



Driveline modeling and estimation of individual clutch torque during gear shifts for dual clutch transmission

Jiwon J. Oh ^{a,1}, Seibum B. Choi ^{a,*}, Jinsung Kim ^{b,2}

^a KAIST, Daejeon, Republic of Korea

^b Hyundai Motor Company, Hwasung, Republic of Korea

ARTICLE INFO

Article history:

Received 19 August 2013

Accepted 18 April 2014

Available online xxx

Keywords:

Dual clutch transmission

Clutch torque

Output shaft torque

Torque observer

Estimation

Adaptation

ABSTRACT

The purpose of the proposed work centers on the unprecedented task of accurately estimating the torque transmitted through each clutch of the dual clutch transmission. This is to possibly improve the clutch control performance during vehicle launch and gearshifts and to elongate the clutch life. Such goal is attained by only using the measurements and data that are already available in current production vehicles. The suggested estimator requires the speed measurements of engine, input shafts, and wheels, and nominal engine torque information obtained as a function of driver input and engine speed. By synthesizing the estimations obtained by shaft model-based observer, unknown input observers, and adaptive output torque observer, the novel algorithm to estimate the torque of each clutch separately is developed. Stability of the entire combined observer system is analyzed as well. In the process of developing the transfer shaft model-based observer, an original approach for the driveline modeling is proposed. The driveline modeling procedures include compliance model of each transfer shafts with the convenient methods to express the clutch torque without having to use complex clutch friction models, and the developed model is compared with the actual experiment data to validate its accuracy. The effectiveness of the individual clutch torque estimator is demonstrated both through simulations using SimDriveline, and tests on an actual vehicle equipped with a dual clutch transmission.

© 2014 Elsevier Ltd. All rights reserved.

1. Introduction

Surely, the advancement and popularization of the automobile technologies have brought convenience to the modern lives, but at the same time lead to serious environmental and global energy issues. As countermeasures against such problems, various types of automotive technologies such as those related to the development of high efficiency engine, hybridization, or weight reduction have emerged. Among them, one of the research areas that can provide the most prominent improvements regarding the fuel efficiency and thus emission is the transmission technologies.

Significant amount of energy is wasted at the transmission due to slipping nature of the torque converter applied in the widely-used planetary gear-type or continuously variable automatic transmissions. This disadvantage no longer exists when it comes to the manual transmission, in which the engine and the

wheels can be mechanically linked by the clutch. However, manual transmissions require the drivers to skillfully operate the clutch pedal and switch the gear selector, and most drivers find such tasks troublesome.

Such tradeoffs that exist in the automatic and manual transmissions led to the automated manual transmission (AMT). The AMT enables physical engagement of the engine and wheels through the use of clutch that is automatically controlled using electromagnetic or hydraulic actuators [1]. However, awkward shift shocks and discontinuous torque drops were shown to be critical hindrances for the AMTs to gain popularity in the market.

Dual clutch transmissions (DCT) resolve such known issues of other types of transmissions, and have emerged as a promising vehicular technology along with other state-of-the-art transmission technologies [2–4]. DCTs basically operate similarly to the AMTs, but they have two sets of clutch and transfer shafts, often packaged coaxially to minimize volume. By using a set of clutch and transfer shaft for odd-number gears and another set for even-number gears, the target gear can be preselected prior to the actual torque hand-over during a gear shift. This way, by alternatively engaging the two clutches for gear shifts using actuators, seamless torque transmission as well as all of the advantages

* Corresponding author. Tel.: +82 42 350 4120; fax: +82 042 350 3210.

E-mail addresses: jwo@kaist.ac.kr (J.J. Oh), sbchoi@kaist.ac.kr (S.B. Choi), jinsung.kim@hyundai.com (J. Kim).

¹ Tel.: +82 42 350 4104; fax: +82 042 350 3210.

² Tel.: +82 10 5446 9584.

Nomenclature

ω_e	engine speed	i_{t1}	transmission (T/M) gear ratio for transfer shaft 1
ω_d	external damper speed	i_{t2}	transmission (T/M) gear ratio for transfer shaft 2
ω_{c1}	clutch 1 speed (input shaft 1 speed)	i_{f1}	final reduction gear ratio for transfer shaft 1
ω_{c2}	clutch 2 speed (input shaft 2 speed)	i_{f2}	final reduction gear ratio for transfer shaft 2
ω_o	output shaft speed	J_e	engine inertia
ω_w	wheel speed	J_d	external damper inertia
T_e	net engine torque	J_{e1}	effective T/M inertia from clutch 1 perspective
$T_{e,n}$	nominal net engine torque	J_{e2}	effective T/M inertia from clutch 2 perspective
T_d	external damper torque	J_o	output shaft inertia
T_{c1}	clutch 1 torque	J_v	vehicle inertia
T_{c2}	clutch 2 torque	α_{th}	throttle angle
T_{t1}	transfer shaft 1 torque	θ	shaft angle
T_{t2}	transfer shaft 2 torque	k	torsional stiffness
T_o	output shaft torque	b	torsional damping coefficient
T_v	vehicle resistance torque		

of the AMTs are attainable, and ideally provide fast and smooth response, high transmission efficiency, and convenience. Further information on the DCT can be found in [5].

In order to properly manage the clutch actuation for launching and gear shifts, application of the feedback control tactics is crucial, in addition to the conventional feed-forward control tactics which is solely based on the experimentally obtained relationship between the actuation input and the clutch torque output. Such feedback control tactics can be realized with the knowledge of the clutch torque, and it can minimize the shifting time, jerk, and clutch damage. Knowledge of the individual clutch torque may provide the optimum timing for the torque hand-over to reduce the torque interruption during the gear shift and hence reduce the shifting time and jerk. Also, avoiding the clutch tie-up based on the knowledge of each clutch torque during gear shift can prevent unnecessary clutch wear, and moreover, component failure.

One may argue that the clutch torque information can be replaced by the hydraulic pressure of the actuator in case of the wet clutch actuated by hydraulic pressure, or by the actuator position information in case of the dry clutch actuated by electromagnetic actuators. However, this research focuses on the application to the dry type DCTs actuated by electromagnetic actuators, with the presence of the diaphragm spring of high stiffness. This indicates that the actuator stroke is minimized to increase the actuation efficiency. In this case, no clutch pressure information is available like in case of the hydraulic actuation, and moreover, knowing the actuator position is not helpful for effective clutch actuation, since attempting to estimate the clutch normal force from the short actuator stroke can be extremely difficult.

Another way to attain the clutch torque information is to use the torque measuring sensors. However, because of the high cost of the torque measurement devices, the cars equipped with DCTs may lose their economic competitiveness in the market, especially when two sensors of sufficiently high accuracy are used for two transfer shafts of the DCTs to separately measure the torque transmitted through each individual clutch.

In the hope to replace the torque sensors by torque estimating algorithms, previous efforts have attempted to develop estimators for clutch torque of AMTs or combined output shaft torque of DCTs, but no work so far has attempted the separated torque estimation of each individual clutch of the DCTs. Turbine torque estimation methods are introduced in [6–8] as torque estimator for automatic transmissions with torque converters, but are not applicable to DCTs that involve different structures and mechanisms. For the torque estimation of AMTs, Kalman filter-based observers are

designed [9–12], whose estimation performance is limited by the linearized models. The Luenberger observer-based estimation schemes [13,14] overlook the effects of the nominal engine torque and vehicle inertia uncertainties. The sliding mode observer in [15] and unknown input observers proposed in [16,17] involve chattering and phase lag issues. Also, the reduced order nonlinear torque observer for AMT proposed in [18] overlooks the importance of experimental validation. In addition, the torque estimator for DCTs [19–23] exists, but since they only estimate the driveline torque for launch control that only involves a single clutch, the applicability of such estimator is technically the same as that of the aforementioned estimators for AMTs. Torque curve adaptation for DCT actuators using sliding mode observer compensation is proposed in [24], and actuator control tactics are presented in [25,26], but their methods are majorly position-based, which do not provide high estimation accuracy in all clutch phases.

In addition, numerous previous works also have proposed driveline modeling methods. However, their accuracy compared to the driveline of an actual vehicle is limited because the torsional compliance in the shafts is overly simplified [27–29] or neglected [30]. Recent work has investigated the high order (15 DOF) dynamics of the driveline to develop a DCT model [31], but using a single method to compute the engaged clutch torque provides limited flexibility for application on the observer design which may follow, and in most works, the experimental validation is not conducted [27,28]. Also, they often require complex friction model to express the clutch dynamics that involve numerous model parameters.

Hence this study proposes the unprecedented estimator for the transmission torque of each individual clutch aimed for application on DCT with electromagnetic clutch actuators with stiff diaphragm spring, as well as a novel DCT driveline modeling method which provides simple and effective ways to express the clutch torque without using complex friction models. The paper is organized as follows. Section 2.1 first briefly introduces the conventional driveline model, and proposes the novel driveline modeling method which incorporates separated shaft compliance model for each shaft in Section 2.2. It is then simulated and also tested against experimental data in Section 2.3. Section 3 proposes the torque estimator by introducing the transfer shaft model-based observer, adaptive output shaft torque observer, and unknown input observers in order. The stability of the integrated estimator is verified. Then in Section 4, the results of the simulation and experiment performed to validate the torque estimation performance are presented.

2. Driveline model

2.1. Conventional driveline model

The conventional driveline model used to describe the flow of the torque from the engine to the wheels through the dual clutch transmission can be basically shown as follows.

Using the torque balance relationships, the above driveline model can be represented as shown next.

$$J'_e \dot{\omega}_e = T_e - T_d \quad (1)$$

The torque balance relationship of the engine and the torsional damper is shown in (1). As done here, modeling of the torsional damper can be optionally included, or otherwise the engine torque can directly express the damper output torque.

$$J_d \dot{\omega}_d = T_d - T_{c1} - T_{c2} \quad (2)$$

Now, treating the rotating mass between the clutch and the output shaft as a single body of inertia, the following lumped dynamics of the transfer shaft can be reached.

$$J_{e1} \dot{\omega}_{c1} = T_{c1} + T_{c2} \frac{i_{t2}}{i_{t1}} - \frac{T_o}{i_{t1}} \quad (3)$$

This dynamics is obtained under the assumption that the lumped body of the transfer shaft only experiences a single DOF rotational motion, and involves two torque inputs from clutch 1 and 2 and a single torque output to the output shaft. The clutch 1 dynamics shown in (3) can also be expressed from the perspective of clutch 2 as shown in (4), and one may realize that (4) is the exact scalar multiple of the dynamics shown in (3).

$$J_{e2} \dot{\omega}_{c2} = T_{c1} \frac{i_{t1}}{i_{t2}} + T_{c2} - \frac{T_o}{i_{t2}} \quad (4)$$

$$J_v \dot{\omega}_w = T_o i_{fi} - T_v \quad (5)$$

Here, the equivalent transfer shaft inertia computed from the clutch 2 perspective, or J_{e2} , shall be attached to the transfer shaft next to clutch 2, if it is to be drawn in Fig. 1. Also, for the type of DCTs that involve two separate final reduction gears, since the final reduction gear ratio may vary depending on which clutch is engaged, it is denoted as i_{fi} . For each dynamics, the related torque is modeled as follows.

$$T_e = f(\alpha_{th}, \omega_e) \quad (6)$$

$$T_d = k_d(\theta_e - \theta_d) + b_d(\omega_e - \omega_d) \quad (7)$$

$$T_{c1} = F_{n1} C_{c1} \mu \text{sgn}(\omega_d - \omega_{c1}) \quad (8)$$

$$T_{c2} = F_{n2} C_{c2} \mu \text{sgn}(\omega_d - \omega_{c2}) \quad (9)$$

$$T_o = k_o \left(\frac{\theta_{c1}}{i_{t1}} - i_{f1} \theta_w \right) + b_o \left(\frac{\omega_{c1}}{i_{t1}} - i_{f1} \omega_w \right) \text{ or } k_o \left(\frac{\theta_{c2}}{i_{t2}} - i_{f2} \theta_w \right) + b_o \left(\frac{\omega_{c2}}{i_{t2}} - i_{f2} \omega_w \right) \quad (10)$$

$$T_v = r_w \left(\underbrace{m_v g \sin(\theta_{road})}_{\text{road gradient}} + \underbrace{K_{rr} m_v g \cos(\theta_{road})}_{\text{rolling resistance}} + \underbrace{\frac{1}{2} \rho v_x^2 C_d A}_{\text{aerodynamic drag}} \right) \quad (11)$$

Here, F_n , C_c , μ , r_w , θ_{road} , K_{rr} , m_v , ρ , v_x , C_d , and A each indicate the clutch normal force, clutch normal force coefficient, kinetic friction coefficient, wheel radius, road gradient angle, tire rolling resistance, vehicle mass, air density, vehicle velocity, aerodynamic drag coefficient, and vehicle frontal area, respectively. The engine torque is determined based on the empirically obtained relationship between the engine load conditions and produced net torque.

