Nonlinear Estimation Method of a Self-Energizing Clutch Actuator Load

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Abstract— **Clutch torque estimation is an important problem in automotive transmission control. The self-energizing clutch actuator system has a novel mechanism to amplify the clutch torque for reducing actuation energy. The nonlinear disturbance observer is designed such that the clutch torque is estimated with guaranteeing asymptotic convergence by using the actuation motor measurement signal. The simulation result shows that the developed method is useful to identify the load characteristics of a self-energizing clutch actuator system in comparison with the result of proportional-integrative type observer.**

I. Introduction

In automotive powertrain systems, actuator control performance is the main issue in automotive transmissions during gear-shifting and launch operation. Particularly in dry-clutch cases, this problem is closely related to how well a clutch torque control works. To obtain an effective solution of a clutch control, the clutch torque information during engagement should be needed for a feedback control design. But, direct clutch torque measurement is very difficult because torque transducers are quite expensive for practical engineering applications. Moreover, the generated torque is detected only when both sides of the clutch disks remain slipping. One possible solution is to make an observer in order to estimate unknown reaction torque with respect to the actuator dynamics. This solution is very effective approach because it only requires output measurements.

For clutch load estimation in a view of actuator dynamics, there are some researches in which electro-pneumatic and hydraulic actuators are used for clutch control systems. For electro-pneumatic actuators, clutch load characteristics are investigated for preliminary study of a load estimation scheme [1]. Adaptive observers are designed for estimation of the clutch load, the friction coefficient, and the actuator velocity by a reduced order type [2], and full order type [3], respectively. Based upon estimated states from such observers, a dual-mode switched control is proposed [4], [5]. For electro-hydraulic actuators, a clutch pressure observer is designed for automatic transmissions (ATs) with clutch-to-clutch technology [6], [7]. Disturbances are considered as unknown inputs and the resulting observer is input-to-state stable.

Another class of research for clutch torque estimation is driveline model-based approach. Sliding mode observer is employed to identify the shaft torque [8]. Similarly, adaptive sliding observer estimates the turbine torque with torque converter parameters in ATs [9]. Neural network based nonlinear observer is suggested incorporating extended Kalman filter as training algorithm [10]. The shaft torque is estimated by using driveline dynamics dividing the engagement procedure into the stick phase and the slip phase [11].

Compared with results for clutch actuator systems with electro-pneumatic or hydraulic actuators, the little attention has been paid for electromechanical actuator cases. Until now, many automotive manufacturers seem to prefer hydraulic and electro-pneumatic actuators due to some advantages such as force multiplication, and lubrication. For these reasons, a clutch actuator system controlled by electromechanical actuators is at an early stage.

In this paper, some effort focuses on the systematic design of a load estimation of electro-mechanically controlled clutch actuator system particularly for the case where the system has a self-energizing mechanism [12]. Within an actuator control framework, the nonlinear observer for estimating the load torque of the actuator will be designed by using the motor position measurement only. It enables us to design the estimation system without considering vehicle driveline information.

The control issue of self-energizing clutch actuator systems (SECA) is large reaction torque when the clutch is in contact with. It implies that the torque phase is very short and the reaction torque to the actuator has impact-like behavior. Thus, the position-based force control has limitations because of its engagement characteristic that is highly sensitive to the actuator position near the contact point. Thus, the development of the clutch torque observer will be useful for the clutch engagement control design purpose. A SECA system has self-energizing mechanism whose characteristic is nonlinear, and instantaneous transition during contact is problematic. The linear disturbance observer [13] has some limitation to overcome such problems because of noise sensitivity.

The proposed nonlinear disturbance observer has the Robust Integral of the Sign of the Error (RISE) structure that is recently developed robust control method [14]. The RISE technique has been originally considered to guarantee continuous asymptotic tracking. It is highly robust to unstructured uncertainties and has implicit learning characteristics [15] [16]. This method has been used to identify unknown parameters and detect fault signal in dynamic systems as stated in literature [17] [18]. By inspiration from such a property, nonlinear disturbance observer based on the RISE technique is designed to estimate the reaction torque of the

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clutch. Simulation results will show the effectiveness of the proposed method.

The outline of this paper is as follows. A brief introduction of a SECA system is given in section II. In section III, nonlinear disturbance observer for a clutch torque is developed and analyzed. In section IV, simulation results illustrate the clutch torque estimation by the proposed method. Finally, conclusion and future works are discussed in section V.

