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Control of Dry Clutch Engagement for Vehicle Launches via a Shaft Torque Observer

Jinsung Kim and Seibum B. Choi

***Abstract*— Automated Manual Transmission vehicles are generally equipped with the servo-actuated clutch engagement system. In that system, shift quality depends on the engaging time and controlling the clutch normal force. It is therefore crucial to pay attention to the control performance in order to guarantee smooth engagement of the clutch. This article proposes the control architecture that consists of the speed control for synchronization and the torsion control for reducing shift shocks incorporating a shaft torque observer. The proposed control structure assures a fast engagement while preventing an abrupt change in vehicle acceleration at the same time.**

I. Introduction

Launch control of a vehicle from a standstill is the most difficult part for the entire automatic transmission control system without a torque converter. It is especially problematic to control the engagement of dry clutch systems. Gear shift or vehicle launch control with dry clutch has conflict between driverability and fast engagement [1], [2]. In manual transmission (MT) vehicles, the duration of clutch engagement has not an effect on the shift feeling because the starting and end points of clutch operations are determined by driver's subjective skill and intention. On the other hand, a control strategy to determine the optimal shifting time is particularly important for a vehicle equipped with servo-actuated clutches like Automated Manual Transmissions (AMT) or Dual Clutch Transmissions (DCT) since its engagement point is determined by the transmission control unit (TCU) regardless of driver's intention. In other words, the drivers on AMTs or DCT vehicle are more sensitive than the one on MT vehicles. Therefore, highly sophisticated clutch control strategy must be needed to satisfy driver's comfort.

Many control methodologies of dry clutch engagement have been discussed through many research activities. Serrarens et al. introduce the dynamic behavior of automotive dry clutches and a decoupling PI controller for engine and torque control [3]. The methods based upon an optimal control approach are presented in several works. Heijden et al. present a piecewise LQ controller and compare its results with a model predictive control strategy [2]. Garofalo et al. suggest the engagement controller for a dry clutch by using a linear quadratic regulator [4]. Gliemo et al. also suggest LQ method that makes use of minimum time for clutch lock-up and derivative of normal force as a cost function [1], [5]. Dolcini et al. propose a lurch avoidance strategy based on a LQ method with the dynamic Lagrangian multipliers [6]. Nonlinear control method based on backstepping technique [7] and a hierarchical approach consists of decoupled feedback loops have been proposed in terms of gearshift control [8], respectively. Flatness-based clutch control for AMT is developed by Horn et al [9].

Jinsung Kim is with the Department of Mechanical Engineering, KAIST, Daejeon, Korea. jsk@kaist.ac.kr

Seibum B. Choi is with the faculty of the Department of Mechanical Engineering, KAIST, Daejeon, Korea. sbchoi@kaist.ac.kr

This paper provides the development of a smooth but fast launch controller based on a shaft torque observer. It is useful to reduce shift shocks and make the system reach the stick phase as fast as possible within a physical limitation of the actuator. Shaft torque observer is introduced and employed to minimize residual vibrations of the shaft and also to generate its reference trajectory.

II. Powertrain Model

The automotive clutch is a power transmitting element from the engine to wheels. Therefore, the connecting devices of the clutch such as engine and shaft are necessary to be evaluated for the feasibility study of a developed clutch controller [10]. This section gives the detailed driveline model which is based on the works of Petterson [11] and Serraens et al. [3]. Some modifications are conducted to make it simple enough to be suitable for subsequent controller design.

A. Engine Model

Since the works of this research concentrates only on the transmission parts, the detailed dynamic behavior of the engine is not considered here. Thus, the net engine torque used in a first order dynamics (1) is produced from a steady-state engine map whose data is obtained based upon experimental tests using a dynamometer.

$$T_e = T_e(\omega_e, \alpha) \tag{1}$$

Consequently, the engine dynamics can be derived from (1) by replacing the load torque with the clutch torque.

$$J_e \dot{\omega}_e = T_e - T_c \tag{2}$$

where, J_e is the moment of inertia of the engine, ω_e the engine speed, T_e the engine torque calculated from (1), and T_c the load torque corresponding to the clutch torque generated by an electromechanical or hydraulic actuator.

