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Gear Shift Control of Dual Clutch Transmissions with a Torque Rate Limitation Trajectory

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***Abstract*— Gearshift control is one of the main functions of automotive transmissions. The proposed control utilizes the functionality of Dual Clutch Transmissions (DCTs) that two control inputs are available incorporating each other. During gearshift operation, each clutch is controlled uni-directionally to prevent undesirable effects such as dead zone nonlinearity, saturation, and etc. Even though they have such a constraint, the desired clutch profile is tracked well via cross-coupling effect of two actuators. In addition, engine torque is reduced based upon the predetermined target engine speed. A simple proportional-integrative (PI) controller is implemented for a feedback control of the proposed gear shifting management. Simulation results show that gear shifting is performed without excessive oscillations of the shaft torque.**

I. INTRODUCTION

Recently, advanced transmission technologies are introduced to improve fuel efficiency. Since planetary type Automatic Transmissions (ATs) generally require torque converters to connect the engine with the transmission with few exceptions, it is inevitable that there are energy losses in spite of smooth gear shifting. Automated Manual Transmissions (AMTs) have a servo-controlled clutch and gear actuators along with mechanism of Manual Transmissions (MTs). However, AMTs have a limitation of shift quality due to severe torque interruption [1], [2], [3], [4].

On the other hand, Dual Clutch Transmissions (DCTs) combine high efficiency of MTs with convenience of automatic transmissions by using two clutches for engagement and release, simultaneously. Thus, shift quality and driving comfort can be significantly improved even without lack of torque transmission. In spite of such advantages, controllability plays a crucial role in determining overall performance of DCTs. There have been some researches on the modeling and analysis of dynamic characteristics of DCTs [5], [6], [7], [8]. The methods suggested in several researches are open-loop based control and PID controller with experimental calibration [9], [10]. Since DCT vehicles are not equipped with a torque converter, it is difficult to achieve sophisticated torque control in the real environment.

Thus, the upper level controller for gear shifting management, that generates a clutch profile accurately to the lower controller for actuator control, should provide a desired control input to the electric or hydraulic actuators. Moreover, gear shifting should be performed in a short period of time to prevent the vehicle from having torque interruption. Compared with a launch control of standing vehicles, it requires very fast engagement and release of a clutch in order to avoid torque interruption so that actuators are needed to have quick response.

In this paper, the gearshift control management system will be presented to utilize the unique characteristic of dual clutch systems. Section II presents the driveline model equipped with the DCT. In Section III, the open-loop based control will be introduced for comparison with the proposed method. In IV, the gearshift control strategy using a torque rate limitation is proposed based on the convexity of control inputs. The engine controller is also introduced to reduce the engine torque. Simulation results and some concluding remarks are drawn in Section V and VI, respectively.

II. DRIVELINE MODEL

In a gear shift operation, a driveline model with the dual clutch transmission is needed to describe dynamic behavior of the system because DCTs do not have a torque converter that dampens shift impact compared with ATs. The schematic of the driveline model is shown in Fig. 1.

The engine crank shaft and the clutch dynamics are given by

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

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

where, J_e and J_d are the moment of inertia of the engine and the mass flywheel, respectively, T_e the engine torque calculated from a steady-state map with respect to the engine speed ω_e, and the throttle angle α, T_{c1} the clutch 1 torque connected with the solid shaft, and T_{c2} the clutch 2 torque connected with the hollow shaft. T_d represents a fly wheel compliance given by $T_d = k_d(\theta_e - \theta_d) + b_d(\omega_e - \omega_d)$ where k_d and b_d denote stiffness and viscous coefficients. By assuming a Coulomb friction model, the clutch torque is represented as

$$T_{ck} = \mu_{ck} R_{ck} F_{nk} sign(\omega_d - \omega_{ck}), \quad for\ k = 1,2 \quad (3)$$

where, μ_c is the friction coefficient of the clutch surface, R_c the effective radius of the clutch disk, and F_n the clutch normal force. The dynamics of the clutch is represented as

$$J'_{c1} \dot{\omega}_{c1} = T_{c1} + \frac{i_{t2}}{i_{t1}} T_{c2} - \frac{T_s}{i_{t1} i_{f1}} \quad (4)$$

$$J'_{c2} \dot{\omega}_{c2} = \frac{i_{t1}}{i_{t2}} T_{c1} + T_{c2} - \frac{T_s}{i_{t2} i_{f2}} \quad (5)$$

