L}_\infty$ and $z \in \mathcal{L}_2$, Barbalat’s Lemma can be applied [33] to show that

$$\lim_{\epsilon \to \infty} z = 0. \hspace{1cm} (38)$$

Consequently, the clutch normal force tracking is also achieved by (20). For the convergence of adaptation parameters, the persistence of excitation (PE) condition [34] has to be satisfied. If the condition (28) in A4 holds, the PE condition is satisfied. Note that the $\hat{\phi}_1$ and $\hat{\phi}_2$ are utilized only for improved the motion tracking performance. The parameter for $\hat{\mu}$ is only active when the clutch is engaging during the slip phase. It means that $\hat{r} \rightarrow r$ in $\Omega_\mu \times \Omega_\tau$.

Finally, since the purpose of this scheme is the good estimation of the friction coefficient $\mu$, it is required to derive an equation to estimate $\hat{\mu}$ from $\hat{r}$. The time derivative of $\hat{r}$ can be obtained from (25) as,

$$\dot{\hat{r}} = \frac{w}{\dot{\xi}} \left[ \frac{\hat{k}_b}{\hat{k}_b + k_a} \right] = \frac{w}{k_a (\hat{k}_b + k_a)^2} \dot{\hat{k}}_b, \hspace{1cm} (39)$$

The equation for $\dot{\hat{k}}_b$ is rewritten by

$$\dot{\hat{k}}_b = \frac{(\hat{k}_b + k_a)^2}{w k_a} \dot{r}, \hspace{1cm} (40)$$

where $k_b$ is defined as a function of $\mu$ in (15a) for the notational simplicity. The estimated parameter of $k_b$ and its time derivative are:

$$\dot{\hat{k}}_b = \hat{\xi}(\hat{\mu}) (k_p \beta)$$  \hspace{1cm} (41)

$$\dot{\hat{\mu}} = - (k_p R_c \beta) \hat{\mu}. \hspace{1cm} (42)$$

Combining (40) and (42) yields

$$\dot{\hat{\mu}} = - \frac{w (\hat{k}_b + k_a)^2}{w k_a R_c \beta} \dot{r}. \hspace{1cm} (43)$$

The adaptation law in terms of $\hat{\mu}$ is given by

$$\dot{\hat{\mu}} = - \frac{(\hat{\xi}(\hat{\mu}) (k_p \beta) + k_a)^2}{w k_a R_c \beta} \dot{r}. \hspace{1cm} (44)$$

To guarantee the boundness of parameter estimates, the
projection operator is used at each step. The relationship from (39) to (44) concludes that $\hat{k}_b \to k_b$. Hence, the unknown parameter $\hat{\mu}$ also converges to an actual parameter $\mu$ in $\Omega_\mu \times \mathcal{P}_\mu$.

D. Disturbance Compensation by the Disturbance Observer

In order to enhance control robustness, a disturbance observer (DOB) is incorporated with the adaptive sliding control developed in the previous subsection. The DOB estimates the lumped disturbances $d_l$ defined in (18) and (19e) by using a low-pass filter $Q(s)$ and a nominal model [35]–[37]. After plugging this estimated disturbance into the control input, the uncertain plant behaves like the nominal model.

The DOB-based control has advantages that it can effectively eliminate unstructured uncertainty under the specified pass-band of $Q(s)$.

The actuator position control for clutch applications requires a short settling time. The use of the larger feedback gain is limited particularly due to the actuator saturation. Therefore, the disturbance rejection capability is necessary to have the motor control bandwidth increased as much as possible through the accurate feed-forward control. The DOB-based control is incorporated to improve the robustness to matched uncertainties $d_l$ at a low frequency range while the adaptive sliding control plays the role in dealing with structured uncertain parameter compensation in the high frequency parts. The conventional DOB for linear systems [35] is modified for our nonlinear system.