B. Clutch Model

The dynamics of the clutch is represented as

$$J_c \dot{\omega}_c = T_c - T_d - b_c(\omega_c - \omega_t) \tag{3}$$

where J_c is the moment of inertia of the clutch, b_c the damping coefficient, ω_c and ω_t the rotational speed of clutch

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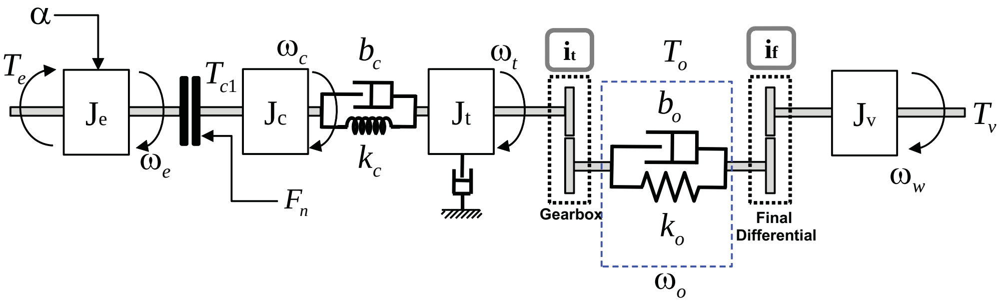

Fig. 1. Vehicle driveline diagram

and transmission, respectively. T_c is the torque from the clutch actuator in a slip phase with a form,

$$T_c = \mu R_c F_n sgn(\omega_e - \omega_c) \tag{4}$$

where μ is dry friction coefficient, R_c the clutch radius, and F_n the normal actuation force. T_d is added to model the damper spring built in series on the friction disk. The main purpose of using the damper and spring is to insulate vibrations which especially occur frequently during clutch engagement. It has a nonlinear characteristic that the damping torque varies with the range of torsion angle [12]. The clutch nonlinearity should be therefore considered if the inertia of the clutch is not neglected. It can be modeled as a piecewise linear function as given by the functions (5) and (6).

$$T_d(x) = \begin{cases} k_{c1}x & \text{if } \theta_{c1} \leq x \leq \theta_{c2} \\ k_{c2}\theta_{c2} + k_{c2}(x-\theta_{c2}) & \text{if } x > \theta_{c2} \\ k_{c1}\theta_{c1} + k_{c2}(x-\theta_{c1}) & \text{if } x < \theta_{c1} \end{cases} \tag{5}$$

where,

$$k_c(x) = \begin{cases} k_{c1} & \text{if } |x| \leq \theta_{c1} \\ k_{c2} & \text{else.} \end{cases} \tag{6}$$

Note that the wider the range of allowable clutch torsion the higher the clutch damper spring effect [13]. Coupled with this, the dynamics around clutch cover can be described as

$$J_t \dot{\omega}_t = T_d + b_c(\omega_c - \omega_t) - b_t \omega_t - \frac{T_o}{i_t} \tag{7}$$

where b_t is the damping coefficient of the transmission, i_t the speed reduction ratio of a gearbox, and T_o the drive shaft torque which is represented in the following subsection.

C. Driveshaft and Gearbox Model

The drive shaft is modeled with a spring-damper element to take into account torsional flexibility. It is given by

$$T_o = k_o\left(\frac{\theta_t}{i_t} - \theta_o\right) + b_o\left(\frac{\omega_t}{i_t} - \omega_o\right) \tag{8}$$

where, the variables k_o and, b_o denote the stiffness and damping coefficient of the output shaft, respectively. Speed reduction may be directly represented by the gear ratio i_t for

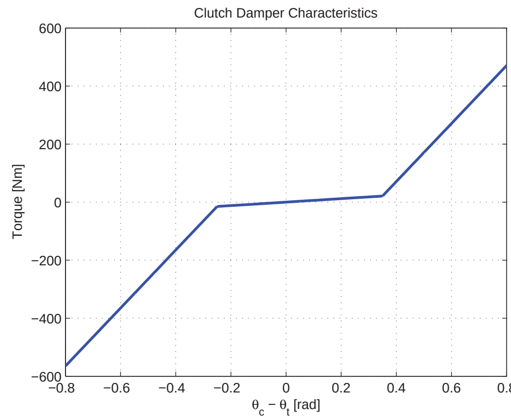

Fig. 2. Clutch damper characterisitics

transmission and i_f for final differential, i.e., $\omega_t = i_t \omega_o$ and $\omega_o = i_f \omega_w$.