II. Problem Formulation

A. System Overview

A self-energizing clutch actuator system (SECA) is recently developed as a new clutch mechanism for Automated Manual Transmissions (AMT) and Dual Clutch Transmissions (DCT) [12], [19], [20]. It can generate an additional force/torque to engage a clutch disk. The ability to amplify the engagement torque is dependent on a self-energizing mechanism that has some wedge structure as shown in the left part of Fig. 1. The implementation of such a mechanism can be realized by a rack and pinion gear-set that enables a system to manipulate the clutch disk and gives a wedge structure at the same time. The resulting system can be operated without having diaphragm spring mechanism of conventional systems. Such a structure can be analyzed as a self-energizing mechanism.

Since this system is driven by an electric motor and a dry friction disk, the engagement torque measurement of the clutch disk is difficult. Moreover, industrial manufacturers do not allow expensive torque sensors. Thus, a nonlinear clutch torque observer should be required for control design and monitoring by utilizing the system dynamics.

B. Full System Model for Simulations

In the following, a model of the dynamics of motion is presented to describe the system behavior.

1) Dynamic Model of the Electric Motor: The motor electric and mechanical equations are as follows, respectively.

$$L_m \dot{i}_m + R_m i_m + k_m \omega_m = u \tag{1a}$$

$$J_m \dot{\omega}_m + T_{fm}(\omega_m, z_m) + \frac{T_a}{N_g} = k_t i_m \tag{1b}$$

In (1a), u is the voltage applied to the motor, i_m the motor current, R_m the resistance, L_m the inductance, and k_m the back electromotive force constant. In (1b), J_m is the motor moment of inertia, ω_m the rotor speed, k_t the motor torque constant, N_g the gear ratio between the motor rotor and the mechanical subsystem. Note that T_a is the driving torque for the mechanical subsystem, which will be introduced. The nonlinear motor friction torque T_{fm} is modeled by the LuGre friction model [21] as

$$\begin{aligned} T_{fm}(\omega_m, z_m) &= \sigma_{0m} z_m + \sigma_{1m} \dot{z}_m + \sigma_{2m} \omega_m \\ \dot{z}_m &= \omega_m - \frac{|\omega_m|}{g_m(\omega_m)} z_m \\ \sigma_{0m} g_m(\omega_m) &= f_{cm} + (f_{sm} - f_{cm}) e^{-(\omega_m/\omega_{ms})^2} \end{aligned} \tag{2}$$

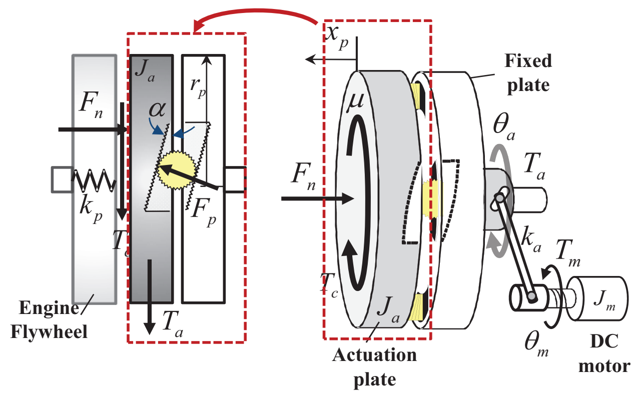


Fig. 1. Schematic of the equivalent clutch actuator model with self-energizing mechanism.

where, z denotes the internal friction state, g_m Stribeck effect function, f_{cm} Coulomb friction, f_{sm} the static friction level, and ω_{ms} Stribeck velocity. In (2), σ_{0m}, σ_{1m}, and σ_{2m} are unknown friction parameters that represent bristle stiffness, bristle damping, and viscous coefficient, respectively.