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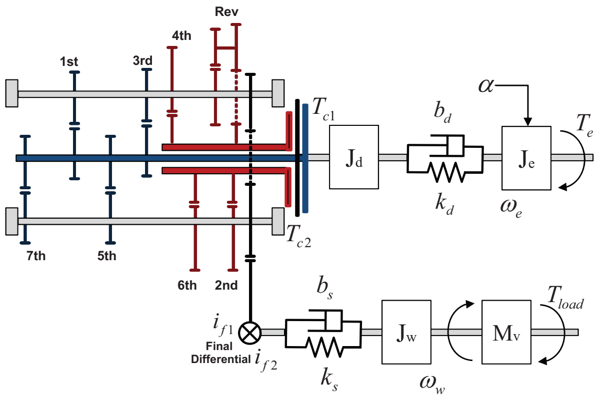

Fig. 1. Vehicle driveline dynamics with a DCT

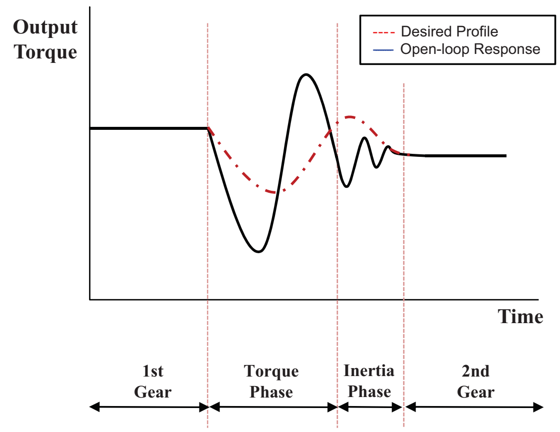

Fig. 2. Schematic of the control concept for a DCT

where J'_{c1} and J'_{c2} are the equivalent moment of inertia of the solid and hollow shafts, which can be calculated as

$$J'_{c1} = J_{c1} + J_{t1} + \left(\frac{i_{t2} i_{f2}}{i_{t1} i_{f1}}\right)^2 (J_{c2} + J_{t2})$$

$$J'_{c2} = J_{c2} + J_{t2} + \left(\frac{i_{t1} i_{f1}}{i_{t2} i_{f2}}\right)^2 (J_{c1} + J_{t1}).$$

Here, i_t and i_f denote the transmission gear ratio and the final differential ratio, and J_c and J_t stand for moment of inertia of the clutch and transmission, respectively. The drive shaft torque T_s is modeled as

$$T_s = k_s\left(\frac{\theta_t}{i_t i_f} - \theta_w\right) + b_s\left(\frac{\omega_t}{i_t i_f} - \omega_w\right) \tag{6}$$

where, the variable k_s denotes the stiffness, b_s the damping coefficient of the output shaft, θ_w the wheel angle, and ω_w the wheel speed, respectively. ω_t denotes the transmission output speed that is set to the corresponding clutch speed depending on the gear ratio (i.e. $\omega_t = \omega_{c1}$ for odd number gearbox, and $\omega_t = \omega_{c2}$ for even gearbox).

It is reasonable to assume that the tire is considered as a rolling element without slip. In addition, the distribution of traction force on the axles is also neglected. These assumptions make the relationship between the wheel speed ω_w and vehicle speed v_v simple as $v_v = r_w \omega_w$. Subsequently, the wheel dynamics is

$$J_w \dot{\omega}_w = T_w - T_{trac} \tag{7}$$

where, J_w is the wheel inertia. T_{trac} the traction torque applied on the tire produced by the friction between the tire and road surface, i.e. $T_{trac} = F_{trac} r_w$, and the driving torque T_w the output shaft torque multiplied by the final drive ratio i_f, i.e. $T_w = i_f T_f$, and T_f the torque applied to the final differential. Note that T_w is equal to the output shaft torque T_s under the assumption that the final differential is stiff. In turn, overall vehicle is considered as a point mass m_v. Accordingly, vehicle longitudinal dynamics (8) with the rolling force F_{roll} and the aerodynamic force F_{aero} is described as

$$M_v \dot{v}_v = F_{trac} - F_{roll} - F_{aero} - m_v g \sin\theta_r \tag{8}$$

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

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 vehicle, v_x the vehicle speed, and r_w is 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. Combining equation (7) and (8) yields

$$J_v \dot{\omega}_w = T_s + T_{load} \tag{9}$$

where the vehicle inertia $J_v = J_w + m_v {r_w}^2$ is obtained by adding the wheel inertias to the equivalent inertia of the vehicle mass. The external load torque T_{load} in above equation is given by

$$T_{load} = \left(M_v g \sin\theta_r + K_r M_v g \cos\theta_r + \frac{1}{2}\rho C_d A_F {v_x}^2\right) r_w. \tag{10}$$