The nominal system is denoted as $P_n : u_n \to \theta_m$ which can be obtained from neglecting the unknown disturbance $d$ in the uncertain nonlinear system (18).

$$P_n : \quad \tilde{J}_m \omega_m + q \omega_m + T_f(\omega_m) + r(\mu)\theta_m = pu_n$$ (45)

where $u_n$ is the nominal control input. Note that $u_n$ can be computed from numerically solving (45) inversely ($P_n^{-1} : \theta_m \to u_n$) with given initial conditions $\theta_m(0)$ and $\omega_m(0)$, and measurements $\theta_m$ and $\omega_m$. Motivated from the conventional DOB [35], unknown disturbance $d_l$ is approximated as the difference between the nominal input and the actual control input denoted by $d_l(z) \approx u_n - u$. Hence, $\hat{d}_l$ is given by

$$\hat{d}_l = Q(s)d_l = Q(s)[u_n - u].$$ (46)

Since $Q(s)$ is a stable low-pass filter, (46) can be rewritten by

$$\tau_q \dot{\hat{d}}_l = -\hat{d}_l + d_l, \quad \hat{d}_l(0) = 0$$ (47)

where $\tau_q$ is a small parameter to adjust the pass-band of (47). The following theorem shows the boundedness and convergence of the unstructured disturbance estimator-based control incorporating the ASMC proposed in Theorem 1.

Theorem 2: Consider the case where the unstructured disturbance $d_l$ is nonzero in the tracking error system (22). Given compact sets $\Omega_z \subset \mathbb{R}$ and $\Omega_{d_l} \subset \mathbb{R}$ of initial conditions $z(0)$ and $d_l(0)$, there exists $\tau_q > 0$ for all $0 < \tau_q < \tau_q^*$ and for all $z(0) \in \Omega_z$ and $d_l(0) \in \Omega_{d_l}$. The controller given by (23) in

\[ \begin{align*}
\dot{\hat{z}}_m &= -\hat{d}_l - \hat{\mu}q\hat{\omega}_m - \hat{\mu}T_f(\hat{\omega}_m) + r(\hat{\mu})\hat{\theta}_m + \hat{\mu}p, \\
\dot{\hat{\theta}}_m &= \hat{d}_l - \hat{\mu}q\hat{\omega}_m - \hat{\mu}T_f(\hat{\omega}_m) + r(\hat{\mu})\hat{\theta}_m + \hat{\mu}p,
\end{align*} \]

is robust to all uncertainties (structural and unstructured) and achieves the tracking error to zero for all $z(0) \in \Omega_z$ and $d_l(0) \in \Omega_{d_l}$. The controller given by (23) in
where the result of Theorem 1 is used for\[ V(z, \tilde{d}_l) \leq -Kz^2 - p\tilde{d}_l z - \frac{1}{\tau_q} \tilde{d}_l^2 + \tilde{d}_l \left| \frac{\partial d_l(z)}{\partial z} \right| \left[ c_1 |z| + p |\dot{\tilde{d}}_l| \right] \]

hold on the level set $\Omega_c$ of $W$. Using (53) and (54), (51) is expressed as

$$W(z, \tilde{d}_l) \leq -Kz^2 - \frac{1}{\tau_q} \tilde{d}_l^2 + \tilde{d}_l \left| \frac{\partial d_l(z)}{\partial z} \right| \left[ c_1 |z| + p |\dot{\tilde{d}}_l| \right]$$

provided that the following inequalities are satisfied.

