

Estimation of Vehicle Clutch Torque Using Combined Sliding Mode Observers and Unknown Input Observers

Kyoungseok Han¹, Seibum Choi^{2*} and Jiwon Oh³

¹School of Mechanical Aerospace Systems Engineering, Division of Mechanical Engineering, KAIST, Daejeon, Korea

(Tel: +82-42-350-4160; E-mail: hks8804@kaist.ac.kr)

²School of Mechanical Aerospace Systems Engineering, Division of Mechanical Engineering, KAIST, Daejeon, Korea

(Tel: +82-42-350-4160; E-mail: sbchoi@kaist.ac.kr) * Corresponding author

³School of Mechanical Aerospace Systems Engineering, Division of Mechanical Engineering, KAIST, Daejeon, Korea

(Tel: +82-42-350-4160; E-mail: jwo@kaist.ac.kr)

Abstract: Precise clutch control is a crucial factor in determining the ride quality of the production vehicle and its performance has been enhanced by accurate estimation of involved clutch torque or pressure, especially during gear changing phase (torque phase / inertia phase). Unfortunately, torque or pressure sensor is not available in production cars because of its cost, installation, and maintenance issues. This study mainly focuses on the accurate estimation of the clutch torques for the vehicles with automatic transmission using sliding mode observers and unknown input observers. Sliding mode observer is selected because of its applicability to nonlinear systems and characteristics of its robustness to model uncertainties. The required values for estimation include the engine map, torque converter characteristic curve, and various vehicle parameters such as gear ratio and inertia. Using the above-mentioned information, sliding mode observer is constructed without additional sensors. The proposed observer is validated by computer simulation using commercial software Matlab & Simulink. The estimated output shaft torque can be used as a reference for optimal clutch control and torque sensor which is not possible in the production car can be replaced with those developed observers.

Keywords: clutch torque, output shaft torque, sliding mode observers, unknown input observers.

1. INTRODUCTION

In both automatic transmission (AT) and automated manual transmission (AMT), the change of gear ratio proceeds while controlling the relevant clutches. This process is commonly referred to as clutch to clutch shift, in which one clutch engages and others disengage.

Information of clutch torque or pressure, especially during a gear change, is crucial factors for enhancing vehicle ride quality. But clutch torque or pressure are unmeasurable values on the majority of production cars because of the cost, installation, and maintenance problem. Therefore, it is required to estimate a clutch torque with accessible values in production cars such as wheel speed, engine speed and output shaft speed with the sensors which are embedded in the car.

For these reasons, several studies have been conducted to estimate various torques for optimal clutch control. [1]–[5], [8]. For example, R. A. Masmoudi et al. [1] utilized sliding mode observers to estimate output shaft torque. However, these estimation scheme doesn't consider clutch torques which are essential to estimate output shaft torque. K. S. Yi et al. [2] employed adaptive sliding mode observers to estimate turbine torque which is output torque of the torque converter. In this paper, however, turbine torque is assumed to be a known value with steady - state torque converter characteristic curves. In

addition to the above-mentioned conventional observers, well known observers like Luenberger and Kalman filter are also used in linear system. However vehicle powertrain model contains highly nonlinear properties, such as engine maps, torque converter characteristics curve, and various loads. In order to overcome these limitations, this paper suggests a model reference sliding mode observer to effectively estimate clutch torque and output shaft torque simultaneously. Several assumptions are applied in this study. First, dual clutch transmission (DCT) based driveline model is utilized to design the observers. Because of the complexity of the automatic transmission structure, only two clutches are considered in this powertrain model. Second, dynamic characteristics of engine and torque converter are not considered in this driveline model, which means only steady - state maps are used. Third, vehicle drives on a flat surface and aerodynamic effect can be negligible.