III. Problem Statement

A. Shift Sequence

The gearshift operation particularly in the case of upshift can be divided into two phases in terms of transmitted torque shown in Fig. 2: one is the torque phase which is from the shift command to the lowest value of the output torque. The other is the inertia phase where the speed is synchronized with some oscillation due to variation of the kinetic energy from newly connected gear set. In this paper, two clutches are classified into the oncoming and the off-going clutch corresponding to their own functions. The oncoming clutch and the off-going clutch only involve engagement and release operation, respectively.

In particular, the 1-2 upshift case is considered for the feasibility study of the proposed method. In Fig. 3, the solid line denotes the engine speed, the upper dotted line $\omega_{c1a} - \omega_{c1c}$ is the speed of the oncoming clutch, and the lower dotted line $\omega_{c2a} - \omega_{c2c}$ is the speed of the off-going clutch. t_c means the shift command point, t_t the end point of the torque phase, t_i

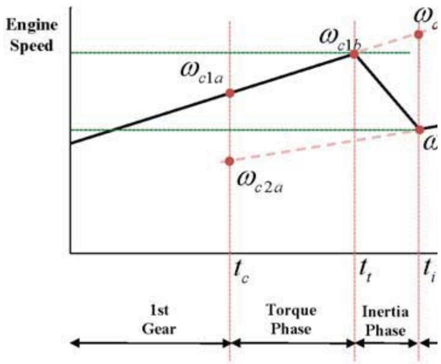

Fig. 3. Engine and clutch speed characteristics

the end point of inertia phase, t_s the syncl
with newly engaged clutch. In $t \in [t_c, t_t]$, the
torque is ramped up for engagement, whi
clutch torque is gradually reduced to zero.
oncoming clutch torque reaches the desired
some slippage. To minimize the shifting tim
oncoming clutch torque should be large in th
excessive torque change may lead to the drive
In $t \in [t_i, t_s]$, the oncoming clutch slip may occur and the drive shaft may also oscillate. After the synchronization point t_s, the gear shifting operation is completed.

B. Open-loop Control

Since it is difficult to determine the magnitude of the clutch torque and the synchronization points of dual clutches, there exist open-loop gear shifting methods depending heavily on experimental calibrations. In a nominal situation, gear shifting performance could be accepted without any undesirable effect. However, there will be cases in which the unexpected behavior occurs associated with uncertainties, and so, the driver and passengers may feel discomfort.

For comparison with the subsequent control strategy, the open-loop based control with ramp-type normal force trajectories as shown in Fig. 4 is simply designed as one of conventional ways. After the oncoming clutch is engaged, some oscillations are observed in the clutch speed. They may lead to the vehicle jerk that is undesirable. Although such a profile can be further improved from a series of test results, it has a limitation due to the lack of feedback information of both clutches in the controller. If the feedback controller employs real-time information as well as prescribed torque trajectories, control performance will be improved.

IV. Gear Shifting Control for DCTs

A. Basic Concept and Motivation

In this section, a torque rate limitation control strategy is suggested. The key of the idea is that the controlled clutch is only actuated in one direction. For example, the oncoming clutch is only controlled such that it has the normal force to be ramped up or held. On the other hand, the off-going clutch is only actuated such that it has the

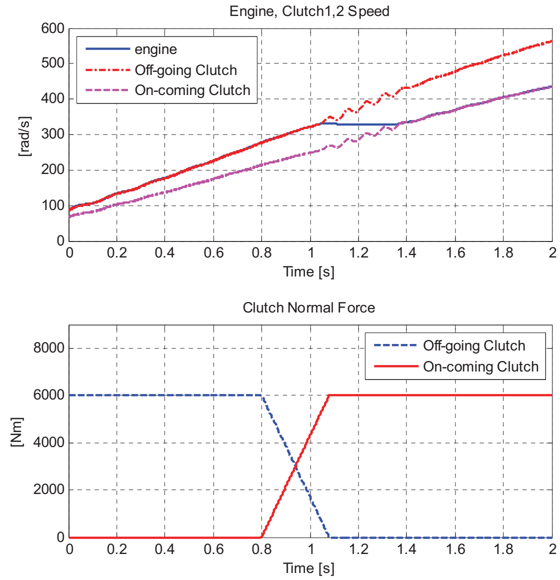

Fig. 4. Open-loop based control for gear shifting (1-2 upshift)