Proof: The disturbance estimation error at low-frequencies is defined as $\tilde{d}_l = d_l - \hat{d}_l$. Using (47) and (27), the time derivative of $\tilde{d}_l$ is given as

$$\dot{\tilde{d}}_l = -\frac{1}{\tau_q} \tilde{d}_l + \frac{\partial d_l(z)}{\partial z} \dot{z}.$$ (49)

Consider the Lyapunov function candidate $W(z, \tilde{d}_l) \in \mathbb{R}$ with $V$ defined in (34)

$$W(z, \tilde{d}_l) = V(z, \dot{r}, \phi_1, \phi_2) + \frac{1}{2} \tilde{d}_l^2.$$ (50)

The time derivative of (50) along the trajectory of (49) is given as

$$\dot{W}(z, \tilde{d}_l) = \dot{V}(z, \dot{r}, \phi_1, \phi_2) + \dot{\tilde{d}}_l \dot{\tilde{d}}_l = -Kz^2 - p\tilde{d}_l z - \frac{1}{\tau_q} \tilde{d}_l^2 + \frac{\partial d_l(z)}{\partial z} \dot{z}.$$ (51)

where the result of Theorem 1 is used for $\dot{V}$ and the term $p\tilde{d}_l z$ is added for the case $\dot{d}_l \neq 0$. Since the actuator is initially at rest, it follows that $\dot{d}_l(0) = 0$ with $z(0) = 0$ in (49). It means that for any compact set $\Omega_z$ and $\Omega_{d_l}$ of $z(0)$ and $\dot{d}_l(0)$, respectively, such a compact set can be found as

$$\Omega_z \times \Omega_{\dot{d}_l} \subseteq \Omega_c.$$ (52)

Fig. 7. Experimental results of the engagement control without adaptations (a) Motor position (actuator stroke). (b) The input shaft (AC motor) and the output shaft speed. (c) Zoom of position response in the slip-ready position. (d) Zoom of position response in the contact position.

In Theorem 1, (34) and (37) show that $V$ is bounded. By the fact that $z$ and the additional term from $\tilde{d}_l$ are bounded, there exist $c_1 > 0$ and $c_2 > 0$ such that the inequalities

$$\dot{z} \leq c_1 |z| + p |\dot{\tilde{d}}_l|$$

hold on the level set $\Omega_c$ of $W$. Using (53) and (54), (51) is expressed as

$$\dot{W}(z, \tilde{d}_l) \leq -Kz^2 - \frac{1}{\tau_q} \tilde{d}_l^2 + \tilde{d}_l \left| \frac{\partial d_l(z)}{\partial z} \right| \left[ c_1 |z| + p |\dot{\tilde{d}}_l| \right]$$

where $c_3 = c_1 c_2 - p$ and $c_4 = c_2 p$ with $c_1 c_2 > p$. By selecting (48), it can be concluded that there exists $0 < \tau_q < \tau_q^*$ such that $\dot{W}$ is negative semi-definite on the level set $\Omega_c$. The remaining part for parameter adaptation convergence can be stated by the similar way as in Theorem 1.

Remark 1: This approach is inspired from the conventional DOB. $\tau_q$ is a time constant of (47) which corresponds to the Q-filter parameter. The level set $\Omega_c$ defined in (52) implies the low-frequency range to be rejected by (46). From (48), it depends on the growth rate of disturbance with respect to the tracking error and the feedback gain $K$. Compared with the conventional DOB for linear systems, such a restriction is imposed for the extension to nonlinear systems. The block diagram of the closed-loop control system is shown in Fig. 4.

IV. EXPERIMENTAL VERIFICATION

A. Experimental Setup

The schematic of the overall control system architecture is shown in Fig. 5. Speed and torque sensor outputs are transferred to dSPACE MicroAutobox DS1401, which is a controller board for rapid control prototyping system. For the purpose of feedback control, the motor position is measured using an incremental encoder with the resolution 0.026° per pulse that is attached at the back of the motor shaft. Note that the speed signal is obtained by pseudo-differentiation of the position measurement. The dc motor for the actuator is driven by a H-bridge driver which provides pulse width modulated input. In addition, since the SECA system functions as a power transmission element, the drivetrain test bed including engine and vehicle inertia is required for validation. An AC motor is installed as an alternative of an automotive engine. An inertia disk is used to represent the vehicle mass. The speed of the AC motor and the inertia disk are measured by torque/speed sensors and the optical encoder, respectively. The torque sensors provide real-time measurement of the shaft torque but are used only for validation purpose of the disk friction coefficient when the clutch makes contact.
4.5 mm away from free motion to clutch engagement. This is 10Hz Q-filter bandwidth. It can overcome the hysteretic effects. In this experiment, the overall control strategy for clutch engagement has three modes: approaching, slip-ready, engaging mode. 1) In the approaching mode, the clutch is initially at rest. The controller only considers the positioning work since the clutch is initially disengaged. Hence, the adaptation law (30) for $\hat{\phi}_1$ is not active and (32) and (33) are only activated. 2) When the clutch is positioned to the near contact point, this is called the slip-ready mode. 3) Then, the clutch is positioned to the desired stroke corresponding to the desired force for full engagement.