With the conventional DCT driveline model constructed this way, the primary vibration mode of the shafts can be expressed. However, the given clutch torque expressions may not accurately resemble the stick-slip phenomenon of the actual clutch, since they assume that the clutch plates are constantly rubbing against each other with the given friction coefficient.

Also, the most critical limitation of the conventional driveline model is that the two transfer shafts are modeled as a lumped inertia. This means that the dynamics of the first clutch (henceforth referred to as the clutch 1) is exactly identical to that of the clutch 2 when multiplied by the appropriate gear ratio. This is obviously not true for the case of an actual driveline. Since all rotational shafts involve compliance to the torsional efforts, clutch 1 speed may differ significantly from clutch 2 speed multiplied by the gear ratio, especially when high amount of torque is applied to the transfer shafts.

For an instance, the backward torque recirculation phenomenon within the transmission cannot be expressed at all with the conventional model due to this shortcoming. The backward torque recirculation refers to the cases when the torque transmitted from one clutch is transmitted back to the other clutch instead of being transmitted forward onto the output shaft. This may happen when both clutches are going through the stick-slip, which commonly happens in DCTs for continuous torque transmission. If both clutches are slipping, the states obtained from the conventional model may be close enough to the actual states. However, when one clutch is slipping and the other is tightly engaged, identification of the torque transmitted through the engaged side is not possible with the simplified model.

2.2. DCT driveline model with transfer shaft compliance

In order to deal with the limitations of the conventional model, the DCT driveline model with transfer shaft compliance is suggested, whose structure is shown in Fig. 2.

In the proposed model, the following transfer shaft dynamics are included.

$$J_{e1} \dot{\omega}_{c1} = T_{c1} - \frac{T_{t1}}{i_{t1}} \quad (12)$$

$$J_{e2} \dot{\omega}_{c2} = T_{c2} - \frac{T_{t2}}{i_{t2}} \quad (13)$$

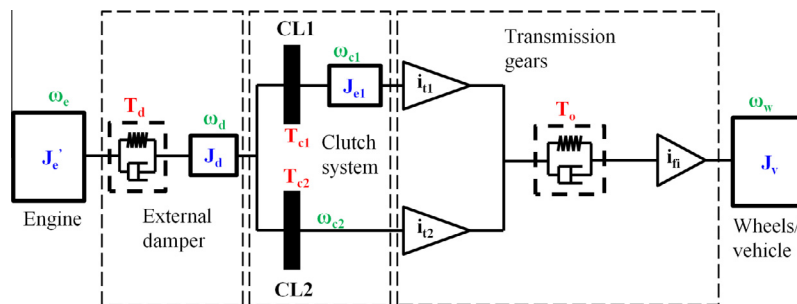


Fig. 1. Illustration of the conventional driveline model of the dual clutch transmission system (J : inertia, T : torque, ω : angular velocity).

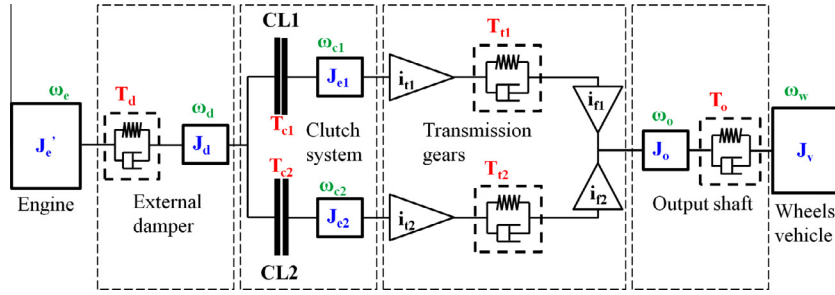


Fig. 2. Illustration of the driveline model of the dual clutch transmission system that involves transfer shaft compliance (J : inertia, T : torque, ω : angular velocity).

With the above transfer shaft dynamics, the output shaft dynamics is altered and is expressed as follows.

$$J_o \dot{\omega}_o = i_{f1} T_{t1} + i_{f2} T_{t2} - T_o \quad (14)$$

Since the transfer shaft dynamics is modeled separately, the torque transmitted at each transfer shaft can be modeled as shown next.

$$T_{t1} = k_{t1} \left(\frac{\theta_{c1}}{i_{t1}} - i_{f1} \theta_o \right) + b_{t1} \left(\frac{\omega_{c1}}{i_{t1}} - i_{f1} \omega_o \right) \quad (15)$$

$$T_{t2} = k_{t2} \left(\frac{\theta_{c2}}{i_{t2}} - i_{f2} \theta_o \right) + b_{t2} \left(\frac{\omega_{c2}}{i_{t2}} - i_{f2} \omega_o \right) \quad (16)$$

Similar to how the transfer shaft torque is obtained by taking the torsional compliance in each shaft into consideration, the output shaft torque can be obtained by accounting for the compliance of the output shaft connected to the wheels.

$$T_o = k_o (\theta_o - \theta_w) + b_o (\omega_o - \omega_w) \quad (17)$$

Now, the clutch torque is calculated using three different methods. The first method is based on the external damper dynamics, where each clutch torque is obtained by altering – Eq. (2).

$$T_{c1} = \begin{cases} 0, & \text{when disengaged} \\ \mu R_{c1} F_{n1} \text{sgn}(\omega_d - \omega_{c1}), & \text{when slipping} \\ T_d - T_{c2} - J_d \dot{\omega}_d, & \text{when engaged} \end{cases} \quad (18)$$

$$T_{c2} = \begin{cases} 0, & \text{when disengaged} \\ \mu R_{c2} F_{n2} \text{sgn}(\omega_d - \omega_{c2}), & \text{when slipping} \\ T_d - T_{c1} - J_d \dot{\omega}_d, & \text{when engaged} \end{cases} \quad (19)$$

Here, the clutch is considered disengaged when the normal force is zero. The threshold between the engaged and slipping phases is designed using the clutch slip and the clutch actuator position, and detailed procedures to decide the clutch phase are shown in Fig. 3.

Another method to calculate the clutch torque is based on the transfer shaft dynamics. Here, each clutch torque is obtained by altering Eqs. (12) and (13).

$$T_{c1} = \frac{T_{t1}}{i_{t1}} + J_{e1} \dot{\omega}_{c1}, \quad \text{when engaged} \quad (20)$$

$$T_{c2} = \frac{T_{t2}}{i_{t2}} + J_{e2} \dot{\omega}_{c2}, \quad \text{when engaged} \quad (21)$$

Similar to the first method, the clutch state is separated into three different phases, and the clutch torque calculated based on the transfer shaft dynamics as shown in (20) and (21) may substitute the third lines in (18) and (19).

The last method to calculate the clutch torque is based on the integrated model, in which the clutch torque is directly calculated by taking all of the damper dynamics and two transfer shaft dynamics into consideration. For this, from (12) and (13), the following transfer shaft torque can be reached.

$$T_{t1} = T_{c1} i_{t1} - J_{e1} \dot{\omega}_{c1} i_{t1} \quad (22)$$

$$T_{t2} = T_{c2} i_{t2} - J_{e2} \dot{\omega}_{c2} i_{t2} \quad (23)$$

Substituting the above expressions of the transfer shaft torque into (14) gives the following.

$$J_o \dot{\omega}_o = T_{c1} i_{t1} i_{f1} - J_{e1} \dot{\omega}_{c1} i_{t1} i_{f1} + T_{c2} i_{t2} i_{f2} - J_{e2} \dot{\omega}_{c2} i_{t2} i_{f2} - T_o \quad (24)$$

Now solving (24) for the clutch torque leads to the following expressions.

$$T_{c1} = \frac{J_o}{i_{t1} i_{f1}} \dot{\omega}_o + J_{e1} \dot{\omega}_{c1} - T_{c2} \frac{i_{t2}}{i_{t1}} + J_{e2} \dot{\omega}_{c2} \frac{i_{t2}}{i_{t1}} + \frac{T_o}{i_{t1} i_{f1}} \quad (25)$$

$$T_{c2} = \frac{J_o}{i_{t2} i_{f2}} \dot{\omega}_o + J_{e1} \dot{\omega}_{c1} \frac{i_{t1}}{i_{t2}} - T_{c1} \frac{i_{t1}}{i_{t2}} + J_{e2} \dot{\omega}_{c2} + \frac{T_o}{i_{t2} i_{f2}} \quad (26)$$

Equating the above expressions with the third lines in (18) and (19), and rearranging the equations give the following expressions of the clutch torque of each side.

$$T_{c2} \left(\frac{i_{t2}}{i_{t1}} - 1 \right) = J_d \dot{\omega}_d - T_d + \frac{J_o}{i_{t1} i_{f1}} \dot{\omega}_o + J_{e1} \dot{\omega}_{c1} + J_{e2} \dot{\omega}_{c2} \frac{i_{t2}}{i_{t1}} + \frac{T_o}{i_{t1} i_{f1}} \quad (27)$$

$$T_{c1} \left(\frac{i_{t1}}{i_{t2}} - 1 \right) = J_d \dot{\omega}_d - T_d + \frac{J_o}{i_{t2} i_{f2}} \dot{\omega}_o + J_{e1} \dot{\omega}_{c1} \frac{i_{t1}}{i_{t2}} + J_{e2} \dot{\omega}_{c2} + \frac{T_o}{i_{t2} i_{f2}} \quad (28)$$

Having three different methods to express the clutch torque provide flexibility in the construction of the model since the clutch torques are available as a function of different variables. Furthermore, this indicates that, in the construction of the related estimation algorithms that make use of the designed model, since not all required parameters or measurements are available in real application, clutch torque observer can be designed using different sets of variables depending on which of the variables are practically available for production car. For instance, the first method requires the information of damper torque and the torque of the other clutch, which can be useful for the case when only one clutch is slipping. The second method requires the transfer shaft torque only, which can be an advantageous situation if torque sensors are attached on the transfer shafts for validation purpose. Lastly, the third method does not require the clutch torque but instead requires the output shaft torque, which then can be advantageous for validation when output shaft torque sensor is available or for estimation if output shaft torque is observable.

2.3. Validation of developed model via experiment

In order to directly show the effectiveness of the proposed model, it must be compared experimentally with the actual transmission. The experiment is conducted under the supervision Valeo Pyeong-Hwa, by which the test vehicle – Hyundai Avante MD (Elantra) – with its original transmission replaced by DCT, the corresponding TCU commands, and actuator controller are arranged. The measured parameters of the transmission used for experiment are given in the following.