normal force to be decreased or held. Thus, it does not allow the clutch to put it back into the direction of the initial position once the clutch is started. It is possible for the clutch to complete the engagement more quickly while avoiding some actuator nonlinearities such as dead zone, saturation, and etc. Here, one issue may arise: *How does one make a feedback signal including negative rate control input?* This question can be answered by using the physical peculiarity of DCTs. Since DCTs can employ two clutches as control inputs, it is extremely useful to complement each other during clutch control. The feed-forward control is given as a baseline controller for gearshift operation. It is based on two smoothed profiles for engagement of the oncoming clutch and disengagement of the off-going clutch, respectively. The details are illustrated in the following.

1) Oncoming clutch: When the decreasing normal force trajectory is applied on the oncoming clutch, the output of the controlled input is limited as hold status since the normal force rate is non-negatively constrained. At this moment, if the off-going clutch is also to be held, the oncoming clutch can obtain the effect that its force is decreasing.
2) Off-coming clutch: When the increasing normal force trajectory is applied on the off-going clutch, it remains held without positive rate. At this moment, if the on-going clutch is also to be held, this is equivalent to negative rate force of the off-coming clutch.

Although this control strategy restricts the negative rate of the normal force of an on-going clutch, it can be realized incorporating an off-going clutch as another actuator. As a result, clutch is engaged as fast as possible without clutch tie-up and backward power circulation.

B. Control Strategy

Let $\mathbf{x}(t,\mathbf{x}_0,\mathbf{u}) \in \mathbb{R}^n$ denote the trajectory o dynamic model of a DCT driveline system corre an initial state $\mathbf{x}_0 \in \mathbb{R}^n$ and a control input $\mathbf{u} = [u$ $\mathbb{R}^2$. u_{on} and u_{off} denote the oncoming and th clutch torque, respectively. In our synthesis, th assumptions can be made on the system.

A1) The vector valued function $\mathbf{u}$ of the c is continuously differentiable and satisf

$$\mathbf{u}_{min}(\mathbf{x},\dot{\mathbf{x}}) \leq \mathbf{u}(\mathbf{x},\dot{\mathbf{x}}) \leq \mathbf{u}_{max}(\mathbf{x},$$

A2) The time derivative of $\mathbf{u}(t)$ is bounded

$$|\dot{\mathbf{u}}| \leq f(\mathbf{x},\dot{\mathbf{x}}) \qquad \forall t \geq t_c$$

The control strategy described in the precedin is defined in the discrete time domain as follow

$$u_{on}[(i+1)\Delta T] = \begin{cases} u_{on}(i\Delta T) & \text{if } \dot{u}_{off} > 0,\ \forall t \in [t_c, t_s] \\ u_{on,ci}(i\Delta T) & \text{if } \dot{u}_{off} \leq 0,\ \forall t \in [t_c, t_i] \\ u_{on,is}(i\Delta T) & \text{if } \dot{u}_{off} \leq 0,\ \forall t \in [t_i, t_s] \end{cases} \tag{11}$$

$$u_{on,ci}(i\Delta T) = u_{onFF}(i\Delta T)$$
$$u_{on,is}(i\Delta T) = u_{on,FF}(i\Delta T) + u_{on,FB}(i\Delta T)$$
$$u_{on,FB}(i\Delta T) = k_p e_{on}(i\Delta T) + \int_{i\Delta T}^{(i+1)\Delta T} k_I e_{on}(i\Delta T)\,dt$$

$$u_{off}[(i+1)\Delta T] = \begin{cases} u_{off}(i\Delta T) & \text{if } \dot{u}_{on} < 0,\ \forall t \in [t_c, t_s] \\ u_{off,FF}(i\Delta T) & \text{if } \dot{u}_{on} \geq 0,\ \forall t \in [t_c, t_s] \end{cases} \tag{12}$$

where, $u_{on,FF}$ and $u_{on,FB}$ are the feed-forward and feedback control input of the oncoming clutch, $u_{off,FF}$ the feed-forward control input of the off-going clutch, i the sampling instant, and ΔT the sampling period, respectively. The torque tracking error of the oncoming clutch is defined as $e_{on} = T_{cd} - T_c$.