Since the vehicle jerk performance depends primarily on driveshaft torque oscillations, this part will be used as a criterion to determine the driver's and passenger's comfort.

D. Wheel dynamics

In terms of overall vehicle dynamics, the tire is a very important element due to its material property and complexity. However, since it has much more effect on the dynamic behavior of the vehicle mass rather than the clutch comfort, the tire is accounted as a rolling element without slip, i.e. $v_v = r_w \omega_w$. Subsequently, the wheel dynamics is

$$J_w \dot{\omega}_w = T_w - T_{trac} \tag{9}$$

where

$$T_w = i_f T_f = i_f T_o, \qquad T_{trac} = F_{trac} r_w$$

and T_f is the torque applied to the final differential. It is equal to the output shaft torque T_o under the assumption that the final differential is stiff. The driving torque T_w is the output shaft torque multiplied by final drive ratio i_f. J_w is the wheel inertia. T_{trac} is the traction torque applied on the tire produced by the friction between the tire and a road surface.

E. Vehicle longitudinal model

In turn, overall vehicle is considered as a lumped mass M_v. Accordingly, vehicle longitudinal dynamics is described as

$$M_v \dot{v}_v = F_{trac} - F_{roll} - F_{aero} - M_v g \sin\theta_r \tag{10}$$

$$F_{roll} = K_r M_v g \cos\theta_r, \qquad F_{aero} = \tfrac{1}{2}\rho C_d A_F {v_x}^2$$

where, M_v is the vehicle mass, g the acceleration of gravity, θ_r road grade, K_r the rolling stiffness coefficient, ρ mass density of air, C_d coefficient of aerodynamic resistance, A_F the frontal area of a vehicle, v_x the vehicle speed, and r_w the effective wheel radius. This equation can be explained by force balance between the tractive force and the loads such as aerodynamic and rolling resistance [14], [15]. Combining equations (9) and (10) yields

$$J_v \dot{\omega}_w = i_f T_o - T_{load} \tag{11}$$

where,

$$J_v = J_w + m_v {r_w}^2. \tag{12}$$

The vehicle inertia J_v can be obtained by adding the wheel inertias to the equivalent inertia of the vehicle mass. The external load torque T_{load} in above equation is

$$T_{load} = \left(M_v g \sin\theta_r + K_r M_v g \cos\theta_r + \frac{1}{2}\rho C_d A_F {v_x}^2 \right) r_w. \tag{13}$$

The overall drivetrain dynamics developed in this section is described in Fig. 1.

III. Engagement Control

In this section, the speed control and the shaft torsion control strategies are. They are designed based on the theory of sliding mode control [16], [17] and applied with respect to the relative speed between the engine and the clutch. In addition, the shaft torque observers are employed to improve the performance of the designed two engagement control laws. [3], [11], [12].

A. Speed Control

Intuitively, the objective of clutch engagement can be achieved by synchronizing engine speed and clutch speed. Such a controller will be designed using the normal force from the clutch actuator as a control input.