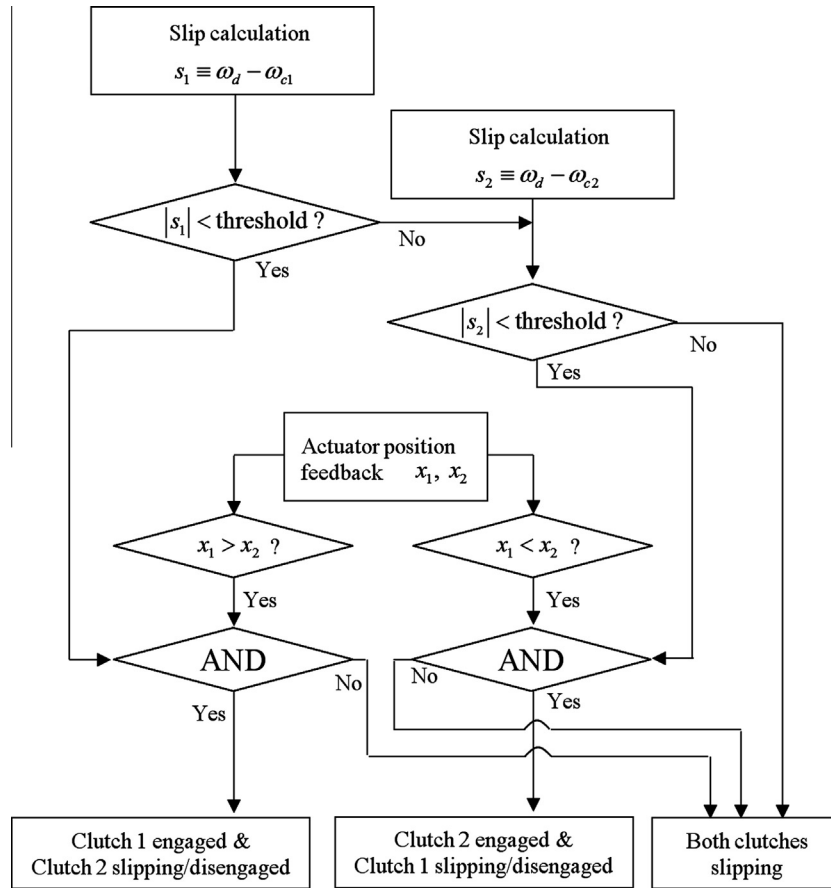


Fig. 3. Clutch phase decision procedure based on clutch slip and actuator position.

Engine inertia	$J_e = 0.2$	Final reduction gear 1 ratio	$i_{f1} = 4.657$
Torsional damper inertia	$J_d = 0.086$	Final reduction gear 2 ratio	$i_{f2} = 3.601$
Clutch 1 inertia	$J_{c1} = 0.043$	Effective clutch 1 radius	$R_{c1} = 0.21$
Clutch 2 inertia	$J_{c2} = 0.047$	Effective clutch 2 radius	$R_{c2} = 0.1684$
Gear 1 inertia	$J_{g1} = 0.00602$	Clutch 1 friction coefficient	$\mu_{c1,kinetic} = 0.27$
Gear 2 inertia	$J_{g2} = 0.005415$	Clutch 2 friction coefficient	$\mu_{c2,kinetic} = 0.27$
Gear 3 inertia	$J_{g3} = 0.006926$	Wheel radius	$r_w = 0.338$
Gear 4 inertia	$J_{g4} = 0.00737$	Wheel inertia	$J_w = 1.7747$
Gear 5 inertia	$J_{g5} = 0.008503$	Vehicle mass	$m_v = 1596$
Gear 6 inertia	$J_{g6} = 0.008973$	Tire rolling resistance coefficient	$K_{rr} = 0.015$
Gear 7 inertia	$J_{g7} = 0.01102$	Aerodynamic drag coefficient	$C_d = 0.325$
Reverse gear inertia	$J_{gR} = 0.006432$	Effective frontal area	$A_f = 2.2126$
Output shaft inertia	$J_o = 0.04$		
Torsional damper constant	$b_d = 10$		
Transfer shaft 1 spring constant	$k_{t1} = 314,200$		
Transfer shaft 1 damping constant	$b_{t1} = 53$		
Transfer shaft 2 spring constant	$k_{t2} = 301,200$		
Transfer shaft 2 damping constant	$b_{t2} = 51$		
Output shaft spring constant	$k_o = 9520$		
Output shaft damping constant	$b_o = 591$		
Gear 1 ratio	$i_{t1} = 3.688$		
Gear 2 ratio	$i_{t2} = 2.850$		
Gear 3 ratio	$i_{t3} = 1.387$		
Gear 4 ratio	$i_{t4} = 0.951$		
Gear 5 ratio	$i_{t5} = 0.943$		
Gear 6 ratio	$i_{t6} = 0.780$		
Gear 7 ratio	$i_{t7} = 0.674$		
Reverse gear ratio	$i_{tR} = 3.537$		

[Units are SI derived (kg, m, s, A)]

These parameters are identified and provided by Valeo Pyeong-Hwa, TCU target clutch position is provided by Hyundai Motor Company, and TCU clutch actuator tracking controller is provided by Continental AG. The logic is integrated and mounted on vehicle by Hyundai Motor Company. No additional sensor other than those required to operate the given DCT-mounted vehicle is installed, and a torque measurement device is used for the output shaft.

The driveline speeds obtained by the model via simulation are compared with the measured driveline speeds of the actual vehicle mounted with DCT in Fig. 4. Here, only the system inputs – including the throttle input, TCU target actuator position, and nominal engine torque information – required in the model is replaced by those acquired by the vehicle CAN, and no sensor measurements are provided for the model. Namely, a fully open-loop comparison was conducted.

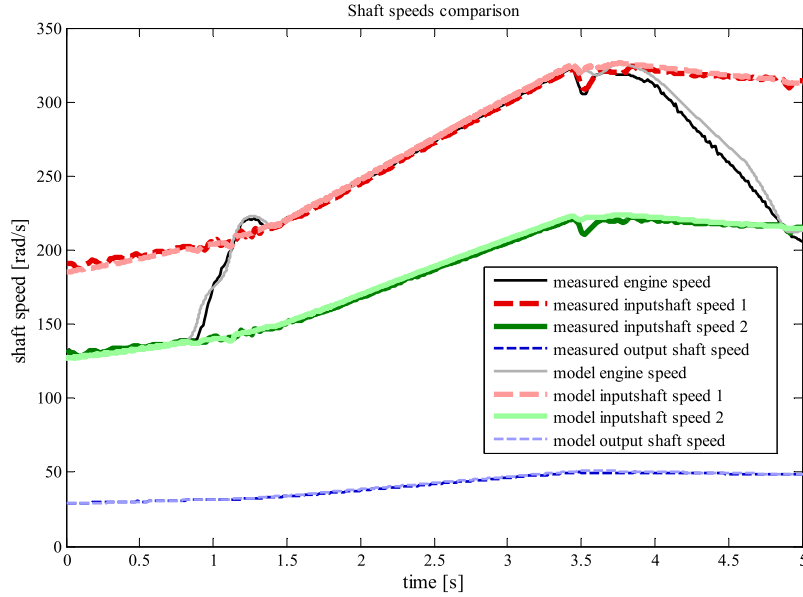


Fig. 4. Comparison of the open-loop simulation results with those obtained by measuring the actual states using the actual vehicle.

The test involves a downshift and an upshift of gear. During the gear shifts and engaged phases, the driveline speeds obtained by simulation effectively follow those that are actually measured by the sensors of the actual vehicle. Considering the fact that no feedback was given for the validation process, each driveline speed obtained by the simulation is highly agreeable to the corresponding driveline speed measured in the test vehicle. Furthermore, being able to express the engine speed overshoot at approximately 1.2 s with the magnitude and interval that are agreeable to the actually measured values indicates that the shaft inertia model indeed reflects that of the actual transmission. Also, the converged driveline speeds and the engine acceleration similar to that of the actual transmission during gear shifts depicts effective representation of the clutch transmitted torque.

3. Clutch torque estimator

Making use of the proposed model which involves the individual transfer shaft compliances, an observer that comprises of multiple sub-observers to estimate each clutch torque of DCTs is proposed. Each sub-observer is designated for the estimation of specific targets, whose results interact to provide the final estimation of clutch 1 torque and clutch 2 torque. To facilitate the understanding, Fig. 5 shows the schematics of the proposed clutch torque observer system. Given in the following is the overview of the major components of the estimator.

The primary role of the adaptive output shaft torque observer is to obtain the output torque. Utilizing this result, the two unknown input observers estimate the interconnecting states, $\hat{T}_c$ and $\hat{T}_a$, based on the nominal engine torque $T_{e,n}$. Estimations of each clutch torque based on these results have high reliability during the steady state but involve phase lag error during the transient state. On the other hand, the individual transfer shaft model-based observer estimations swiftly respond during the transient states, while they involve drift issue during the steady state. Hence, the individual transfer shaft model-based observer takes the form of a Luenberger-like observer, in which the estimations obtained by the unknown input observers prevent the final estimation results $\hat{T}_{c1}$ and $\hat{T}_{c2}$ from drifting during the steady state.

Although the principle applied behind the individual transfer shaft model-based observer or unknown input observers is capable

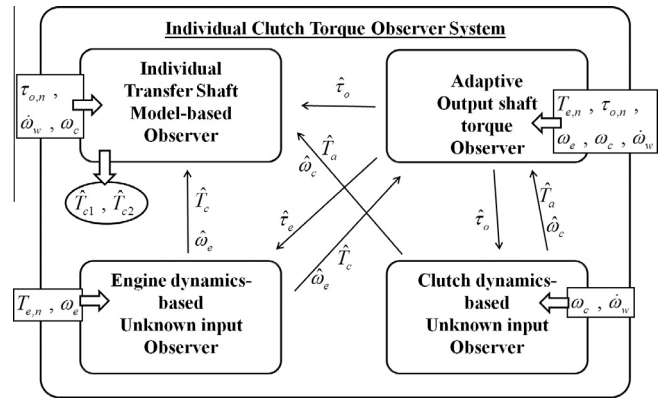


Fig. 5. Schematics of the individual clutch torque observer system.

of estimating each clutch torque separately without having to merge multiple observers, these multiple parts are deliberately designed to operate together to give higher estimation accuracy both during the steady and transient states.

The options to partially use the above-mentioned observer system are also available to reduce the computational load when applied on a real vehicle, but the experiment results – to be displayed in the later part – show that the amount of computation required in the current system is sufficiently low for the real-time application of the suggested observer.

3.1. Individual transfer shaft model-based observer

In the individual transfer shaft model-base observer design procedures, the two transfer shafts of DCTs are separately modeled using the proposed modeling schemes introduced in the previous section, so that the clutch torque of each side can be separately identified.

Based on the clutch dynamics modeled in (12) and (13), we have the following approximation of the clutch torques.

$$i_{t1}T_{c1} \approx T_{t1} \quad (29)$$

$$i_{t2}T_{c2} \approx T_{t2} \quad (30)$$

Also, the following equality between the sum of transfer shaft torques and the output shaft torque can be deduced.

$$T_o \approx i_{f1}T_{c1} + i_{f2}T_{c2} \quad (31)$$

Here, the equivalent transmission inertia and output shaft inertia are assumed to be sufficiently small. Such assumption is suggested for the noise rejection purpose during the estimation schemes, and it still preserves the model accuracy since the transmission and output shaft inertia can be expressed as a part of lumped vehicle inertia and high driveline torsional stiffness does not allow significant amount of relative angular deflection among their internal parts.