The organization of this paper is as follows. Section 2 deals with the design of driveline model with automatic transmission(AT) based on dual clutch transmission(DCT). Section 3 shows lumped clutch torque estimation methods using unknown input observers. Section 4 presents sliding mode observer combining with Section 3 results, and finally, Section 5 shows the results of computer simulations with Matlab & Simulink

Nomenclature

J	inertia
ω	angular velocity
θ	rotational angle
T	torque
i	gear ratio
α	engine throttle angle
d	torque ratio
k	capacity factor
k	spring constant
b	damping coefficient
μ_s	static friction coefficient
μ_d	dynamic friction coefficient
F	clamping force
R	clutch effective radius
$\hat{\bullet}$	estimated value

Subscript

$T_{e,p,t,c,L}$	engine, pump, turbine, clutch, load
$J_{e,t,c,o,v}$	engine, turbine, clutch, output shaft, vehicle
$i_{f,t1,t2}$	final, 1 speed, 2 speed

2. DRIVELINE MODELING

The vehicle driveline with automatic transmission(AT) can be simplified based on Dual Clutch Transmission(DCT) for simulations. Conventional [7,9] AT driveline consists of engine, torque transfer devices like torque converter and clutches, transmission (planetary gear sets) and differential gear sets. In this paper, however, it is assumed that vehicle has only two clutches, one clutch engages and others disengages like DCT [3].

Whole structure of DCT based driveline is showed at Fig. 1. The differences between conventional AT are complex structure of a planetary gear sets which simplify to respective two clutches in this DCT based driveline model. It doesn't matter provided that vehicle has only 2-speed gear system and complex structure of whole planetary gear sets doesn't considered in this paper.

Precise AT modeling is out of the research scope those assumption is make sense.

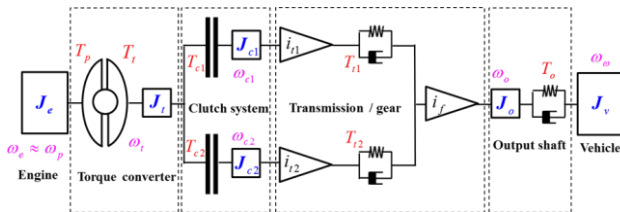


Fig. 1 Driveline based on dual clutch transmission

Engine torque transferred to the wheel based on moment equilibrium relationships as follows.

$$J_e \dot{\omega}_e = T_e(\alpha, \omega_e) - T_p, \quad (1)$$

$$J_t \dot{\omega}_t = T_t - T_{c1} - T_{c2}, \quad (2)$$

$$J_{c1} \dot{\omega}_{c1} = T_{c1} - \frac{T_{t1}}{i_{t1}}, \quad (3)$$

$$J_{c2} \dot{\omega}_{c2} = T_{c2} - \frac{T_{t2}}{i_{t2}}, \quad (4)$$

$$J_o \dot{\omega}_o = i_f(i_{t1}T_{c1} + i_{t2}T_{c2}) - T_o, \quad (5)$$

$$J_v \dot{\omega}_v = T_o - T_L, \quad (6)$$

Where related torques are modeled as follows :

$$T_L = r_w \left\{ \underbrace{m_v g \sin(\theta_{road})}_{road \text{ inclination}} + \underbrace{K_{rr} m_v g \cos(\theta_{road})}_{rolling \text{ resistance}} + \underbrace{\frac{1}{2} \rho v_x^2 C_d A_f}_{aerodynamic \text{ drag}} \right\}, \quad (7)$$

$$T_t = T_r(s)C(s)\omega_e^2, \quad (8)$$

Driveshaft was modeled by spring-damper system which transfers torque by torsional compliance principle.

$$T_{t1} = k_{t1} \left(\frac{\theta_{c1}}{i_{t1}} - i_f \theta_o \right) + b_{t1} \left(\frac{\dot{\theta}_{c1}}{i_{t1}} - \dot{\omega}_o \right), \quad (9)$$

$$T_{t2} = k_{t2} \left(\frac{\theta_{c2}}{i_{t2}} - i_f \theta_o \right) + b_{t2} \left(\frac{\dot{\theta}_{c2}}{i_{t2}} - \dot{\omega}_o \right), \quad (10)$$

$$T_o = k_o (\theta_o - \theta_v) + b_o (\dot{\omega}_o - \dot{\omega}_v), \quad (11)$$