Before designing a control law, driveline model introduced in Section II is simplified. Combining equations (3), (7), and (11) gives the following relationship under the assumption that the spring-damper of the friction disk is negligible.

$$J_{c1} \dot{\omega}_c = T_c - b_t \omega_c - \frac{T_{load}}{i_t i_f} \tag{14}$$

where J_{c1} is lumped inertia represented as $J_{c1} = J_c + J_t + \frac{J_v}{{i_t}^2 {i_f}^2}$. A speed controller can be investigated for clutch engagement control. Intuitively, the objective of clutch engagement can be achieved by synchronizing engine and clutch speed, i.e., $\omega_e = \omega_c$. A sliding surface is defined as a function of the speed difference between the engine and the clutch.

$$S_d = (\omega_e - \omega_c) + \lambda_d(\theta_e - \theta_c) \tag{15}$$

Note that the second term in above definition is expected to play a role in reducing the steady-state error for synchronization by means of an integral action. Subsequently, the time derivative of the sliding surface is

$$\dot{S}_d = \dot{\omega}_e - \dot{\omega}_c + \lambda_d(\omega_e - \omega_c) \tag{16}$$

To make the surface (15) attractive, define the desired surface dynamics as

$$\dot{S}_d = -K_d S_d \tag{17}$$

where, K_d is a design parameter.

Combining equations (2), (14) and (17) gives a speed control law with integral action:

$$F_n = \frac{1}{\mu R_c J_{c2}} \left[\frac{T_e}{J_e} + \frac{b_t \omega_c}{J_{c1}} + K_d \omega_r + \lambda_d \omega_r + K_d \lambda_d \int \omega_r \right] \tag{18}$$

where, ω_r is the relative speed between the engine and the clutch speed, i.e., $\omega_r = \omega_e - \omega_c$ and $J_{c2} = \frac{J_{c1}+J_e}{J_{c1}J_e}$. In order to further simplify a gain tuning procedure, two design parameters are set to the same value, i.e., $K_d = \lambda_d$, then equation (18) becomes:

$$F_n = \frac{1}{\mu R_c J_{c2}} \left[\frac{T_e}{J_e} + \frac{b_t \omega_c}{J_{c1}} + 2\lambda_d \omega_r + {\lambda_d}^2 \int \omega_r \right] \tag{19}$$

This control law includes a proportional and an integral feedback term in addition to feedforward information. Note that since the external load is quite uncertain and non-measurable, it cannot be used as a control input. However, the value of the external load can be estimated indirectly through vehicle acceleration since it reflects the difference between the engine torque and the external load. Generally, the engine torque signal from the engine control unit is very reliable. Therefore, only the engine torque is used for speed synchronization instead of the external load information. Based upon above equation and discussion, the speed controller for the clutch engagement can be designed as follows.

$$F_n = \frac{1}{\mu R_c} \left(\frac{J_{c1} J_e}{J_{c1} + J_e} \right) \left[\frac{T_e}{J_e} + \frac{b_t \omega_c}{J_{c1}} + K_d S_d \right] \tag{20}$$

During the vehicle launch, the speed difference between the engine and the clutch is relatively high since the engine speed has to remain always above a certain idle speed. Therefore, the control input should be rate limited as indicated in (21) to avoid an abrupt engagement of the dry clutch and potential engine stall.

$$F_n|_k - F_n|_{k-1} \quad \leq \quad F_{limit} \Delta T \tag{21}$$

where, $F_n|_k$ is the normal force at a time step k, F_{limit} a constant rate limit, and ΔT a sampling rate. The simulation results are as follows. Fig. 3 shows the resulting speed of

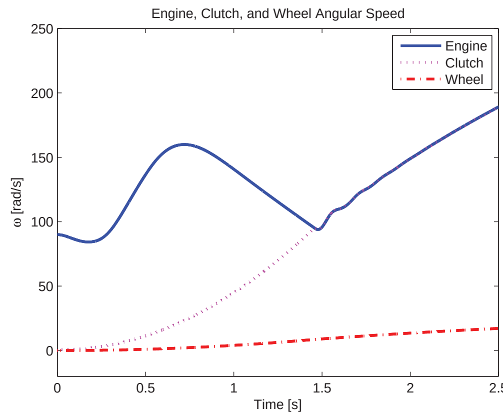

Fig. 3. Modified speed control for clutch engagement during a vehicle launch: angular speed of each element