The clutch makes contact with the opposing surface from the near contact point. The adaptation law (30) is turned on just in order to identify the disk friction coefficient. This is called the engaging mode. All modes utilize one position-based force control so that there is no switching phenomenon in transitions between each mode.

It is assumed that the stroke for the clutch to be engaged is known. This is a reasonable assumption since real vehicles generally have initial stroke search logic. The desired trajectory can be generated by the nonlinear filter [38] such that $\theta_{md}, \dot{\theta}_{md}, \ddot{\theta}_{md} \in \mathcal{L}_\infty$.

1) Without Adapations: The actuator position control for engagement without adaptations has been conducted for comparison. Fig. 7(a)-(b) show that the position tracking and the shaft speed synchronization are achieved well. In Fig. 7(c)-(d), the slip-ready position and the contact position in the engaging mode are magnified from Fig. 7(a). Although, the desired clutch position is well tracked by the controller without any adaptation scheme, residual vibrations arise mainly due to the highly stiff contact.

2) With Adaptations: With the adaptation laws turned on at each mode, the same experiments have been conducted. Fig. 8(a) show that the actuator position can be controlled by the motor position. It shows that the clutch control starts with

The **position tracking results and the tracking errors** are shown in Fig. 6. They clearly show the difference in tracking capability among them. The PD control result has irregular tracking errors due to abrupt overshoot errors are observed. The ASMC-DOB shows that the tracking performance and robustness are effectively improved compared with other results. In this experiment, $\tau_q$ is selected as 0.016 which is 10Hz Q-filter bandwidth. It can overcome the hysteretic motion friction and unmodeled effect. And, the settling time is reduced to 0.25s during which the clutch actuator stroke moves 3 mm away from free motion to clutch engagement. This is satisfactory for design specification of production vehicles. Therefore, experimental results confirm that the ASMC-DOB is the best candidate for the position control. The rms error and the maximum error are shown in Table. I.

C. Clutch Engagement Control

The overall control strategy for clutch engagement has three modes: approaching, slip-ready, engaging mode. 1) In the approaching mode, the clutch is initially at rest. The controller only considers the positioning work since the clutch is initially disengaged. Hence, the adaptation law (30) for $\hat{\phi}_1$ is not active and (32) and (33) are only activated. 2) When the clutch is positioned to the near contact point, this is called the slip-ready mode. 3) Then, the clutch is positioned to the desired stroke corresponding to the desired force for full engagement.

The clutch makes contact with the opposing surface from the near contact point. The adaptation law (30) is turned on just in order to identify the disk friction coefficient. This is called the engaging mode. All modes utilize one position-based force control so that there is no switching phenomenon in transitions between each mode.

It is assumed that the stroke for the clutch to be engaged is known. This is a reasonable assumption since real vehicles generally have initial stroke search logic. The desired trajectory can be generated by the nonlinear filter [38] such that $\theta_{md}, \dot{\theta}_{md}, \ddot{\theta}_{md} \in \mathcal{L}_\infty$.