2) Dynamic Model of the Mechanical Subsystem: The fixed plate shown in Fig. 1 is interposed between the clutch cover and the friction disks in order to adjust the axial displacement of the actuation plate while rotating simultaneously. When the clutch is in contact with the opposite surface for engagement operation, the equation of motion for the actuation plate in the slip phase is represented by

$$J_a \dot{\omega}_a = T_a + T_c - 2 r_p F_p \sin\alpha - T_{fa}(\omega_a, z_a) \tag{3}$$

where, J_a is the moment of inertia of the actuation plate, T_c the clutch torque, F_p the reaction force on the rack and pinion surface, r_p the radius of bevel gear position, and α the inclined surface angle on the actuation plate and the fixed plate, respectively. 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

$$T_a = k_a \left(\frac{\theta_m}{N_g} - \theta_a \right) \tag{4}$$

where θ_m and θ_a are the motor and the actuator angular position, respectively. The equivalent torsional stiffness k_a of the mechanical link includes the torsional stiffness of the ball-screw and the lever part. In (3), the frictional torque T_{fa} on the worm shaft is represented by

$$\begin{aligned} T_{fa}(\omega_a, z_a) &= \sigma_{0a} z_a + \sigma_{1a} \dot{z}_a + \sigma_{2a} \omega_a \\ \dot{z}_a &= \omega_a - \frac{|\omega_m|}{g_a(\omega_a)} z_a \\ \sigma_{0a} g_a(\omega_a) &= f_{ca} + (f_{sa} - f_{ca}) e^{-(\omega_a/\omega_{as})^2} \end{aligned} \tag{5}$$

where, all parameter notations are the same as those in (2) with the use of the subscript 'a' instead of 'm'. For the positive slip phase, the clutch torque T_c is obtained as

$$T_c = \mu R_c F_n \tag{6}$$

where, μ is the dry friction coefficient, R_c the clutch radius, and F_n the applied normal force. Due to the presence of the rack and pinion mechanism, the relationship between F_p and F_n is geometrically determined by the inclined surface angle α as shown in Fig. 1. As a result, F_p is:

$$F_p = \frac{F_n}{\cos\alpha}. \quad (7)$$

Note that the third term at the right hand side in (3) is related with self-energizing effect. The wedge structure from the rack and pinion mechanism can induce the reaction force F_p on the surface of the actuation plate and the fixed plate with respect to the applied normal force F_n.

The axial displacement of the actuation plate can be calculated through the geometric relation as shown in Fig. 1. It is therefore given by

$$x_p = 2r_p\theta_a \tan\alpha \quad (8)$$

where, θ_a is the angular position of the actuation plate which can be obtained from integration of (3). The normal force applied on the friction disk is

$$F_n = k_p x_p = 2k_p r_p \theta_a \tan\alpha \quad (9)$$

where, k_p is the stiffness of the actuation plate. The normal force F_n is proportional to the actuator stroke x_p in axial direction [12].

According to (8), (9), and (7), then the actuator dynamics (3) in the positive slip phase can be rewritten as

$$J_a\dot{\omega}_a = T_a + \mu R_c F_n - 2r_p \tan\alpha F_n - T_{fa}(\omega_a, z_a). \quad (10)$$

C. Simplified Model for Control

The full order model provides a detailed description of the system so that it is sufficiently reliable for the simulation purpose. However, it is not suitable for designing estimators and controllers due to the complexity. Considering the computation issues, the simplified model is developed with the following system properties.

P1) The electrical dynamics of the DC motor is faster than the mechanical motions.
P2) The bandwidth of the actuation plate is very high, and the rotating angle of it is very small.
P3) In mechanical subsystem, frictional motion at pre-sliding can be negligible.

P1 means that the inductance of the DC motor could be neglected by which L_m is much smaller than J_m. Thus, (1a) is simplified into the following algebraic equation.

$$u = R_m i_m + k_m\omega_m \quad (11)$$

From P2, we assume that the actuation plate is rigidly connected with the rotor of the electric motor. Consequently, (10) can be rewritten as

$$2r_p F_n \tan\alpha = T_a + \mu R_c F_n. \quad (12)$$

Combining (4) and (12), it can be shown that the clutch normal force is a function of compliance between the motor and the mechanical subsystem:

$$F_n = \frac{k_a\left(\theta_m/N_g - \theta_a\right)}{2r_p\tan\alpha - \mu R_c} \quad (13)$$

According to the expressions (9) and (13), the angular deflection between the motor θ_m and the actuation plate θ_a is given by

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

where the auxiliary function $k_b(\mu)$ is defined for notational simplicity as

$$\beta \triangleq 2r_p \tan\alpha. \quad (15a)$$

$$k_b(\mu) \triangleq \xi(\mu)(2r_p k_p \tan\alpha) = \xi(\mu)k_p\beta, \quad (15b)$$

$$\xi(\mu) \triangleq (2r_p\tan\alpha - \mu R_c) = (\beta - \mu R_c). \quad (15c)$$