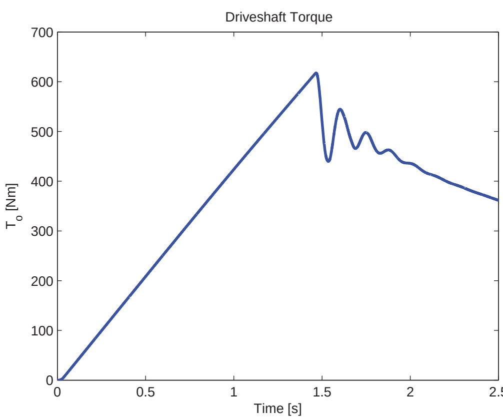

Fig. 4. Modified speed control for clutch engagement during a vehicle launch: Output torque

each driveline element for the speed control. Although the clutch is synchronized, the output shaft torque is oscillating as shown in Fig. 4 after the clutch is locked up. The vehicle jerk is a very important factor since it is associated with driverability. Fig. 5 shows that the speed controller causes the jerk to drop abruptly which means the driving comfort is deteriorated.

B. Torsion Control

The speed controller can achieve its basic objective for clutch engagements. However, it gives rise to another problem of driveline oscillation or shuffle due to excessive slip. The slew rate of the normal force must be increased to realize fast engagement. However, this does not ensure smooth engagements and may induce residual oscillation of the drive shaft. Therefore, this is considered a conflicting problem between the driving comfort and minimizing the engagement time. Accordingly, optimal control approach has been discussed by many researchers as mentioned before [1], [6]. In that case, two actuators are generally taken into account as control inputs. One is the clutch normal force and the other

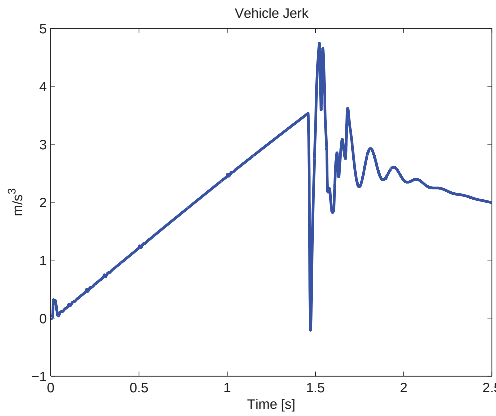

Fig. 5. Modified speed control for clutch engagement during a vehicle launch: vehicle jerk

is the engine torque. In this study, a desired performance is considered to be achieved using only the clutch actuator without any engine control. Therefore, a driveshaft torsion controller different from the speed controller is added as a function of the output shaft torsion and its rate. It uses only the clutch actuator to control the engagement.

First, define a sliding surface for the output shaft torsion as

$$S_t = \omega_{tor} + \lambda_t \theta_{tor} \tag{22}$$

where $\dot{\theta}_{tor} = \frac{\omega_t}{i_t} - i_f \omega_w$. Let S_t satisfy the following condition

$$\dot{S}_t = \dot{\omega}_{tor} + \lambda_t \omega_{tor} := -K_t S_t \tag{23}$$

For the purpose of torsion control, driveline model could be simplified by neglecting the compliance of the transmission. Combining equation (3) and (7) under the condition of $\omega_c = \omega_t$ and neglecting the transmission viscous damping yields

$$J_{t2}\dot{\omega}_t = T_c - \frac{T_o}{i_t} \tag{24}$$

where J_{t2} is the lumped inertia defined as $J_{t2} = J_c + J_t$. Substituting equation (24) and (11) into (23) yields:

$$\frac{T_c}{J_{t2} i_t} - \frac{T_o}{J_{t2} i_t^2} - \frac{i_f{}^2 T_o}{J_v} + \frac{i_f T_{load}}{J_v} + \lambda_t \omega_{tor} = -K_t S_t \tag{25}$$

The resulting control law is determined as follows.

$$F_n = \frac{J_{t2} i_t}{\mu R_c} \left[\frac{T_o}{J_{t2} i_t^2} + \frac{i_f{}^2 T_o}{J_v} - \lambda_t \omega_{tor} - K_t S_t \right] \tag{26}$$