When clutch transferred torque to the other clutch, clutch torque calculated by the wet clutch torque formula as follows.

$$T_{c1} = \mu(\omega_{slip}) R_{c1} F_{n1} = (\mu_s + \mu_d |\omega_{slip}|) \text{sgn}(\omega_{slip}) R_{c1} F_{n1}, \quad (12)$$

$$T_{c2} = \mu(\omega_{slip}) R_{c2} F_{n2} = (\mu_s + \mu_d |\omega_{slip}|) \text{sgn}(\omega_{slip}) R_{c2} F_{n2}, \quad (13)$$

All of those above-mentioned value's name are shown in the nomenclature which is located on the left top of the page.

3. LUMPED CLUTCH TORQUE ESTIMATION USING UNKNOWN INPUT OBSERVERS

3.1 Lumped clutch torque observers

As shown in the Fig. 1 and eq. (2), turbine torque is transferred to the respective clutches. Before designing

respective clutch torque observers, lumped clutch torque observer was designed using Unknown Input Observer (UIO).

Assuming that lumped clutch torque (T_c) is slowly varying state, UIO could be designed as follows based on torque converter dynamics eq. (2).

Assume that $\dot{T}_c \approx 0$

$$\begin{bmatrix} \dot{\hat{\omega}}_t \\ \dot{\hat{T}}_c \end{bmatrix} = \begin{bmatrix} 0 & -\frac{1}{J_t} \\ 0 & 0 \end{bmatrix} \begin{bmatrix} \hat{\omega}_t \\ \hat{T}_c \end{bmatrix} + \begin{bmatrix} \frac{1}{J_t} \\ 0 \end{bmatrix} T_t + \dots \\ \dots + \begin{bmatrix} L_1 \\ -L_2 \end{bmatrix} (\omega_t - \hat{\omega}_t) \quad (14)$$

Where : $T_c = T_{c1} + T_{c2}$

Looking at the eq. (14), UIO looks like first order low pass filter format which doesn't have feed forward terms and the weaknesses of this form are including time delay and uncertainty in unknown input. The characteristics of the UIO are well documented in [4] which assume that the engine torque as unknown input.

Respective clutch torque can be calculated as follows by combining estimated clutch torque (14) and (15) which express torque transfer procedure from clutch to output shaft.

$$(\hat{T}_{c1} i_{t1} + \hat{T}_{c2} i_{t2}) i_f = \hat{T}_o \quad (15)$$

$$\hat{T}_{c1} = \frac{1}{i_{t1} - i_{t2}} \left(\frac{\hat{T}_o}{i_f} - i_{t2} \hat{T}_c \right) \quad (16)$$

$$\hat{T}_{c2} = \frac{1}{i_{t2} - i_{t1}} \left(\frac{\hat{T}_o}{i_f} - i_{t1} \hat{T}_c \right) \quad (17)$$

Eq. (16,17), output shaft torque estimation is needed to estimate respective clutch torque. In other words all the relevant torques are strongly coupled. Therefore Section 4 proposed sliding mode observer to estimate output shaft torque with a measurable values in production car.

3.2 Stability analysis

Designed UIO observer has a one correction term (turbine speed) which makes not only estimated lumped clutch torque converges to the actual value but also turbine speed could track the actual turbine speed.

Designed observer has an own internal dynamics because of insufficient correction term compared to model states. The stability of designed UIO observer is validated through error dynamics.