In (15b) and (15c), k_b and ξ are unknown parameters because they include the friction coefficient μ that varies with materials, a clutch slip speed and temperature. Substituting (14) into (13) derives an equation for the relationship between the normal force and the motor position.

$$F_n = \frac{k_a}{\xi(\mu)}\left[\frac{k_b(\mu)}{N_g\left(k_b(\mu) + k_a\right)}\right]\theta_m \quad (16)$$

From P3, the steady-state friction model can be obtained. Since the actuator motion is behaved very fast, the internal state z in (5) and (17) is negligible. Note that describing the motion at pre-sliding may lead to large initial error and peaking phenomenon in the clutch torque observer. Accordingly, the following static friction model depending only on the velocity is given by

$$T_f(\omega_m) = \phi_1 sgn(\omega_m) + \phi_2\omega_m \quad (17)$$
$$\phi_1 \triangleq \sigma_0 g(\omega_m), \quad \phi_2 \triangleq \sigma_2$$

where σ_0 and σ_1 are dominant friction parameters obtained from the full model. The difference between (2), (5) and (17) is not too significant because the result of the simplification only disregard the effect near zero velocity.

Finally, the simplified model can be derived with respect to the motor states that are measurable. Combining equation (1b), (11), (12), (14), and (17) yields the following second order dynamics:

$$\bar{J}_m\dot{\omega}_m + q\omega_m + T_f(\omega_m) + r(\mu)\theta_m = pu \quad (18)$$

where $\bar{J}_m$ is the equivalent moment of inertia, and other auxiliary variables p, q and w are defined as

$$\bar{J}_m \triangleq J_m + \frac{J_a}{N_g^2}, \qquad p \triangleq \frac{k_t}{R_m},$$
$$q \triangleq \left(\frac{k_t k_m}{R_m} + b_m\right), \qquad w \triangleq \frac{k_a}{{N_g}^2},$$
$$r(\mu) \triangleq \frac{k_a k_b(\mu)}{{N_g}^2[k_b(\mu) + k_a]} = w\left(\frac{k_b(\mu)}{k_b(\mu) + k_a}\right).$$

The state space representation for the simplified system (18) is given as follows:

$$\begin{aligned} \dot{x}_1 &= x_2 \\ \dot{x}_2 &= \frac{1}{\bar{J}_m}\left[pu - q\omega_m - T_f(\omega_m) - r(\mu)\theta_m\right] \end{aligned} \quad (19)$$

And, (19) is rewritten by the compact form:

$$\dot{x} = f(x, u, t) + Fd(t) \quad (20)$$

where the state variables $x = [x_1,\ x_2]^T = [\theta_m,\ \omega_m]^T \in \mathbb{R}^2$, the disturbance input gain vector $F = [0, 1]^T$, and the disturbance $d = r(\mu)\theta_m/\bar{J}_m$. The nominal function vector $f(x, u, t) = [f_1(x, u, t), f_2(x, u, t)]^T \in \mathbb{R}^2$ is $f_1(x, u, t) = x_2$ and $f_2(x, u, t) = \frac{1}{\bar{J}_m}\left[pu - q\omega_m - T_f(\omega_m)\right]$.

Note again that this simplified model is for the purpose of the clutch torque observer design. The system parameters used in this study were validated experimentally in [12].

D. Objective

The objective of this paper is to design a nonlinear estimator of the load of a SECA system, which corresponds to the clutch torque. The value of $r(\mu)\theta_m$ in (18) implicitly includes the engagement torque and uncertain effect when the clutch comes into contact. Although the clutch torque is obtained by (6) and (16), it is difficult to find accurate value because $k_b(\mu)$ uncertain in practice. In (18), the value of $r(\mu)\theta_m$ will be identified as a reaction load of the actuator.

The available measurements are the motor voltage as an input, and the motor position and the motor velocity as outputs. The motor position is measured by the hall-sensor based encoder. The velocity signal can be obtained from numerical differentiation between two consecutive position signals.