Now, from (15) and (16), we have the following expressions for the clutch torque rates, when the effect of transfer shaft damping is small.

$$\dot{T}_{c1} = \frac{k_{t1}}{i_{t1}} \left(\frac{\omega_{c1}}{i_{t1}} - i_{f1}\omega_o \right) \quad (32)$$

$$\dot{T}_{c2} = \frac{k_{t2}}{i_{t2}} \left(\frac{\omega_{c2}}{i_{t2}} - i_{f2}\omega_o \right) \quad (33)$$

To attempt estimation of clutch torques by using (32) and (33), the output shaft speed information is required. To estimate the output shaft speed, (29)–(31) are merged to reach the following.

$$\dot{T}_o = k_{t1}i_{f1} \left(\frac{\omega_{c1}}{i_{t1}} - i_{f1}\omega_o \right) + k_{t2}i_{f2} \left(\frac{\omega_{c2}}{i_{t2}} - i_{f2}\omega_o \right) \quad (34)$$

This equation is altered to isolate the output shaft speed as shown.

$$\omega_o = \frac{k_{t1}i_{f1} \frac{\omega_{c1}}{i_{t1}} + k_{t2}i_{f2} \frac{\omega_{c2}}{i_{t2}} - \dot{T}_o}{k_{t1}i_{f1}^2 + k_{t2}i_{f2}^2} \quad (35)$$

Based on this, the output shaft speed estimation is obtained as follows.

$$\hat{\omega}_o = \frac{k_{t1}i_{f1} \frac{\omega_{c1,m}}{i_{t1}} + k_{t2}i_{f2} \frac{\omega_{c2,m}}{i_{t2}} - \hat{T}_o}{k_{t1}i_{f1}^2 + k_{t2}i_{f2}^2} \quad (36)$$

Where the output shaft torque rate is replaced by the derivative of the output shaft torque estimation obtained by the adaptive output shaft torque observer, which is to be introduced later.

Now, by using the estimated output shaft speed, the following individual clutch torque observer is designed.

$$\begin{aligned} \hat{\dot{T}}_{c1} = & \frac{k_{t1}}{i_{t1}} \left(\frac{\omega_{c1,m}}{i_{t1}} - i_{f1}\hat{\omega}_o \right) + L_1(\hat{T}_{c1,UIO} - \hat{T}_{c1}) \\ & - \frac{\gamma_l k_{t1} i_{f1} \dot{\omega}_w}{i_{t1}(k_{t1}i_{f1}^2 + k_{t2}i_{f2}^2)} (\tau_{o,n} - \hat{\tau}_o) \end{aligned} \quad (37)$$

$$\begin{aligned} \hat{\dot{T}}_{c2} = & \frac{k_{t2}}{i_{t2}} \left(\frac{\omega_{c2,m}}{i_{t2}} - i_{f2}\hat{\omega}_o \right) + L_2(\hat{T}_{c2,UIO} - \hat{T}_{c2}) \\ & - \frac{\gamma_l k_{t2} i_{f2} \dot{\omega}_w}{i_{t2}(k_{t1}i_{f1}^2 + k_{t2}i_{f2}^2)} (\tau_{o,n} - \hat{\tau}_o) \end{aligned} \quad (38)$$

where L_1 and L_2 are feedback gains and $\hat{T}_{c1,UIO}$, $\hat{T}_{c2,UIO}$, $\tau_{o,n}$ and $\hat{\tau}_o$ are the clutch torque estimations obtained by the unknown input observers, and normalized nominal and estimated output shaft torque that are dealt in detail in the later sections that introduce the unknown input observers and adaptive output shaft torque observer. γ_l is the adaptive gain which is to be introduced with the adaptive observer as well.

The individual transfer shaft model-based observer presented in this section takes a Luenberger-like form, and the stability of its error dynamics is obvious. The stability of the entire observer system error is analyzed in the later parts.

3.2. Adaptive output shaft torque observer

The adaptive output shaft torque observer has been designed in [32] for the case of automated manual transmission to estimate the output shaft torque only with the knowledge of the engine speed, clutch speed, and wheel speed measurements. However, such work must be majorly altered to be applicable to the case of DCTs that involve two separate routes by which the torque may flow.

By altering (2) and (3) to include the final reduction gear ratio, the following equations are reached.

$$J_e \dot{\omega}_e = T_e - T_{c1} - T_{c2} \quad (39)$$

$$J_{e1} \dot{\omega}_{c1} = T_{c1} + \frac{i_{t2}i_{f2}}{i_{t1}i_{f1}} T_{c2} - \frac{T_o}{i_{t1}i_{f1}} \quad (40)$$

Here, the dynamics of the torsional damper is merged into the engine dynamics, and the engine and damper inertia are expressed as a lumped parameter J_e . When the clutch 1 torque is isolated, we have

$$T_{c1} = T_e - T_{c2} - J_e \dot{\omega}_e \quad (41)$$

$$T_{c1} = J_{e1} \dot{\omega}_{c1} - \frac{i_{t2}i_{f2}}{i_{t1}i_{f1}} T_{c2} + \frac{T_o}{i_{t1}i_{f1}} \quad (42)$$

and equating the obtained clutch 1 torque leads to the following relationship.

$$T_e - T_{c2} - J_e \dot{\omega}_e = J_{e1} \dot{\omega}_{c1} - \frac{i_{t2}i_{f2}}{i_{t1}i_{f1}} T_{c2} + \frac{T_o}{i_{t1}i_{f1}} \quad (43)$$

By replacing the terms on the right hand side with the variables defined in the later sections on the unknown input observers, the following is reached.

$$J_{e1} \dot{\omega}_{c1} + J_e \dot{\omega}_e = T_e + T_{a1} - T_c - \frac{T_o}{i_{t1}i_{f1}} \quad (44)$$

Here, T_c and T_{a1} represent the combined clutch torque and total transmitted clutch torque from clutch 1 perspective, respectively. Now the above relationship is separated into known and unknown terms. The identifiable terms are collected on the left hand side, and the unknown terms on the right hand side.

$$J_{e1} \dot{\omega}_{c1} + J_e \dot{\omega}_e - T_{a1} + T_c = T_e - \frac{T_o}{i_{t1}i_{f1}} \quad (45)$$

The term T_c is considered known since it can be estimated using the unknown input observer or engine/damper dynamics. The term T_{a1} is considered as a known term as well, since it can be estimated using the unknown input observer based on the clutch dynamics. The engine torque and output shaft torque are considered unknown, since the nominal engine torque information often involves transient state error and the output shaft torque is the estimation target. To lessen the burden on adaptation, the engine torque is separated into nominal and unknown parts, and the output shaft torque is made as a function of undriven wheel acceleration.

$$T_e = T_{e,n} + \omega_e \tau_e \quad (46)$$

$$T_o = \dot{\omega}_w \tau_o \quad (47)$$

As shown in (46) and (47), the unknown part of the engine torque T_e is expressed in terms of the engine speed ω_e and the normalized engine torque τ_e . Similarly, the output shaft torque T_o is expressed as a product of the wheel acceleration $\dot{\omega}_w$ and normalized output shaft torque τ_o .

Another reason behind such maneuver is to satisfy the convergence criterion for the update laws to be designed. For a plant with parameter θ which is to be estimated by the update law $\hat{\theta} = \Gamma \varphi \varepsilon$ with order n , where $\Gamma = \Gamma^T > 0$, $\varepsilon = z - \hat{z}$, and $\phi = H(s)u$ with

$H(j\omega_1), \dots, H(j\omega_n)$ linearly independent on $\mathbb{C}^n$ for all $\omega_1, \omega_2, \dots, \omega_n \in \mathbb{R}$, the convergence criterion is met with PE (persistence of excitation) condition [33], if and only if u is sufficiently rich of order n . Here, a stationary signal u is sufficiently rich of order n , if the support of the spectral measures of u contains at least n points. More specifically, a signal u is sufficiently rich of order n , if it consists of at least $n/2$ distinct frequencies. Thus, by analogously taking the estimation target $\hat{\tau}_o$ as θ , which is to be estimated by the update law that takes the form of the described $\hat{\theta} = \Gamma \phi \varepsilon$, $\phi = H(s)u$ or the coefficient for the estimation error in the update law cannot be a constant. Thus, to avoid making ϕ a constant, the estimation targets are normalized by variables.

Now the adaptation variable is defined as the following.

$$z \equiv J_e \dot{\omega}_e + J_{e1} \dot{\omega}_{c1} - T_{e,n} - T_{a1} + T_c = \omega_e \tau_e - \frac{\dot{\omega}_w \tau_o}{i_{t1} i_{f1}} \quad (48)$$

Here, since direct use of the sensor measurement derivatives may makes the system oversensitive to measurement noise, the engine acceleration and clutch 1 acceleration are substituted by those obtained by the unknown input observers to be introduced. For further noise rejection, proportional feedback term in the estimated engine acceleration and that in the estimated clutch 1 acceleration are excluded when being substituted to calculate z for $\hat{\tau}_o$ and $\hat{\tau}_e$ estimation, respectively.

$$\hat{z} \equiv \omega_e \hat{\tau}_e - \frac{\dot{\omega}_w \hat{\tau}_o}{i_{t1} i_{f1}} \quad (49)$$

To ensure that the system is causal and to draw an adaptive scheme from it, a filter is applied.

$$\dot{z} = -\gamma \left(z - \omega_e \tau_e + \frac{\dot{\omega}_w \tau_o}{i_{t1} i_{f1}} \right) \quad (50)$$

Thus the estimated dynamics is obtained as the following.

$$\dot{\hat{z}} = -\gamma \left(\hat{z} - \omega_e \hat{\tau}_e + \frac{\dot{\omega}_w \hat{\tau}_o}{i_{t1} i_{f1}} \right) \quad (51)$$

Here, the update laws are designed as shown next.

$$\dot{\hat{\tau}}_e = -\gamma_e \hat{\tau}_e + \gamma_e \omega_e \varepsilon \quad (52)$$

$$\dot{\hat{\tau}}_o = \gamma_l (\tau_{o,n} - \hat{\tau}_o) - \frac{\gamma_l}{i_{t1} i_{f1}} \dot{\omega}_w \varepsilon \quad (53)$$

where γ_e and γ_l are the adaptation gains and $\varepsilon \equiv z - \hat{z}$. γ_l is the filter gain that decides the level of dependency on the nominal normalized output shaft torque value.