Actual system (\dot{x})

$$\begin{bmatrix} \dot{\omega}_t \\ \dot{T}_c \end{bmatrix} = \begin{bmatrix} 0 & -\frac{1}{J_t} \\ 0 & 0 \end{bmatrix} \begin{bmatrix} \omega_t \\ T_c \end{bmatrix} + \begin{bmatrix} \frac{1}{J_t} T_t \\ 0 \end{bmatrix} \quad (18)$$

Unknown Input observer (\hat{x})

$$\begin{bmatrix} \dot{\hat{\omega}}_t \\ \dot{\hat{T}}_c \end{bmatrix} = \begin{bmatrix} 0 & -\frac{1}{J_t} \\ 0 & 0 \end{bmatrix} \begin{bmatrix} \hat{\omega}_t \\ \hat{T}_c \end{bmatrix} + \begin{bmatrix} \frac{1}{J_t} T_t + L_1(\omega_t - \hat{\omega}_t) \\ -L_2(\omega_t - \hat{\omega}_t) \end{bmatrix} \quad (19)$$

Error dynamics ($\tilde{x} = \hat{x} - x$)

$$\begin{bmatrix} \dot{\tilde{\omega}}_t \\ \dot{\tilde{T}}_c \end{bmatrix} = \begin{bmatrix} 0 & -\frac{1}{J_t} \\ 0 & 0 \end{bmatrix} \begin{bmatrix} \tilde{\omega}_t \\ \tilde{T}_c \end{bmatrix} - \begin{bmatrix} L_1 & 0 \\ -L_2 & 0 \end{bmatrix} \begin{bmatrix} \tilde{\omega}_t \\ \tilde{T}_c \end{bmatrix} \\ = \begin{bmatrix} -L_1 & -\frac{1}{J_t} \\ L_2 & 0 \end{bmatrix} \begin{bmatrix} \tilde{\omega}_t \\ \tilde{T}_c \end{bmatrix} \quad (20)$$

Stable for $L_1 > 0$, $L_2 > 0$

$$(S + L_1)S + \frac{L_2}{S} = S^2 + L_2S + \frac{L_2}{S} = 0$$

From the above error dynamics, two poles can be set arbitrarily through pole-placement with a linear gains. (L_1, L_2)

4. CLUTCH TORQUE OBSERVERS USING SLIDING MODE OBSERVERS

4.1 Reduced order model

The clutch torque observer is designed based on the reduced order driveline model which well represents whole driveline model despite limited state orders.

$$\dot{\omega}_o = \frac{1}{J_o} [i_f(i_{t1}T_{c1} + i_{t2}T_{c2}) - T_o], \quad (21)$$

$$\dot{\omega}_\omega = \frac{1}{J_v} (T_o - T_L), \quad (22)$$

$$\dot{T}_o = k_o(\omega_o - \omega_\omega) + b_o(\dot{\omega}_o - \dot{\omega}_\omega), \quad (23)$$

In this structure, ω_o, ω_ω are measurable values in production cars and will used as correction terms in the observer.

Eq. (23) contains derivative form of ω_o, ω_ω . Therefore, eq. (21, 22) could be substitute to eq. (23). After manipulating (21)-(23), following rearranged equations could be obtained.

$$x = [x_1 \ x_2 \ x_3]^T = [\omega_o \ \omega_\omega \ T_o]^T \\ \dot{x}_1 = -\frac{1}{J_o}x_3 + \frac{1}{J_o}(i_{t1}T_{c1} + i_{t2}T_{c2})i_f = f_1(x, t) \\ \dot{x}_2 = \frac{1}{J_v}x_3 - \frac{1}{J_v}T_L = f_2(x, t) \\ \dot{x}_3 = k_o x_1 - k_o x_2 + \frac{b_o}{J_o} \left[\left(\frac{i_{t1}}{i_{t1} - i_{t2}} T_{c1} + \right. \right.$$

$$\begin{aligned}
& \left. \frac{i_{t2}}{i_{t2} - i_{t1}} T_{c2} \right) i_f - x_3 \Big] - \frac{b_o}{J_v} (x_3 - T_L) \\
& = k_o (x_1 - x_2) - \frac{b_o}{J_v} x_3 + \frac{b_o}{J_v} T_L \\
& = f_3(x, t)
\end{aligned} \tag{24}$$

Model has a three state and contains nonlinearities such as vehicle load torque which includes rolling resistances, aerodynamics effect and road inclinations.