III. Clutch Torque Observer Design

A. Proportional-Integrative Disturbance Observer

For a comparison study, this section first introduces a simple proportional-integrative (PI) disturbance observer [22], [23] for a clutch torque estimation. Consider the PI disturbance observer for the system (20) of the form:

$$\hat{d} = \gamma_0(F^+x - z), \quad (21)$$

$$\dot{z} = F^+f(x, u, t) + \hat{d} \quad (22)$$

where $z \in \mathbb{R}$ is an auxiliary observer state, $\hat{d} \in \mathbb{R}$ an unknown disturbance and $\gamma_0 \in \mathbb{R}$ is a design parameter. For the estimation error $\tilde{d} := d - \hat{d}$, error dynamics can be obtained as $\dot{\tilde{d}} = -\gamma_0\tilde{d} + \dot{d}$. It can be shown that estimation error dynamics is exponentially stable under the assumption that the disturbance has small variation.

B. Robust Asymptotic Nonlinear Disturbance Observer

The RISE-based nonlinear disturbance observer is designed with following assumptions.

Assumption 1: The nominal system vector field $f(x, u, t)$ is known *a priori*.

Assumption 2: There exist positive constants ρ_1 and ρ_2 such that given disturbances are twice continuously differentiable and have bounded derivatives with respect to time, i.e. $|\dot{d}(t)| < \rho_1, |\ddot{d}(t)| < \rho_2$.

Assumption 1 is for the knowledge of the nominal system dynamics. With this, the nominal system can be reconstructed by the initial condition $x(0)$. Assumption 2 presents the condition for applying a nonlinear disturbance observer that will be determined later.

In the general framework, the auxiliary velocity observer is developed as

$$\dot{\hat{x}}_2 = f(x, \dot{x}, u, t) + \hat{d} \quad (23)$$

where $\hat{x}_2$ is the estimated velocity state and $\hat{d}$ is the estimated disturbance. For the system (19), the velocity observer is given as

$$\dot{\hat{x}}_2 = \frac{1}{\bar{J}_m}\left[pu - qx_2 - T_f(x_2)\right] - \hat{d} \quad (24)$$

where $\hat{d}(0) = r(\mu(0))\theta_m(0)/\bar{J}_m$. Although disturbance d depends on the actuator position, it is often inaccurate because the contact point of the clutch disk in uncertain. Hence, the strategy of this research is that the reaction torque of the clutch is taken into account as lumped disturbance d.

Let $e_2 \in \mathbb{R}$ be the velocity estimation error between (19) and (24), i.e.

$$e_2 := x_2 - \hat{x}_2. \quad (25)$$

Using the time derivative of e_2, (19), and (24), the error dynamics can be written as

$$\dot{e}_2 = d - \hat{d} =: \tilde{d} \quad (26)$$

where $\tilde{d}$ is the disturbance error as defined above. It is clear that the error dynamics of the auxiliary velocity observer is a nonlinear function of the disturbance estimation error under Assumption 1 holds.

The nonlinear disturbance observer $\hat{d} \in \mathbb{R}$ based on a RISE technique is designed as

$$\hat{d} = \int_0^t (K_0 + 1)\alpha e_2(\tau)d\tau + \int_0^t K_1 \mathrm{sgn}(e_2(\tau))d\tau + (K_0 + 1)e_2(t) - (K_0 + 1)e_2(0) \quad (27)$$

where K_0 and K_1 are design parameters to be determined. By taking the time derivative of (27), it can be shown as

$$\dot{\hat{d}} = (K_0 + 1)r + K_1 \mathrm{sgn}(e_2(t)) \quad (28)$$

Define the filtered estimation error $r \in \mathbb{R}$ as

$$r := \dot{e}_2 + \alpha e_2 \quad (29)$$

where α is the filter design parameter. With (26), the time derivative of (29) is written as

$$\dot{r} = \dot{d} + \dot{\hat{d}} + \alpha\dot{e}_2. \quad (30)$$

The estimation error system can be obtained by substituting (27) into (30) as

$$\dot{r} = \dot{d} - (K_0 + 1)r - K_1 \mathrm{sgn}(e_2(t)) + \alpha\dot{e}_2. \quad (31)$$

Here, the following theorem states convergence of nonlinear disturbance observer (27).

Theorem 1: Under Assumption 1 and 2, the disturbance estimate $\hat{d}$ in the auxiliary observer system (24) converges to the actual value d in the actual plant (19) in the sense that If the nonlinear observer gain is chosen as

$$K_1 > \rho_1 + \rho_2 \tag{32}$$

with the filter parameter $\alpha = 1$.