1) **Without Adapations**: The actuator position control for engagement without adaptations has been conducted for comparison. Fig. 7(a)-(b) show that the position tracking and the shaft speed synchronization are achieved well. In Fig. 7(c)-(d), the slip-ready position and the contact position in the engaging mode are magnified from Fig. 7(a). Although, the desired actuator position is well tracked by the controller without any adaptation scheme, residual vibrations arise mainly due to the highly stiff contact.

2) **With Adaptations**: With the adaptation laws turned on at each mode, the same experiments have been conducted. Fig. 8(a) show that the actuator position can be controlled by the motor position. It shows that the clutch control starts with

---

Fig. 7. Experimental results of adaptive sliding mode control with DOB: (a) Motor position (actuator stroke). (b) The input shaft (AC motor) and the output shaft speed. (c) Zoom of position response in the slip-ready position. (d) Zoom of position response in the contact position.

Fig. 8. Experimental results of adaptive sliding mode control with DOB: (a) Motor position (actuator stroke). (b) The input shaft (AC motor) and the output shaft speed. (c) Zoom of position response in the slip-ready position. (d) Zoom of position response in the contact position.

Fig. 9. Experimental results of adaptive sliding mode control with DOB: (a) Adaptation of the disk friction coefficient $\mu$. (b) Adaptation results of motion friction parameters $\phi_1$ and $\phi_2$. 

---

**B. Position Tracking Experiment in Free-Space**

Because the proposed control strategy is based on position control, the high accuracy of the actuator position control is very important to determine the overall performance. The position control experiments in the clutch open phase (free-space) are carried out without considering the clutch contact in order to verify only the motion control performance only.

This experiment considers four different types of controllers which are proportional-derivative (PD), sliding mode control (SMC), adaptive sliding mode control (ASMC), and adaptive sliding mode control with disturbance observer (ASMC-DOB). The position tracking results and the tracking errors are shown in Fig. 6. They clearly show the difference in tracking capability among them.

The PD control result has irregular tracking errors due to the lack of robustness. Since it does not consider the model information, the increasing feedback gains may lead to large transient errors. Even for the SMC, tracking performance is not satisfactory because high gain control input may yield actuator saturations. Even though the tracking error gradually decreases, abrupt overshoot errors are observed. The ASMC-DOB shows a good tracking performance compared with aforementioned both controllers. However, it shows tracking errors during the transient period.

The ASMC-DOB shows that the tracking performance and robustness are effectively improved compared with other results. In this experiment, $\tau_q$ is selected as 0.016 which is 10Hz Q-filter bandwidth. It can overcome the hysteretic motion friction and unmodeled effect. And, the settling time is reduced to 0.25s during which the clutch actuator stroke moves 3 mm away from free motion to clutch engagement. This is satisfactory for design specification of production vehicles. Therefore, experimental results confirm that the ASMC-DOB is the best candidate for the position control. The rms error and the maximum error are shown in Table. I.

---

**Table I.** Tracking Error Performance

<table>
<thead>
<tr>
<th>Controller</th>
<th>Tracking Error (mm)</th>
<th>Maximum Error (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>PD</td>
<td>0.40</td>
<td>0.55</td>
</tr>
<tr>
<td>SMC</td>
<td>0.25</td>
<td>0.35</td>
</tr>
<tr>
<td>ASMC</td>
<td>0.10</td>
<td>0.15</td>
</tr>
<tr>
<td>ASMC-DOB</td>
<td>0.05</td>
<td>0.10</td>
</tr>
</tbody>
</table>
the clutch open phase, which corresponds to the approaching mode under the position control only. It can be also verified in Fig. 8(c). Approximately at 4 sec, the clutch is moved into the near contact position and enters into the slip-ready mode. Then, once the engaging mode starts at 6 sec, the clutch slip is reduced. Fig. 8(b) shows that two shaft speeds are synchronized.