4.2 Sliding mode observers

The sliding mode observer, chosen for its robustness to model uncertainties and applicability to nonlinear systems, was designed based on reduced order model eq. (24). Principle of conventional sliding mode observers are well documented in [6]. Based on DCT-based driveline model, linear correction terms and switching terms could be simultaneously applied to estimate output shaft torque. The final form of designed observer can be written down as follows.

$$\begin{aligned}
\dot{\hat{\omega}}_o &= \frac{1}{J_o} [i_f(i_{t1}\hat{T}_{c1} + i_{t2}\hat{T}_{c2}) - \hat{T}_o] + l_1(\omega_o - \hat{\omega}_o) \\
&= \frac{1}{J_o} \left[\frac{i_{t1}i_f}{i_{t1} - i_{t2}} \left(\frac{\hat{T}_o}{i_f} - i_{t2}\hat{T}_c \right) + \frac{i_{t2}i_f}{i_{t2} - i_{t1}} \left(\frac{\hat{T}_o}{i_f} \right. \right. \\
&\quad \left. \left. - i_{t1}\hat{T}_c \right) - \hat{T}_o \right] + l_1(\omega_o - \hat{\omega}_o) \\
&= 0 + l_1(\omega_o - \hat{\omega}_o) \\
\dot{\hat{\omega}}_\omega &= \frac{1}{J_v} (\hat{T}_o - \hat{T}_L) + l_2(\omega_\omega - \hat{\omega}_\omega) \\
\dot{\hat{T}}_o &= k_o(\hat{\omega}_o - \hat{\omega}_\omega) - \frac{b_o}{J_v} (\hat{T}_o - \hat{T}_L) + l_3(\omega_\omega - \hat{\omega}_\omega) \\
&\quad + l_4(\omega_\omega - \hat{\omega}_\omega) + k_1 \text{sgn}(\omega_o - \hat{\omega}_o) \\
&\quad + \underbrace{k_2 \text{sgn}(\omega_\omega - \hat{\omega}_\omega)}_{\text{switching term}}
\end{aligned} \tag{25}$$

From the designed observers, two measurable state serves as a correction terms and switching terms are only applied to output shaft observers.

Switching gains and linear gains are selected as follows [1].

$$\begin{aligned}
\dot{x} &= f(x, t) \\
\dot{\hat{x}} &= \hat{f}(\hat{x}, t) + L\tilde{x} + k \text{sgn}(\tilde{x}) \\
\dot{x} - \dot{\hat{x}} &= f(x, t) - \hat{f}(\hat{x}, t) - L\tilde{x} - k \text{sgn}(\tilde{x}) \\
\dot{\tilde{x}} &= \Delta f - L\tilde{x} - k \text{sgn}(\tilde{x})
\end{aligned}$$

Let Lyapunov function as

$$\begin{aligned}
V &= \frac{1}{2} S^2 \text{ and } S = x - \hat{x} = \tilde{x} \\
\dot{V} &= S\dot{S} = \tilde{x}\dot{\tilde{x}} = \tilde{x}(\Delta f - L\tilde{x} - k \text{sgn}(\tilde{x})) < 0 \\
&= -L\tilde{x}^2 + \tilde{x}(\Delta f - k' \text{sgn}(\tilde{x})) \quad (k' \text{ includes } L)
\end{aligned}$$

So, $|\Delta f| < k'$

Where Δf is model uncertainty and it assumed that approximated values of $\Delta f_{1,2,3}$ are known, switching gains could be determined to make lyapunov function candidate always be in a negative definite.

5. SIMULATION RESULTS

In this section, designed observers in the previous sections were verified via computer simulation using commercial softwares, Matlab & Simulink. In order to describe the actual plant, driveline sample time was set to 0.1ms and observers sample time was set to 1ms which is reasonable values in the industry. Fig. 2, 3 shows that performances of unknown input observers with a proper linear gain tuning. Estimated lumped clutch torque is tracking the actual lumped clutch torque with a time delay because of property of unknown input observers. Estimated lumped clutch torque utilized to calculate respective torque Fig. 4, 5.