For the stability analysis, let $\tilde{\tau}_e \equiv \tau_e - \hat{\tau}_e$, $\tilde{\tau}_o \equiv \tau_o - \hat{\tau}_o$ and simply choose a positive definite, decrescent, and radially unbounded Lyapunov candidate function as the following.

$$V = \frac{1}{2} \left(\varepsilon^2 + \frac{\gamma}{\gamma_e} \tilde{\tau}_e^2 + \frac{\gamma}{\gamma_l} \tilde{\tau}_o^2 \right) \quad (54)$$

This way, differentiation of the Lyapunov function with respect to time gives the following result.

$$\begin{aligned} \dot{V} &= \varepsilon \dot{\varepsilon} + \frac{\gamma}{\gamma_e} \tilde{\tau}_e \dot{\tilde{\tau}}_e + \frac{\gamma}{\gamma_l} \tilde{\tau}_o \dot{\tilde{\tau}}_o \\ &= \varepsilon \left\{ -\gamma \left(z - \omega_e \tau_e + \frac{\dot{\omega}_w \tau_o}{i_{t1} i_{f1}} \right) + \gamma \left(\hat{z} - \omega_e \hat{\tau}_e + \frac{\dot{\omega}_w \hat{\tau}_o}{i_{t1} i_{f1}} \right) \right\} \\ &\quad - \frac{\gamma}{\gamma_e} \tilde{\tau}_e \dot{\tilde{\tau}}_e - \frac{\gamma}{\gamma_l} \tilde{\tau}_o \dot{\tilde{\tau}}_o \\ &= \varepsilon \left(-\gamma \varepsilon + \gamma \omega_e \tilde{\tau}_e - \frac{\gamma}{i_{t1} i_{f1}} \dot{\omega}_w \tilde{\tau}_o \right) - \frac{\gamma}{\gamma_e} \tilde{\tau}_e \dot{\tilde{\tau}}_e - \frac{\gamma}{\gamma_l} \tilde{\tau}_o \dot{\tilde{\tau}}_o \\ &= -\gamma \varepsilon^2 + \gamma \omega_e \tilde{\tau}_e \varepsilon - \frac{\gamma}{i_{t1} i_{f1}} \dot{\omega}_w \tilde{\tau}_o \varepsilon - \frac{\gamma}{\gamma_e} \tilde{\tau}_e \dot{\tilde{\tau}}_e - \frac{\gamma}{\gamma_l} \tilde{\tau}_o \dot{\tilde{\tau}}_o \\ &= -\gamma \varepsilon^2 + \gamma \tilde{\tau}_e \left(\omega_e \varepsilon - \frac{1}{\gamma_e} \dot{\tilde{\tau}}_e \right) - \gamma \tilde{\tau}_o \left(\frac{\dot{\omega}_w \varepsilon}{i_{t1} i_{f1}} + \frac{1}{\gamma_l} \dot{\tilde{\tau}}_o \right) \\ &= -\gamma \varepsilon^2 \leq 0 \end{aligned} \quad (55)$$

by substituting the adaptive laws obtained in (52) and (53) at steady state assuming that the nominal engine torque and nominal output torque are found to be accurate during the steady state. Using this result, asymptotic stability can be reached from Barbalat's lemma. Further stability analysis for generalized cases is dealt in the later parts.

3.3. Engine dynamics-based unknown input observer

The main objective for this observer is to estimate the combined clutch 1 and clutch 2 torques. Thus the estimation target of the observer is defined as follows.

$$T_c \equiv T_{c1} + T_{c2} \quad (56)$$

Now recall the engine dynamics stated in (39), from which the engine acceleration can be isolated.

$$\dot{\omega}_e = \frac{T_e}{J_e} - \frac{T_{c1}}{J_e} - \frac{T_{c2}}{J_e} \quad (57)$$

Then (57) can be represented in terms of the combined clutch torque defined in (56).

$$\dot{\omega}_e = \frac{T_e}{J_e} - \frac{T_c}{J_e} \quad (58)$$

Based on the above, the observer is designed as shown next.

$$\dot{\hat{\omega}}_e = \frac{1}{J_e} \hat{T}_e - \frac{1}{J_e} \hat{T}_c + l_1 (\omega_e - \hat{\omega}_e) \quad (59)$$

$$\dot{\hat{T}}_c = -l_2 (\omega_e - \hat{\omega}_e) \quad (60)$$

Here, the engine torque is replaced by that obtained by the adaptive output shaft torque observer based on (46), and l_1 and l_2 are the observer gains to be tuned.

Since the sub-observer designed in (59) and (60) takes the form of a typical PI-type unknown input observer, the stability of the engine dynamics-based unknown input observer error can easily be shown under the assumption that the externally obtained variable is close to the actual state; hence the stability analysis is omitted. However, the stability of the entire observer system error is analyzed in the later parts.

3.4. Clutch dynamics-based unknown input observer

Similar to the case of engine dynamics-based unknown input observer, the main objective of the clutch dynamics-based unknown input observer is to estimate combined clutch 1 and clutch 2 torques represented from the perspective of clutch 1. Hence the estimation target is defined as follows.

$$T_{a1} \equiv T_{c1} + \frac{i_{t2} i_{f2}}{i_{t1} i_{f1}} T_{c2} \quad (61)$$

Now recall the clutch dynamics dealt in (3). Here, the clutch 1 speed is isolated to give the following relationship.

$$\dot{\omega}_{c1} = \frac{T_{c1}}{J_{e1}} + \frac{i_{t2} i_{f2}}{J_{e1} i_{t1} i_{f1}} T_{c2} - \frac{T_o}{J_{e1} i_{t1} i_{f1}} \quad (62)$$

Based on this, the unknown input observer is designed as shown next.

$$\dot{\hat{\omega}}_{c1} = \frac{1}{J_{12,1}} \hat{T}_{a1} - \frac{\hat{T}_o}{J_{12,1} i_{t1} i_{f1}} + l_3 (\omega_{c1} - \hat{\omega}_{c1}) \quad (63)$$

$$\dot{\hat{T}}_{a1} = l_4 (\omega_{c1} - \hat{\omega}_{c1}) \quad (64)$$

where l_3 and l_4 are the observer gains to be tuned. As it can be seen in (63), the clutch dynamics-based unknown input observer

requires the output shaft torque information. This output shaft torque is substituted by the output shaft torque estimation obtained in the adaptive output shaft torque observer based on (47).

Now that both $\hat{T}_c$ and $\hat{T}_{a1}$ are estimated, these values can be fused together to give the rough estimation of the clutch 1 torque and clutch 2 torque, and these rough estimations can be expressed as follows.

$$\hat{T}_{c1,UO} \equiv \frac{i_{t1}i_{f1}\hat{T}_{a1} - i_{t2}i_{f2}\hat{T}_c}{i_{t1}i_{f1} - i_{t2}i_{f2}} \quad (65)$$

$$\hat{T}_{c2,UO} \equiv \frac{i_{t1}i_{f1}\hat{T}_c - i_{t1}i_{f1}\hat{T}_{a1}}{i_{t1}i_{f1} - i_{t2}i_{f2}} \quad (66)$$

The clutch dynamics-based unknown input observer can alternatively be designed using the clutch 2 dynamics as well. Such alternative option can be useful for the application on actual vehicles when the gear selector is shifting on the first transfer shaft which involves a discrete change in inertia.

The individual clutch torque estimations acquired by using the unknown input observers are merely considered as the interim results, because the unknown input observers involve the phase lag issue by their nature, and also because the driveline model used to derive the observer equations was overly simplified. On the other hand, such nature provides the strong advantage of robustness and stability at steady states. Hence, these clutch torque estimations obtained by the unknown input observers are designed to assist the individual transfer shaft model-based observer to obtain the final estimation result.

3.5. Stability analysis

The integrated observer comprises of 8 states in total: two from the individual transfer shaft model-based observer, two from the adaptive output shaft torque observer, two from the engine dynamics-based observer, and two from the clutch dynamics-based observer. These states are shown in the following.

$$\hat{x} = [\hat{T}_{c1} \quad \hat{T}_{c2} \quad \hat{\tau}_e \quad \hat{\tau}_o \quad \hat{\omega}_e \quad \hat{T}_c \quad \hat{\omega}_{c1} \quad \hat{T}_{a1}]^T \quad (67)$$

By expanding and simplifying the entire system, the observer dynamics can be expressed as the following.

$$\dot{\hat{x}} = \hat{A}\hat{x} + \hat{B} \quad (68)$$

where

$$\hat{A} = [\hat{a}_{ij}] \quad (69)$$

which denotes that $\hat{A}$ is a matrix with $\hat{a}_{ij}$ as its i -th row j -th column element. Here, $\hat{A}$ is an 8 by 8 matrix with

$$\hat{a}_{11} = -L_1, \quad \hat{a}_{15} = \frac{J_e l_1 \gamma_t k_{t1} \dot{\omega}_w^2}{i_{t1}^2 (k_{t1} i_{f1}^2 + k_{t2} i_{f2}^2)}, \quad \hat{a}_{16} = -\frac{L_1 i_{t2} i_{f2}}{i_{t1} i_{f1} - i_{t2} i_{f2}},$$

$$\hat{a}_{17} = \frac{J_e l_3 \gamma_t k_{t1} \dot{\omega}_w^2}{i_{t1}^2 (k_{t1} i_{f1}^2 + k_{t2} i_{f2}^2)}, \quad \hat{a}_{18} = \frac{L_1 i_{t1} i_{f1}}{i_{t1} i_{f1} - i_{t2} i_{f2}}, \quad \hat{a}_{22} = -L_2,$$

$$\hat{a}_{25} = \frac{J_e l_1 \gamma_t k_{t2} i_{f2} \dot{\omega}_w^2}{i_{t1} i_{t2} i_{f1} (k_{t1} i_{f1}^2 + k_{t2} i_{f2}^2)}, \quad \hat{a}_{26} = \frac{L_2 i_{t1} i_{f1}}{i_{t1} i_{f1} - i_{t2} i_{f2}},$$

$$\hat{a}_{27} = \frac{J_e l_3 \gamma_t k_{t2} i_{f2} \dot{\omega}_w^2}{i_{t1} i_{t2} i_{f1} (k_{t1} i_{f1}^2 + k_{t2} i_{f2}^2)}, \quad \hat{a}_{28} = -\frac{L_2 i_{t1} i_{f1}}{i_{t1} i_{f1} - i_{t2} i_{f2}}, \quad \hat{a}_{33} = -\gamma_e,$$

$$\hat{a}_{35} = -J_e l_1 \gamma_e \omega_e, \quad \hat{a}_{44} = -\gamma_l, \quad \hat{a}_{47} = \frac{J_{e1} l_3 \gamma_t \dot{\omega}_w}{i_{t1} i_{f1}}, \quad \hat{a}_{53} = \frac{\omega_e}{J_e},$$

$$\hat{a}_{55} = -l_1, \quad \hat{a}_{56} = -\frac{1}{J_e}, \quad \hat{a}_{65} = l_2, \quad \hat{a}_{74} = -\frac{\dot{\omega}_w}{J_{e1} i_{t1} i_{f1}},$$

$$\hat{a}_{77} = -l_3, \quad \hat{a}_{78} = \frac{1}{J_{e1}}, \quad \hat{a}_{87} = -l_4,$$

and the rest of the elements are zero, and

$$\hat{B} = [\hat{b}_i] \quad (70)$$

an 8 by 1 matrix with

$$\hat{b}_1 = \left(\frac{k_{t1}}{i_{t1}^2} - \frac{k_{t1}^2 i_{f1}^2}{i_{t1}^2 (k_{t1} i_{f1}^2 + k_{t2} i_{f2}^2)} \right) \omega_{c1} - \frac{k_{t1} k_{t2} i_{f1} i_{f2}}{i_{t1} i_{t2} (k_{t1} i_{f1}^2 + k_{t2} i_{f2}^2)} \omega_{c2} - \frac{J_{e1} l_3 \gamma_t k_{t1} \omega_{c1} \dot{\omega}_w^2}{i_{t1}^2 (k_{t1} i_{f1}^2 + k_{t2} i_{f2}^2)} - \frac{J_e l_1 \gamma_t k_{t1} \omega_e \dot{\omega}_w^2}{i_{t1}^2 (k_{t1} i_{f1}^2 + k_{t2} i_{f2}^2)},$$

$$\hat{b}_2 = -\frac{k_{t1} k_{t2} i_{f1} i_{f2}}{i_{t1} i_{t2} (k_{t1} i_{f1}^2 + k_{t2} i_{f2}^2)} \omega_{c1} + \left(\frac{k_{t2}}{i_{t2}^2} - \frac{k_{t2}^2 i_{f2}^2}{i_{t2}^2 (k_{t1} i_{f1}^2 + k_{t2} i_{f2}^2)} \right) \omega_{c2} - \frac{J_{e1} l_3 \gamma_t k_{t2} i_{f2} \omega_{c1} \dot{\omega}_w^2}{i_{t1} i_{t2} i_{f1} (k_{t1} i_{f1}^2 + k_{t2} i_{f2}^2)} - \frac{J_e l_1 \gamma_t k_{t2} i_{f2} \omega_e \dot{\omega}_w^2}{i_{t1} i_{t2} i_{f1} (k_{t1} i_{f1}^2 + k_{t2} i_{f2}^2)},$$

$$\hat{b}_3 = J_e l_1 \gamma_e \omega_e^2, \quad \hat{b}_4 = \gamma_l \tau_{o,n} - \frac{J_{e1} l_3 \gamma_t \dot{\omega}_w}{i_{t1} i_{f1}} \omega_{c1}, \quad \hat{b}_5 = \frac{T_{e,n}}{J_e} + l_1 \omega_e,$$

$$\hat{b}_6 = -l_2 \omega_e, \quad \hat{b}_7 = l_3 \omega_{c1}, \quad \text{and} \quad \hat{b}_8 = l_4 \omega_{c1}.$$