Since the friction coefficient cannot be measured directly, the torque measurement from the torque sensor is utilized for verification. With these values, the disk friction coefficient is obtained indirectly based on (6) incorporating the normal force which depends on the actuator position. The friction coefficient \( \mu \) on the disk is estimated as shown in Fig. 9(a). In this test, the friction coefficient \( \hat{\mu} \) is initially set by 0.29. It was taken from the repeated experiment at the test bench. In Fig. 9(a), it is verified the estimation \( \hat{\mu} \) varies with the clutch slip speed.

Compared with the experimental measurement, it shows that the proposed \( \mu \) adaptation works reasonably well. The tracking error of the clutch normal force based on the position control can be compensated through the estimation of the friction coefficient on the disk. The closer look at the clutch contact from Fig. 8(a) can be found in Fig. 8(d) where the desired position is modified online as a function of the disk friction coefficient. It can be also verified in (20) and Fig. 4. In Fig. 8(d), the peaking error occurs due to the transition effect from a free space to a stiff contact. This impact reaction may go beyond the DOB bandwidth specified by \( \tau_q \) in (47). It should be noted that the disk friction coefficient \( \hat{\mu} \) is estimated in the slip phase so that \( \Phi(\hat{\mu}) \) in (16) can be subsequently adapted as well.

In Fig. 8, Tracking errors are observed due to the instantaneous contact impact. Fig. 9(a) and Fig. 8(d) show that \( \hat{\mu} \) adaptation yields friction coefficient monitoring and engagement compensation simultaneously. In other words, while the disk friction coefficient is estimated, the desired position depending on the clutch normal force is also corrected at the same time. As a result, this adaptation scheme provides online stiffness inference for the clutch engagement control. Here, \( \hat{r} \) can be considered as the equivalent stiffness of the engagement force.

Fig. 9(b)-(c) shows the adaptation results of the motion friction parameters in the clutch actuator model. In particular, both parameters \( \hat{\phi}_1 \) and \( \hat{\phi}_2 \) are estimated in the dynamic regime of the clutch actuator positioning.

**Remark 2:** It should be noted that the test bench does not have any external damper to absorb residual vibrations of the drivetrain. Hence, this may impose the worst case for testing any clutch apparatus. If the system has a mass flywheel, the control quality of the engagement will be further improved significantly.

V. Conclusion

In this paper, the control strategy for a SECA system is developed based on ASMC with DOB compensation. The proposed adaptation algorithm considers not only parametric uncertainties but also robustness of the clutch engagement. The desired force is adjusted by monitoring the disk friction coefficient in real time. The experimental results show that the control performance can be improved significantly when the controller is combined with the parameter adaptation and the disturbance compensation. The proposed engagement control based on actuator position tracking can be used for the general clutch actuator systems in automotive applications as well as a SECA system.

**References**


Jinsung Kim (S’10-M’13) received the Ph.D. degree in mechanical engineering from KAIST (Korea Advanced Institute of Science and Technology), Daejeon, Korea, 2013. He joined Division of Research and Development, Hyundai Motor Company, Korea, in 2013, where he is currently a Senior Research Engineer in development of powertrain control software. He designed clutch control algorithms of a 7-speed dual clutch transmissions for production vehicles. His current research interests include automotive systems, nonlinear control, observer design, and integrated design and control of complex mechanical systems.

Seibum B. Choi (M09) received his B.S. degree in mechanical engineering from Seoul National University, his M.S. degree in mechanical engineering from KAIST, and his Ph.D. degree in control from the University of California, Berkeley, in 1993. From 1993 to 1997, he was involved in the development of automated vehicle control systems at the Institute of Transportation Studies, University of California. Through 2006, he was with TRW, MI, USA, where he was involved in the development of advanced vehicle control systems. Since 2006, he has been faculty in the Mechanical Engineering Department, KAIST. His current research interests include fuel saving technology, vehicle dynamics and control, and active safety systems.

Jiwon J. Oh obtained his B.S. and M.S. degrees, and Ph.D. in mechanical engineering from KAIST. He is currently a senior research engineer in Hyundai Motor Company. His research interests include vehicle driveline state estimation and hybrid powertrain clutch control.