Proof: Let $P(t) \in \mathbb{R}$ be a positive function defined as

$$P(t) := \zeta_d - \int_0^t N(\tau)d\tau \tag{33}$$

where $\zeta_d := K_1 e_2(0) - e_2(0)\dot{d}(0)$ and $N(\tau) \in \mathbb{R}$ is a function defined as

$$N(t) := r\left[\dot{d} - K_1 \text{sgn}(e_2(t))\right]. \tag{34}$$

Integration of (34) from 0 to t yields

$$\int_0^t N(\tau)d\tau = \int_0^t \dot{e}_2(\tau)\dot{d}(\tau)d\tau - \int_0^t \dot{e}_2(\tau)K_1 \text{sgn}(e_2(\tau))d\tau + \int_0^t \alpha e_2(\tau)\left[\dot{d} - K_1 \text{sgn}(e_2(\tau))\right]d\tau \tag{35}$$

The first term in the right hand side in (35) can be integrated by parts. And, the second term is also integrated. Then, we have that

$$\int_0^t N(\tau)d\tau = e_2(t)\dot{d}(t) - e_2(0)\dot{d}(0) - K_1 e_2(t) + K_1 e_2(0) + \int_0^t \alpha e_2(\tau)\left[\dot{d} - \ddot{d} - K_1 \text{sgn}(e_2(\tau))\right]d\tau. \tag{36}$$

Using the definition of ζ_d, (36) can be upper bounded as

$$\int_0^t N(\tau)d\tau \leq |e_2(t)|\left[\dot{d}(t) - K_1\right] + \zeta_d + \int_0^t \alpha|e_2(\tau)|\left[|\dot{d}| + |\ddot{d}| - K_1\right]d\tau. \tag{37}$$

By applying the sufficient condition (32), the square bracket terms in (37) can be eliminated so that $N(t)$ satisfies

$$\int_0^t N(\tau)d\tau \leq \zeta_d. \tag{38}$$

This result clearly shows that there exists a positive function $P(t)$ in (33). Consider a positive definite Lyapunov function candidate $V(e_2, r, t) \in \mathbb{R}$.

$$V(e_2, r, t) = \frac{1}{2}e_2^2 + \frac{1}{2}r^2 + P(t). \tag{39}$$

The time derivative of $\dot{V}$ is obtained as

$$\dot{V}(e_2, r, t) = e_2\dot{e}_2 + r\dot{r} + \dot{P}(t). \tag{40}$$

Substituting (29) and (31) into (40) yields

$$\dot{V}(e_2, r, t) = e_2(r - \alpha e_2) + \dot{P}(t) + r\left[\dot{d} - (K_0 + 1)r - K_1 \text{sgn}(e_2(t)) + \alpha\dot{e}_2\right]. \tag{41}$$

Using the fact that the time derivative of (33) is $\dot{P}(t) = -N(t)$, (41) is rewritten as

$$\dot{V}(e_2, r, t) = -\alpha e_2^2 + (1-\alpha)e_2 r - (1-\alpha)\dot{e}_2 r - k_0 r^2 - N(t) + r\left[\dot{d} - K_1 \text{sgn}(e_2(t))\right]. \tag{42}$$

By setting $\alpha = 1$ and cancellation of the last two terms, (42) is rewritten by

$$\dot{V}(e_2, r, t) \leq -e_2^2 - k_0 r^2. \tag{43}$$

From (39) and (43), e_2 and $r \in \mathcal{L}_\infty$. It is clear that $\dot{e}_2 \in \mathcal{L}_\infty$, and e_2, $r \in \mathcal{L}_2$. Since the condition (32) is satisfied and Assumption 2 holds, $\dot{d} \in \mathcal{L}_\infty$. Therefore, the estimation error dynamics (31) is bounded so that $\dot{r} \in \mathcal{L}_\infty$. From this observation, $\dot{V}(e_2, r, t)$ is uniformly continuous. Barbalat's lemma can be used to show that $\dot{V} \to 0$. It implies that $e_2 \to 0$ and $r \to 0$ as $t \to \infty$. Based on (26), it can be concluded that $\hat{d} \to d$. ∎

Remark 1: The construction of (24) aims at designing a disturbance estimator (27). The initial error $e_2(0)$ of the velocity state estimate may cause a peaking signal of the disturbance estimate $\hat{d}$. Therefore, the initial value $\hat{x}_2(0)$ should be carefully chosen. Since the actuator system is initially at rest, $\hat{x}_2(0)$ can be easily selected as zero.