Now, by representing the actual plant dynamics with the compliance model in (32) and (33) and the engine and clutch torque balance equations in (39) and (40), the error dynamics of the observer system can be obtained as follows.

$$\dot{\tilde{x}} = \dot{x} - \dot{\hat{x}} = \tilde{A} \tilde{x} \quad (71)$$

where

$$\tilde{A} = [\tilde{a}_{ij}] \quad (72)$$

and $\tilde{A}$ is an 8 by 8 matrix whose elements turn out to be equal to those of $\hat{A}$ – namely, $\tilde{a}_{ij} = \hat{a}_{ij}$ – after simplification.

Here, the unknown states of the nominal engine and output torque are assumed to be accurate during the steady state, and T_c and T_{a1} slowly varying. Now in order to show the asymptotic stability of the entire system with its states coupled together, the following Lyapunov candidate function is chosen, which is positive definite, radially unbounded, and decrescent.

$$V = \frac{1}{2} \left(\frac{1}{J_e^2 l_1 \gamma_e} \right) \tilde{\tau}_e^2 + \frac{1}{2} \left(\frac{1}{J_{e1}^2 l_3 \gamma_t} \right) \tilde{\tau}_o^2 + \frac{1}{2} \tilde{\omega}_e^2 + \frac{1}{2} \left(\frac{1}{J_e l_2} \right) \tilde{T}_c^2 + \frac{1}{2} \tilde{\omega}_{c1}^2 + \frac{1}{2} \left(\frac{1}{J_{e1} l_4} \right) \tilde{T}_{a1}^2 \quad (73)$$

Here, the system order is reduced by 2 for the sake of simplification. Such maneuver can be rationalized since the stability of the two omitted states, $\tilde{T}_{c1}$ and $\tilde{T}_{c2}$, is subordinate to that of the rest of the states. This claim is true since row 3 to 8 in (72) tells that the elements in the first two columns are equal to zero. This indicates that the error dynamics of the individual transfer shaft model-based observer has no effect on any other parts of the observer system. Hence, this error dynamics which corresponds to the first two rows of (72) can be dealt separately, about which Section 3.1 has mentioned.

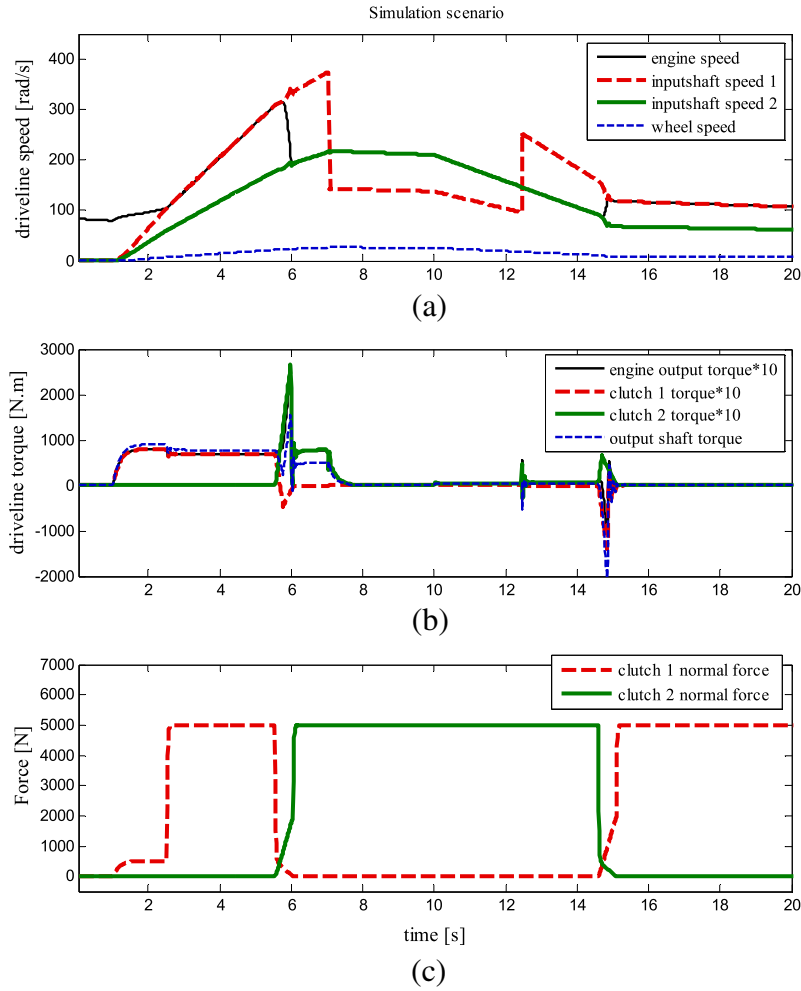


Fig. 6. Simulation scenario: (a) engine, clutch, and wheel speeds, (b) engine, clutch, and output shaft torques and (c) clutch normal forces exerted by the clutch actuators.

Taking the time derivative of (73) leads to the following.

$$\begin{aligned} \dot{V} = & \left(\frac{1}{J_e^2 I_1 \gamma_e} \right) \tilde{\tau}_e \dot{\tilde{\tau}}_e + \left(\frac{1}{J_{e1}^2 I_3 \gamma_t} \right) \tilde{\tau}_o \dot{\tilde{\tau}}_o + \tilde{\omega}_e \dot{\tilde{\omega}}_e + \left(\frac{1}{J_e I_2} \right) \tilde{T}_c \dot{\tilde{T}}_c \\ & + \tilde{\omega}_{c1} \dot{\tilde{\omega}}_{c1} + \left(\frac{1}{J_{e1} I_4} \right) \tilde{T}_{a1} \dot{\tilde{T}}_{a1} \end{aligned} \quad (74)$$

Now, substituting the error dynamics obtained in (71) gives the expanded equation as shown next.

$$\begin{aligned} \dot{V} = & -\frac{1}{J_e^2 I_1} \tilde{\tau}_e^2 - \frac{\gamma_l}{J_{e1}^2 I_3 \gamma_t} \tilde{\tau}_o^2 - l_1 \tilde{\omega}_e^2 - l_3 \tilde{\omega}_{c1}^2 \\ & - \frac{1}{J_e} \omega_e \tilde{\omega}_e \tilde{\tau}_e + \frac{\omega_e}{J_e} \tilde{\tau}_e \tilde{\omega}_e + \frac{\dot{\omega}_w}{J_{e1} I_1 I_3 \gamma_t} \tilde{\omega}_{c1} \tilde{\tau}_o - \frac{\dot{\omega}_w}{J_{e1} I_1 I_3} \tilde{\tau}_o \tilde{\omega}_{c1} \\ & - \frac{1}{J_e} \tilde{T}_c \tilde{\omega}_e + \frac{1}{J_e} \tilde{T}_c \tilde{\omega}_e + \frac{1}{J_{e1}} \tilde{T}_{a1} \tilde{\omega}_{c1} - \frac{1}{J_{e1}} \tilde{T}_{a1} \tilde{\omega}_{c1} \end{aligned} \quad (75)$$

Since multiple terms can be canceled with each other in (75), the following conclusion can be drawn.

$$\dot{V} = -\frac{1}{J_e^2 I_1} \tilde{\tau}_e^2 - \frac{\gamma_l}{J_{e1}^2 I_3 \gamma_t} \tilde{\tau}_o^2 - l_1 \tilde{\omega}_e^2 - l_3 \tilde{\omega}_{c1}^2 \leq 0 \quad (76)$$

Thus $\dot{V}$ is negative semi-definite. Now we define $E \equiv \{\tilde{x} \in \mathbb{R} : \tilde{\tau}_e, \tilde{\tau}_o, \tilde{\omega}_e, \tilde{\omega}_{c1} = 0\}$ in which $\dot{V} = 0$, and it can be deduced using the system description in (71) that the largest

invariant set in E is equal to the origin. Therefore, the entire error dynamics is asymptotically stable at the origin.

4. Estimation performance validation

4.1. Simulation result

The observer performance is tested with the gear shift scenario via simulation using a well-known driveline simulation tool Sim-Driveline. An intensive model just for the validation purpose is constructed to verify the suggested observer performance via simulation, and representative model parameters are shown in the following.

Engine inertia	$J_e = 0.2$
Torsional damper inertia	$J_d = 0.03$
Clutch 1 inertia	$J_{c1} = 0.0023$
Clutch 2 inertia	$J_{c2} = 0.0009$
Gear 1 inertia	$J_{g1} = 0.001$
Gear 2 inertia	$J_{g2} = 0.001$
Gear 3 inertia	$J_{g3} = 0.001$
Gear 4 inertia	$J_{g4} = 0.001$
Gear 5 inertia	$J_{g5} = 0.001$
Output shaft inertia	$J_o = 0.0002$

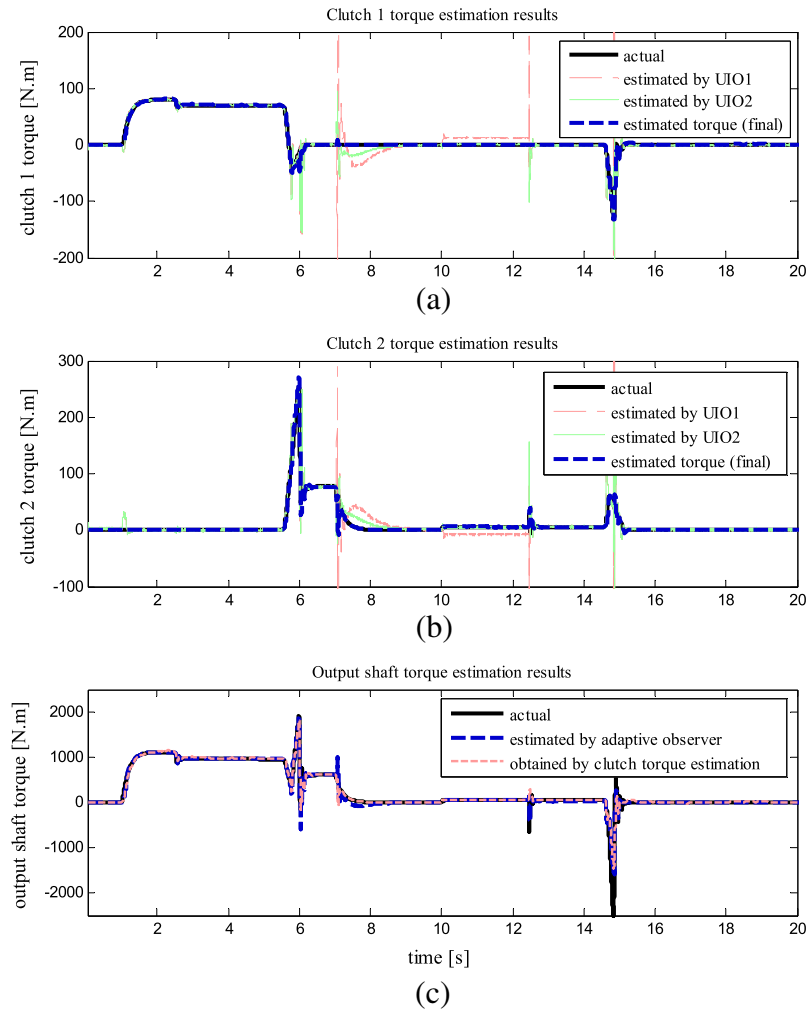


Fig. 7. Torque estimation results by simulation: (a) clutch 1 torque estimation, (b) clutch 2 torque estimation and (c) output shaft torque estimation.