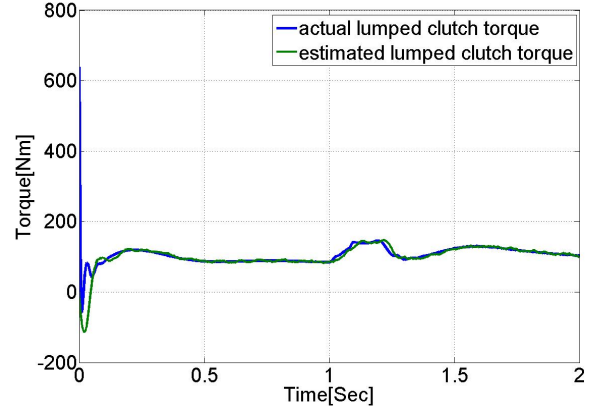


Fig. 2 Lumped clutch torque estimation using the unknown input observers

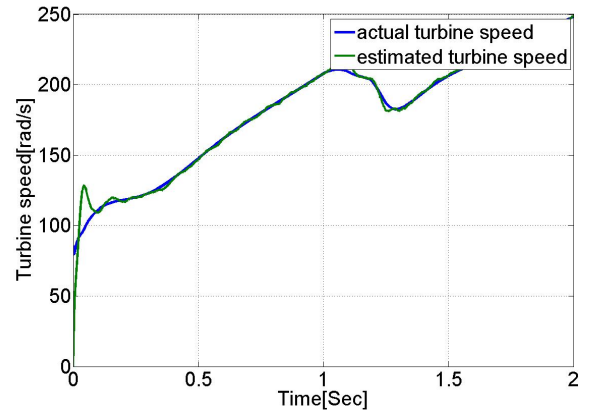


Fig. 3 Turbine speed estimation using the unknown input observers

Fig. 4,5 shows the estimated respective clutch torque (one being disengaging and the other being engag-

ing phase) and it also coupled with output shaft torque estimation performance Fig. 6.

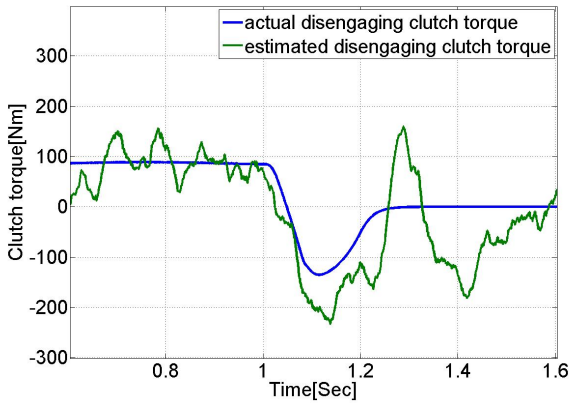


Fig. 4 Disengaging clutch torque estimation using the sliding mode observers and lumped clutch torque

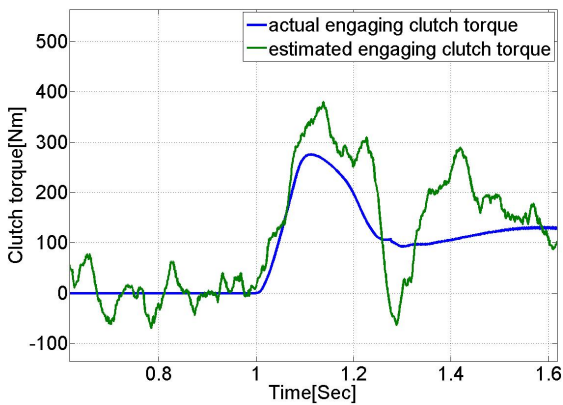


Fig. 5 Engaging clutch torque estimation using the sliding mode observers and lumped clutch torque

Referring again, lumped clutch torque, respective clutch torque, output shaft torque are all coupled system in actual driveline model and affect each other estimation performances simultaneously. Therefore proper gain tuning is crucial to track the actual torque at the same time. In this ways, all the torques which are needed to clutch control could be estimated without torque, pressure, clamping force information. Fig. 7, 8 shows the results of the estimated angular speed about each shaft which are measurable values in production vehicles.