This controller could be used to prevent the torsional vibration of the drive shaft right after the clutch engagement. At the stage of a standing start of vehicle with automated clutch systems, the transmission torque can be reduced incorporating speed synchronization between the engine and clutch speed right before the clutch engagement. Generally, this is implemented by rule-based approach without a closed-loop control. It is disadvantageous to deal with the abrupt change

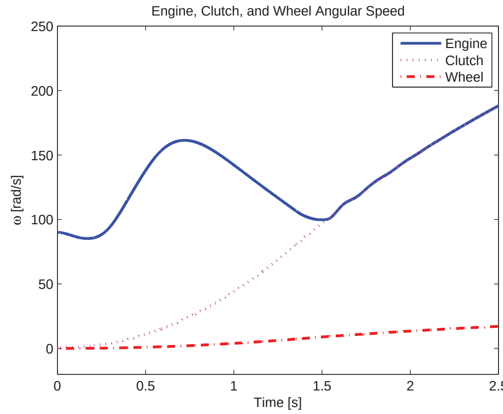

Fig. 6. Torsion control for clutch engagement during a vehicle launch: Angular speed of each element

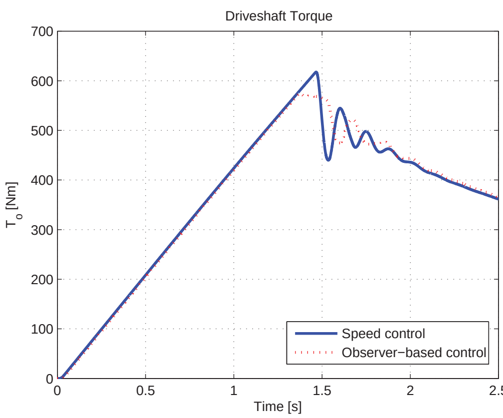

Fig. 7. Simulation results for the output torque: speed control and observer-based torsion control for clutch engagement during a vehicle launch

of external disturbances. Hence, the model based closed-loop control strategy is needed to improve the engagement performance.

It should be noted that the shaft torsion controller is applied only within specific range of relative speed.

IV. Shaft torque observer

A. Concept

In the preceding section, the problem on the torque control is raised. The shaft torsion control law (26) requires the state that cannot be measured in real situation, i.e., the drive shaft torque. This problem is solved by designing an observer. The proposed nonlinear observer can be divided into two cases which are a stick phase observer and a slip phase one. Since these observers will be inserted into the control input as the feedforward term, the oscillating signal from observers after the clutch engagement may deteriorate the performance of the torsion control. Therefore, some considerations on the observer gains should be needed to make a smoothed shaft torque estimate in the sense of average.

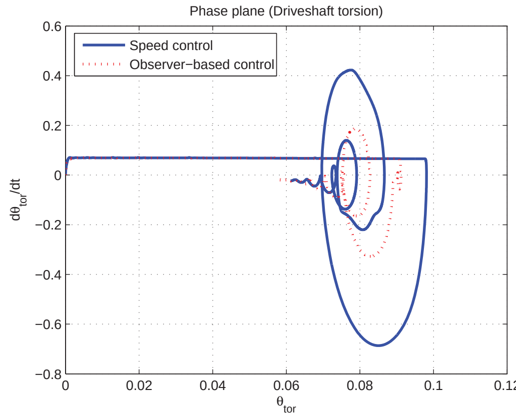

Fig. 8. Simulation results for the phase plane of shaft torsion: speed control and observer-based torsion control for clutch engagement during a vehicle launch

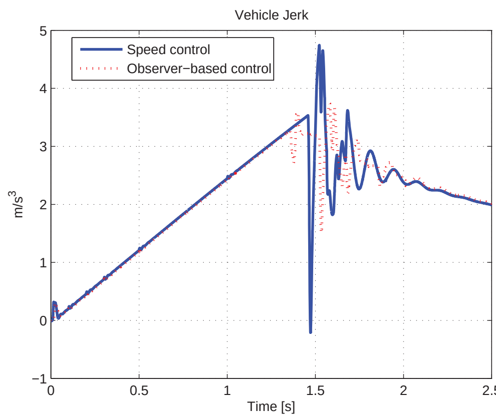

Fig. 9. Simulation results for vehicle jerk: speed control and observer-based torsion control for clutch engagement during a vehicle launch

B. Observer Design

The shaft torque observers for stick phase and the slip phase are formulated in equations (27) and (29), respectively.