Table 1

RMS errors of the estimated output shaft torque.

RMS errors (N m)	Clutch 1 torque estimation	Clutch 2 torque estimation	Output shaft torque estimation
	2.124	3.403	67.233

Torsional damper constant	$b_d = 10$
Transfer shaft 1 spring constant	$k_{t1} = 10,000$
Transfer shaft 1 damping constant	$b_{t1} = 50$
Transfer shaft 2 spring constant	$k_{t2} = 23,000$
Transfer shaft 2 damping constant	$b_{t2} = 50$
Output shaft spring constant	$k_o = 10,000$
Output shaft damping constant	$b_o = 20$
Gear 1 ratio	$i_{t1} = 3.78$
Gear 2 ratio	$i_{t2} = 2.18$
Gear 3 ratio	$i_{t3} = 1.43$
Gear 4 ratio	$i_{t4} = 1.03$
Gear 5 ratio	$i_{t5} = 0.84$
Final reduction gear ratio	$i_{f2} = 3.7$
Effective clutch 1 radius	$R_{c1} = 0.13$
Effective clutch 2 radius	$R_{c2} = 0.13$
Clutch 1 static (kinetic) friction coefficient	0.31(0.3)
Clutch 2 static (kinetic) friction coefficient	0.31(0.3)
Synchronizer static (kinetic) friction coefficient	0.35(0.3)

Transmission viscous drag coefficient	0.005
Transmission gear efficiency	0.94
Final reduction gear efficiency	0.86
Wheel radius	$r_w = 0.312$
Wheel inertia	$J_w = 2$
Vehicle mass	$m_v = 1600$
Tire rolling resistance coefficient	$K_{rr} = 0.015$
Aerodynamic drag coefficient	$C_d = 0.4$
Effective frontal area	$A_f = 3$

[Units are SI derived (kg, m, s, A)]

The effective shaft inertia values used for the models used in the observers are obtained using the above shown parameters. For instance, the shaft 1 and 2 inertia values at the moment the first and second gears are selected are shown in the following:

$$J_{e1} = J_{c1} + J_{g1} + (J_{g2} + J_{c2}) \cdot \left(\frac{i_{t2}}{i_{t1}}\right)^2, J_{e2} = J_{c2} + J_{g2} + (J_{g1} + J_{c1}) \cdot \left(\frac{i_{t1}}{i_{t2}}\right)^2.$$

The simulation involves a typical upshift during acceleration, and a downshift due to the decrease in speed. Detailed plots of the simulation scenario are shown in Fig. 6. Here, step input of 50% throttle is maintained for 6 s from 1 s after the start of simulation, and fuel cut is maintained afterward. In order to conduct a realistic simulation, the engine torque is deliberately reduced when the transmission undergoes gear shifts. Also, noises are

intentionally added to the driveline speed signals in order to emulate the actual sensor measurement noise.

As shown in Fig. 7 row (a) and (b), the simulation results demonstrate that the proposed observer effectively estimates each clutch torque separately, even during the transient states induced by abrupt gear shift. It can be observed that the estimation obtained by the integrated observer system gives higher accuracy and lower level of noise than that obtained by the unknown input observers alone. Such characteristic can be reverted depending on the quality of the sensor measurements and the level of noise. Hence, depending on the noise level of the sensors, making use of the unknown input observer estimation results can also be worthwhile, considering that the accuracy of the raw unknown input observer estimation results shown in Fig. 7 can easily be improved simply by signal processing. Fig. 7 row (c) reveals that, although the individual transfer shaft model-based observer takes a short time interval for convergence toward the feedback terms, the observer accurately identifies the output shaft torque during the transient state. It should be noted that the output shaft torque in the plot is calibrated by the effective gear transmission efficiency.

One may notice that the vehicle decelerates significantly between the time interval 10–15 s, and this is because the simulation was conducted with the road inclination angle of 0.1 radian (approximately 5.7°) uphill. Through the simulation on uphill,

robust estimation performance of the adaptive output torque observer can be shown, and the result plotted in Fig. 7(c) reveals that the adaptive observer estimation successfully estimates the output shaft torque without any steady state error during the vehicle deceleration on uphill. Although the adaptation was conducted as a function of wheel acceleration, the adaptive scheme using the driveline dynamics effectively copes with the disturbance of uphill.

Here, for the purpose of noise rejection, the clutch torque estimations are attenuated to zero when the clutch actuator position indicates that the clutch is before the kissing point and is completely disengaged.

In order to quantitatively show the clutch torque estimation performance, RMS errors of the estimation results are shown in Table 1.

Considering that the clutch torque rises up to as high as 140 N m and 265 N m in case of clutch 1 and 2 and output torque rises up to as high as 2500 N m, the RMS estimation error indicated above is sufficiently small for the estimator to be applied to the clutch control.

4.2. Experiment result

In order to show the production vehicle application potential of the proposed clutch observer, it is tested on the actual vehicle

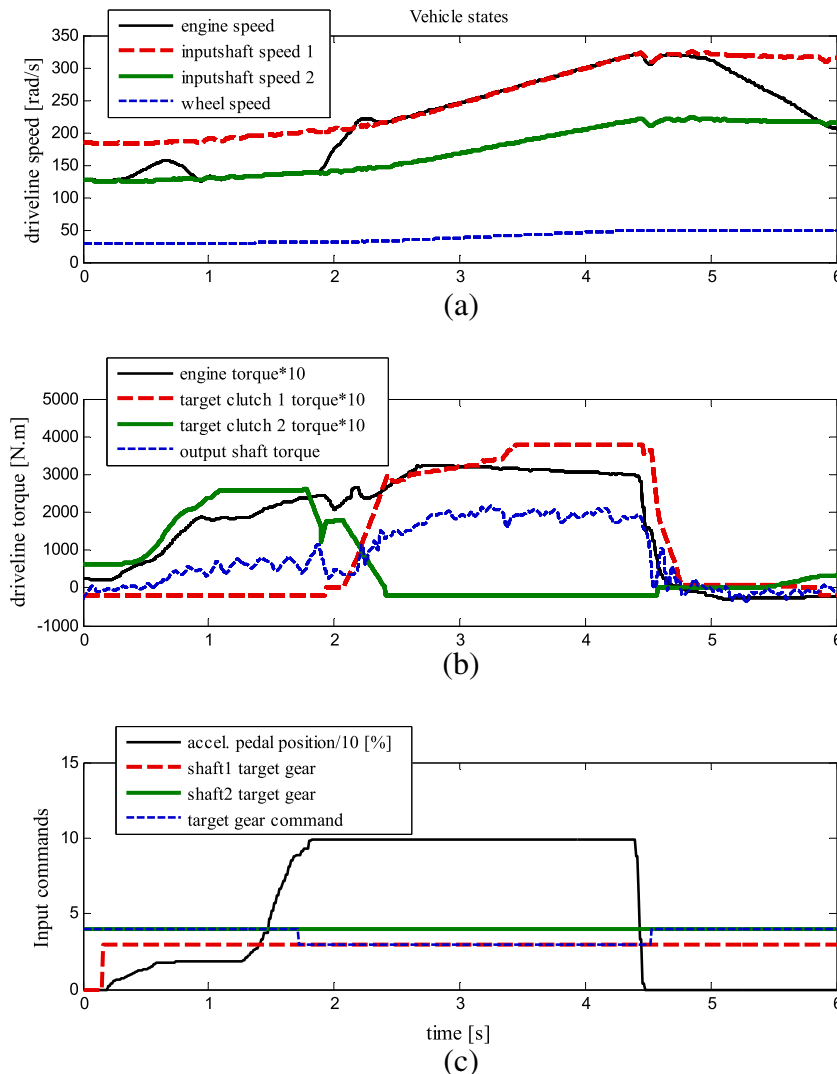


Fig. 8. Experiment scenario: (a) engine, clutch, and wheel speeds, (b) nominal engine torque from CAN, target clutch torques from TCU, measured output shaft torque and (c) throttle input and TCU commands.

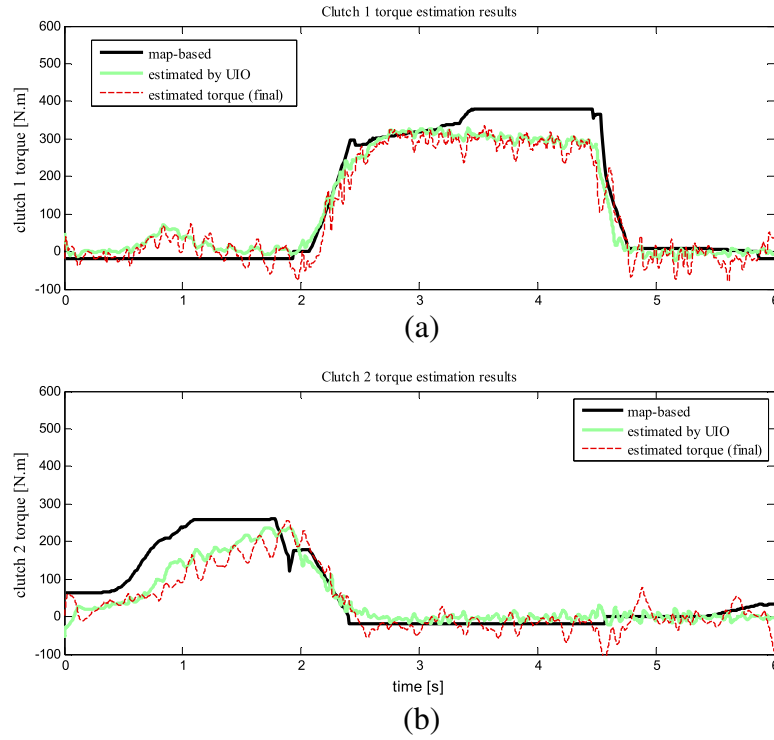


Fig. 9. Torque estimation results by experiment: (a) clutch 1 torque estimation and (b) clutch 2 torque estimation.