Final estimation results show that estimated torque is tracking actual values but degraded in transient area (gear shift phase) because of its insufficient information. However estimated torques have the similar tendency with actual values with limited available values. Therefore through after-treatment about estimated values, it could be used for reference for clutch control.

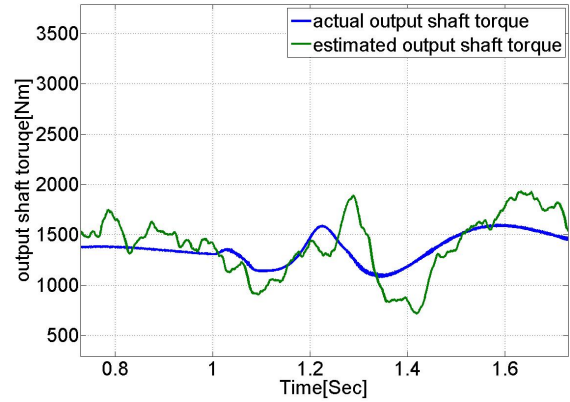


Fig. 6 Output shaft torque estimation using sliding mode observers

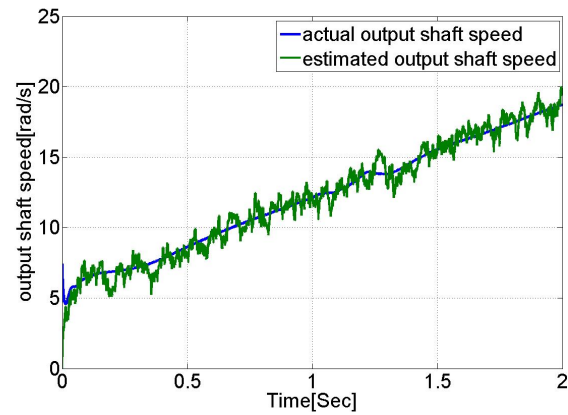


Fig. 7 Output shaft speed estimation using sliding mode observers

6. CONCLUSION

This paper has proposed clutch torque estimation method combining with unknown input observer and sliding mode observer during gear shifting with a simplified DCT-based driveline model. The objective of this work is the use of the estimated clutch torque for optimal gear shifting control. Because the physics or mathematical based actual vehicle model has high-order and nonlinear complexity, this research presented a simplified model to adapt a unknown input observers and sliding mode observers. Performance of designed observers was verified via computer simulation using commercial software, Matlab & Simulink. Although torque or pressure sensors are not applicable to production cars, additional angular velocity sensors could be attached to powertrain model such as clutch speed information. In this ways, the better performance is expected with an additional rotational sensors. The benefits of designed observers could be summarized as follows: improvement of ride quality by reducing vibration or jerk when undergoes gear shifting, improvement of durability of clutches and transmis-

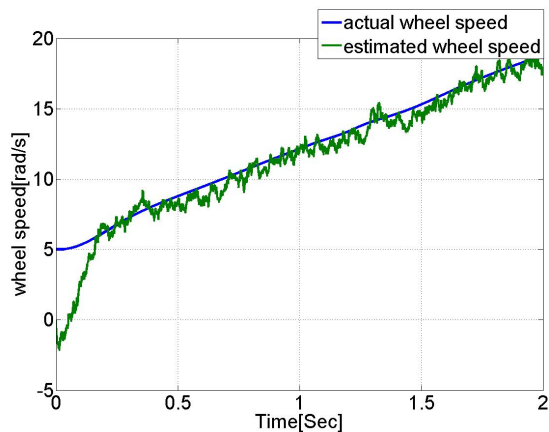


Fig. 8 Wheel speed estimation using sliding mode observers

sions with optimal clutch control.

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