Clutch Stick:

$$\dot{\hat{T}}_o = \dot{T}_{ostick} + L_{stick}(T_{omeas} - \hat{T}_o) \tag{27}$$

where

$$\dot{T}_{ostick} = a_{stick}[i_t(T_e - b_t\omega_e - J_e\dot{\omega}_e) - T_{ostick}] \tag{28}$$

Clutch Slip:

$$\dot{\hat{T}}_o = \dot{T}_{oslip} + L_{slip}(T_{omeas} - \hat{T}_o) \tag{29}$$

where

$$\dot{T}_{oslip} = a_{slip}[i_t(\mu R_c F_n - b_t\omega_t - J_{c2}\dot{\omega}_t) - T_{oslip}] \tag{30}$$

In the observer structure, the known part described in equations (27) and (29) can be obtained from a driveline dynamic model with a first-order phase lag. The lag that can be adjusted by tuning a_{stick} and a_{slip} plays a role of

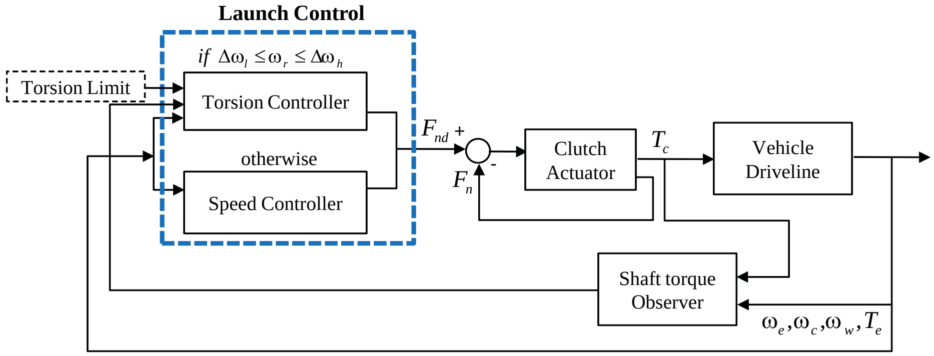

Fig. 10. Structure of the speed-torsion controller with shaft torque observer

filtering out outshaft pulsating torque just after the clutch engagement. Such a scheme is employed since the resulting signal of the observers are fed into the torsion controller. In addition, the observer gains L_{stick} and L_{slip} can also be chosen to adjust the weighting between the speed measurements and the driveline model. As a result, there are two design parameters to estimate the shaft torque. In the slip phase, the clutch torque generated by the normal force is considered as positive value assuming the sliding speed don't change sign. The observer output is linked to the feedfoward control information in (26).

As a result, the non-measurable state T_o in the control law (26) can be replaced by $\hat{T}_o$. The final simulation results of speed and shaft torsion combined control are as shown in Fig. 6 - 9. Fig. 6 shows that the observer-based combined control has better performance compared with speed control shown in Fig. 3. Speed synchronization is perfomed more smoothly than the case of Fig. 3. The outshaft torque oscillation is reduced as shown in Fig. 7, 8 and the jerk performance is improved as shown in Fig. 9 The overall control architecture is as schematically described in Fig. 10. In Fig. 10, $\Delta\omega_l$ and $\Delta\omega_h$ denote the lower and upper bounds for the relative speed, respectively.

V. CONCLUSIONS

To verify the feasibility of the controllers, a drivetrain dynamic model is developed. It considers the engine, transmission, propeller shaft, final differential, and external load. The speed controller is designed to perform a basic operation for the clutch engagement. Also, the torsion control law with the estimation of the shaft torque is applied when the relative speed between the engine and the clutch is under a certain threshold. The proposed overall strategy uses only the clutch normal force as a control input. The simulation results show that performance of the speed controller can be improved significantly when it is combined with the observer based torsion control strategy.

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