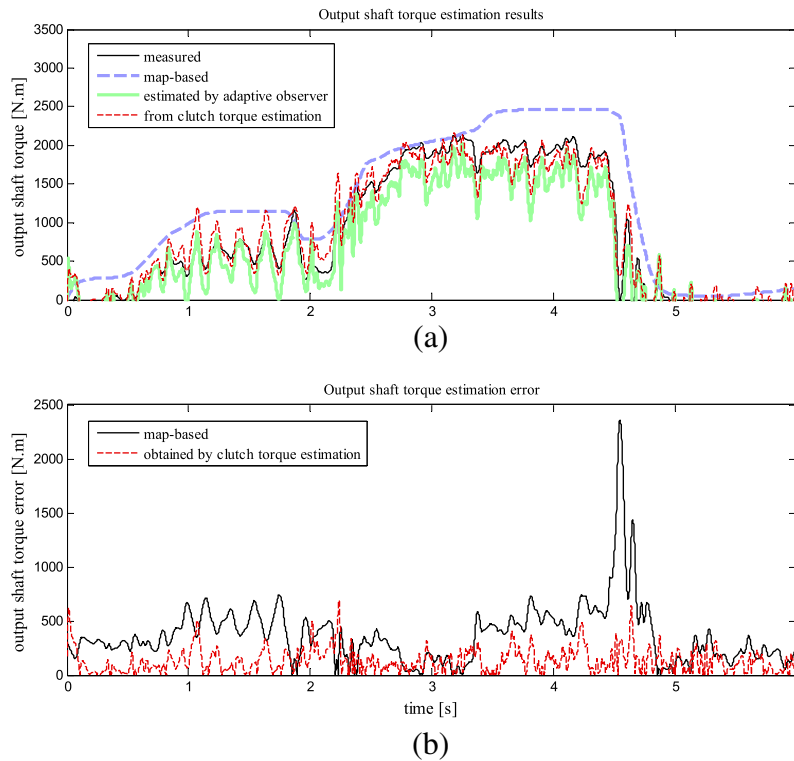


Fig. 10. Output shaft torque estimation results by experiment: (a) plot of measured and estimated output shaft torque, (b) clutch 2 torque estimation and (c) errors of the output shaft torque estimations.

mounted with the dual clutch transmission. Speed sensor measurements and CAN/TCU signals during the experiment are shown in Fig. 8. The model parameters identified in Section 2.3 are directly applied to the corresponding parts of the designed obser-

ver, and the observer performance has been tested on the real-time basis.

Most importantly, the plots in Fig. 9 show the clutch torque estimation results. As implied by the estimation results obtained

based on the simulation, the test results reveal the effectiveness of the proposed observer qualitatively, whose estimated clutch torques indeed changes in accordance with the map-based target clutch torque which directly relates to the actuator position. Here, it must be noted that the map-based information is not shown as the absolutely credible clutch torque value, but instead as a rough reference (which is currently the only existing method to monitor each clutch torque in conventional clutch control units).

As assumed in (29) and (30), the transfer shaft torque is considered equivalent to the clutch torque if corrected by the appropriate gear ratio. Such assumption may lead to a slight discrepancy between the estimated and actual clutch torque, which stands out when the clutch is disengaged. Disengagement of the clutch technically must yield zero torque transmitted through the clutch. However, due to the effect of the clutch inertia, the torque transmitted through the transfer shaft may not be completely zero. As an easy solution to this issue, clutch position information can be exploited to increase the accuracy of the clutch torque estimation.

In fact, the above-mentioned nature of the proposed estimator provides the strength of being able to monitor the stick-slip phenomenon and resonance within the transmission shafts. Indeed, oscillations can be observed in the estimated results, and they are the signs of torsional resonance in the transfer shafts. Furthermore, the proposed method accurately identifies the clutch torque during an abrupt gear engagement or disengagement during gear shifts, such as the case around 4.5 s in Fig. 9, whereas the conventional position-torque map-based approach involves high amount of error even with only monitoring the output shaft torque.

Due to the technical difficulty of installing torque measuring sensors within the clutch pack or on the input shafts inside the transmission housing, clutch torque estimations could not be compared against the measured clutch torques. However, the quantitative validation of the clutch torque estimation can be conducted via using the output shaft torque measurement. To do so, the clutch torque estimation results are converted to the output shaft torque by using the relationship set up in (31), and the output shaft torque estimated this way can be compared with the measured output shaft torque. Such comparison is shown in Fig. 10, in which the output shaft torque estimated by the proposed method indeed shows the highest accuracy with the RMS error of 166.13, whereas the RMS errors of the estimation obtained by the adaptive output shaft observer alone and that obtained by the position-torque map mount up to 257.60 and 468.80, respectively. Considering that the RMS error of the estimation is no more than approximately one-tenth of the utilized output shaft torque, especially during the transmission of maximum torque, the proposed observer is considered to be effective for the real-time monitoring of the driveline torques.

5. Conclusion

This study has proposed a novel modeling method to consider the internal compliance of the dual clutch transmissions so that it reflects individual transfer shaft dynamics to enable expressing clutch torques without having to use complex friction models, and an original observer to effectively identify each clutch torque of the dual clutch transmission system separately. By only using the information and sensors that are already available in current production cars, the proposed observer system with its subcomponents of individual transfer shaft model-based observer, adaptive output shaft torque observer, and unknown input observers effectively identifies individual clutch torque with high accuracy, whose estimation performance is verified via both simulation and experiments using an actual vehicle. Summarizing the paper, noteworthy contributions of the proposed work are primarily twofold:

driveline modeling method which allows internal compliance identification through using individual transfer shaft dynamics, and ability to estimate the torques transmitted by each clutch separately during gear shifts. These original contributions are thoroughly verified through theoretical stability analysis, simulation, and real car experiments. With the application of the proposed work on actual cars, development of clutch actuator controllers with improved precision and efficiency is anticipated.

Acknowledgment

This work was supported by NRF (The National Research Foundation of Korea) Grant funded by the Korea government (MEST) (No. 2012-0000991) and Global Ph.D. Fellowship Program, and MKE (The Ministry of Knowledge Economy), Korea, under the CITRC (Convergence Information Technology Research Center) support program (NIPA-2012-H0401-12-2003) supervised by the NIPA (National IT Industry Promotion Agency).

References

- [1] Szimandl B, Németh H. Dynamic hybrid model of an electro-pneumatic clutch system. *Mechatronics* 2013;23(1):21–36.
- [2] Yoon YS, Kim SJ, Kim KS. Conceptual design of economic hybrid vehicle system using clutchless geared smart transmission. *Int J Autom Technol* 2013;14(5):779–84.
- [3] Kim SJ, Song C, Kim K, Yoon YS. Analysis of the shifting behavior of a novel clutchless geared smart transmission. *Int J Autom Technol* 2014;15(1):125–34.
- [4] Shin JW, Kim JO, Choi JY, Oh SH. Design of 2-speed transmission for electric commercial vehicle. *Int J Autom Technol* 2014;15(1):145–50.
- [5] Matthes B. Dual clutch transmissions—lessons learned and future potential. In: Proceedings of the transmission and driving systems symposium. SAE Paper No. 2005-01-1021; 2005.
- [6] Shin BK, Hahn JO, Lee KI. Development of shift control algorithm using estimated turbine torque. SAE Paper No. 2000-01-1150; 2000.
- [7] Jeong HS, Lee KI. Friction coefficient, torque estimation, smooth shift control law for an automatic power transmission. *J Mech Sci Technol* 2000;14(5):508–17.
- [8] Gao B, Chen H, Zhao H, Sanada K. A reduced-order nonlinear clutch pressure observer for automatic transmission. *IEEE Trans Control Syst Technol* 2010;18(2):446–53.
- [9] Pettersson M, Nielsen L. Gear shifting by engine control. *IEEE Trans Control Syst Technol* 2000;8(3):495–507.
- [10] Pettersson M. Driveline modeling and control. Ph.D. Thesis, Linköping University, Sweden; 1997.
- [11] Pettersson M, Nielsen L. Diesel engine speed control with handling of driveline resonances. *Control Eng Pract* 2003;11(3):319–28.
- [12] Webersinke L, Augenstein L, Kiencke U. Adaptive linear quadratic control for high dynamical and comfortable behavior of a heavy truck. SAE Paper No. 2008-01-0534; 2008.
- [13] Yi K, Hedrick K, Lee SC. Estimation of tire-road friction using observer based identifiers. *Veh Syst Dynam* 1999;31(4):233–61.
- [14] Shin BK, Hahn JO, Yi K, Lee KI. A supervisor-based neural-adaptive shift controller for automatic transmissions considering throttle opening and driving load. *KSME Int J* 2000;14(4):418–25.
- [15] Misawa E, Hedrick J. Nonlinear observers – a state-of-the-art survey. *ASME J Dynam Syst Meas Control* 1992;111:344–52.
- [16] Johnson C. Theory of disturbance-accommodating controllers. *Control Dynam Syst* 1976;12:387–489.
- [17] Kim J, Oh J, Choi SB. Nonlinear estimation method of a self-energizing clutch actuator load. In: International conference on innovative engineering systems. Alexandria, Egypt, December 2012.
- [18] Ompusunggu AP, Sas P, Brussel HV. Modeling and simulation of the engagement dynamics of a wet friction clutch system subjected to degradation: an application to condition monitoring and prognostics. *Mechatronics* 2013;23(6):700–12.
- [19] Yao X, Qin D, Liu Z. Simulation and analysis of the dual-clutch transmission starting control. *J Chong Univ (Nat Sci Ed)* 2007;1(003).
- [20] Kim Jinsung, Choi Seibum B. Control of dry clutch engagement for vehicle launches via a shaft torque observer. In: American control conference (ACC); 2010. p. 676–81.
- [21] Wu MX, Zhang JW, Lu TL, Ni CS. Research on optimal control for dry dual-clutch engagement during launch. *Proc Inst Mech Eng, Part D: J Autom Eng* 2010;224(6):749–63.
- [22] Tran VN, Lauber J, Dambrine M. Sliding mode control of a dual clutch during launch. In: The second international conference on engineering mechanics and automation (ICEMA2). Hanoi, August 16–17 2012.
- [23] Zhao ZG, Jiang JL, Yu ZP, Zhang T. Starting sliding mode variable structure that coordinates the control and real-time optimization of dry dual clutch transmissions. *Int J Autom Technol* 2013;14(6):875–88.

- [24] Han K, Yoon Y. Clutch transmissible torque estimation for dry dual clutch transmission control. In: Proceedings of the FISITA 2012 world automotive congress. Berlin, Heidelberg: Springer; 2013. p. 449–56.
- [25] Deur J, Ivanović V, Herold Z, Kostelac M, Tseng HE. Dry clutch control based on electromechanical actuator position feedback loop. *Int J Veh Des* 2012;60(3):305–26.
- [26] Kim J, Choi SB, Lee H, Kang J, Hur M. System modeling and control of a clutch actuator system for dual clutch transmissions. In: KSAE annual conference & exhibition; 2011. p. 1253–7.
- [27] Liu Y, Qin D, Jiang H, Zhang Y. A systematic model for dynamics and control of dual clutch transmissions. *J Mech Des* 2009;131:061012.
- [28] Ni C, Lu T, Zhang J. Gearshift control for dry dual-clutch transmissions. *Wseas Trans Syst* 2009;8(11):1177–86.
- [29] Gao B, Chen H, Ma Y, Sanada K. Design of nonlinear shaft torque observer for trucks with automated manual transmission. *Mechatronics* 2011;21(6): 1034–42.
- [30] Galvagno E, Velardocchia M, Vigliani A. Dynamic and kinematic model of a dual clutch transmission. *Mech Mach Theory* 2011;46(6):794–805.
- [31] Walker Paul D, Zhang Nong. Modelling of dual clutch transmission equipped powertrains for shift transient simulations. *Mech Mach Theory* 2013;60:47–59.
- [32] Oh Jiwon, Kim Jinsung, Choi Seibum. Design of estimators for the output shaft torque of automated manual transmission systems. In: Proceedings of IEEE conference on industrial electronics and applications 2013. Melbourne, Australia; 2013.
- [33] Ioannou PA, Sun J. Robust adaptive control. New Jersey, Englewood Cliffs: Prentice-Hall; 1964.