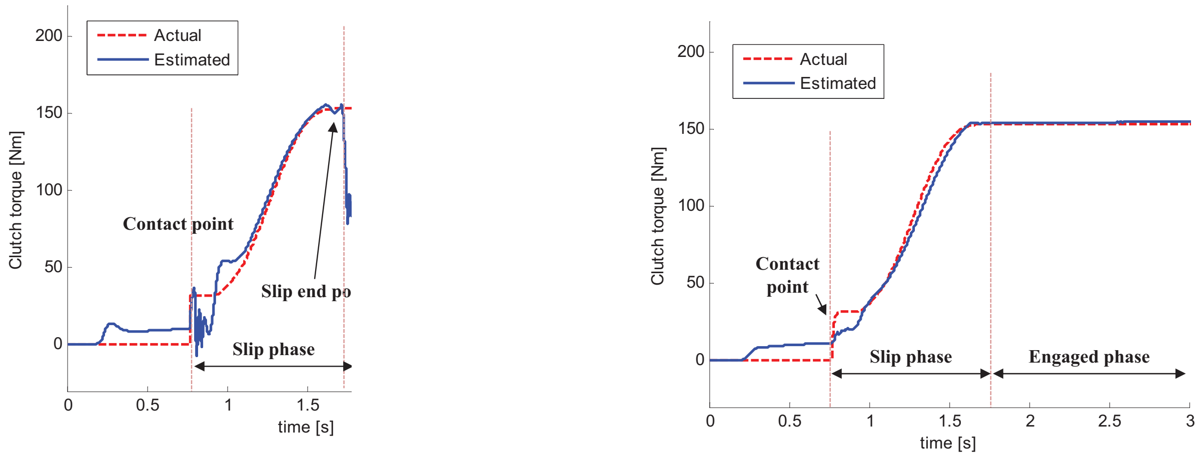
Remark 2: The nonlinear disturbance observer proposed above is taken into account as a high gain approach. The gain selection is an important task because too large gain makes the estimation system unstable. Theorem 1 gives the sufficient condition (32) for gain K_1. It is clear that the knowledge of ρ_1 and ρ_2 for determining K_1 can be obtained from the clutch torque magnitude that depends on the engine torque.

IV. Simulation Results

In this section, the proposed observer is applied to the SECA system. For the underlying engagement operation, the clutch position profile is predetermined with corresponding voltage input.

The simulation result of the PI disturbance observer (21) is shown in Fig. 2 (a). The contact, where the clutch is engaged, occurs at 0.7 sec. After some transient responses, the clutch torque is estimated well. But, steady-state error is significantly large once the clutch is completely engaged at 1.7 sec. This is mainly due to the lack of information for estimation when the clutch is completely engaged with zero velocity. Hence, PI disturbance observer does not guarantee convergence of the clutch torque estimation. Convergence errors cannot be solved even though higher order PI observers are applied.

On the contrary, the simulation result of the disturbance observer (27) in Fig. 2 (b) shows good estimation result in steady-state response and the transient response as well. The parameters for simulation are given as $K_0 = 0.1$ and $K_1 = 4.3$. The contact also occurs at 0.7 sec. The initial error of the estimation until 0.7 sec is due to the compliance error by the model reduction process, which is not physically meaningful. The clutch torque is estimated well in slip phase



(a) (b)

Fig. 2. Simulation results of clutch torque estimation in the actuator position control: actuator torque signal (dashed) and estimated torque value (solid), (a) PI disturbance observer (21) (b) Nonlinear disturbance observer (27).

although the self-energizing mechanism in the actuator gives the hard environment during contact situation. Moreover, this estimation is also valid in the engaged phase because the method is based on the actuator load identification. This may be useful setting for continuous feedback control design of the clutch engagement.

V. Conclusion

Nonlinear disturbance observer is designed to estimate the clutch reaction torque of a SECA system. The proposed method is applicable to other manipulator systems that include uncertainties and requires asymptotic convergence. In future works, a sensorless engagement control of a self-energizing clutch actuator system can be developed by using the proposed reaction torque observer.

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