The suggested feedback controller is a simple proportional-integrative (PI) type controller, and the feedforward controller is composed of predetermined torque profiles and low pass filters. In order to make the control input differentiable, First order phase lags are applied to (11) and (12) as

$$\dot{u}_{f,on} + a_{on} u_{f,on} = a_{on} u_{on} \tag{13}$$

$$\dot{u}_{f,off} + a_{off} u_{f,off} = a_{off} u_{off} \tag{14}$$

where, a_{on} and a_{off} are filter design parameters, and $u_{f,on}$ and $u_{f,off}$ filtered control inputs.

To analysis the characteristic of the proposed method, the following property is required.

Lemma 1: Let X be a convex set in $\mathbb{R}$. Suppose that the function f is differentiable, then its gradient $\triangledown f$ exists at each point $x,\ y \in X$. Then, f is convex such that

$$f(y) \geq f(x) + \triangledown f(x)^T (y - x) \tag{15}$$

for all $x,\ y \in X$.

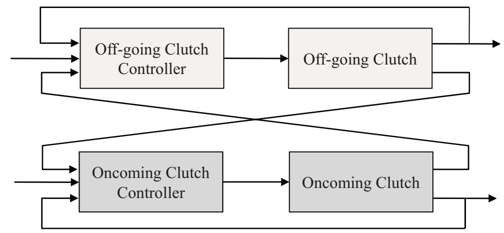

Fig. 5. Schematic of the control concept for DCTs

This result is the first order condition of a convex function, which is easily proved by convexity of a function f. The detailed proof of Lemma 1 can be found in [11]. Similarly, it follows that the corresponding property for a concave function g can be made as

$$g(y) \leq g(x) + \triangledown g(x)^T (y - x) \tag{16}$$

for all $x,\ y \in X$.

The following theorem concerns the feasibility of proposed gear shifting management system.

Theorem 1: In gear shift operations, the control laws (11)-(12) make the dynamic system of a DCT driveline system $\mathbf{x}$ to be constrained as:

$$\mathbf{x}(t,\mathbf{x}_0,\mathbf{u}) \in \mathcal{A}, \quad \forall t \in [t_c, t_s] \tag{17}$$

where

$$\mathbf{x}_0 = [x_1(0), x_2(0), \ldots, x_n(0)]^T$$
$$\mathbf{u}_{min}(t) \leq \mathbf{u}(t) \leq \mathbf{u}_{max}(t), \quad \forall t \geq 0$$
$$\mathcal{A} \triangleq \{\mathbf{x} \in \mathbb{R}^n \mid \dot{u}_{on} \geq 0 \ and\ \dot{u}_{off} \leq 0\}.$$

Proof: Given control inputs (11) and (12), the resulting trajectories of $u_{on}(t)$ and $u_{off}(t)$ satisfies the following properties that can be easily obtained from Lemma 1 along with the assumption A1 and A2.

$$u_{on}(t_s) \geq u_{on}(t_i) + u'_{on}(t_i)(t_s - t_i). \tag{18}$$
$$u_{off}(t_s) \leq u_{off}(t_i) + u'_{off}(t_i)(t_s - t_i). \tag{19}$$

Hence, u_{on} is to be a convex function, and u_{off} is to be a concave function [11]. To illustrate more details of this control scheme, the driveline model described in Section II can be adopted. Since the solid shaft clutch torque T_{c1} and the hollow shaft clutch torque T_{c2} are connected by kinematic constraints (i.e. gearbox), both clutch shaft models (4) and (5) are combined into

$$T_J = \psi_1 T_{c1} + \psi_2 T_{c2} \tag{20}$$

where

$$T_J \triangleq -\frac{i_{t1} i_{f1}}{i_{t2} i_{f2}} J'_{c1} \dot{\omega}_{c1} + J'_{c2} \dot{\omega}_{c2},$$
$$\psi_1 \triangleq \frac{i_{t1}}{i_{t2}} \left(1 - \frac{i_{f1}}{i_{f2}}\right), \quad \psi_2 \triangleq \left(1 - \frac{i_{f1}}{i_{f2}}\right),$$

T_J is the torque that represent the moment of inertia of the solid and the hollow shaft, and ψ_1 and ψ_2 the function of the gear ratio. Since the control inputs are convex and concave functions, (20) can be rewritten by

$$T_J = \alpha u_{on} + (1-\alpha)u_{off}, \quad \alpha \in [0,1]. \tag{21}$$

Assuming 1-2 upshift case, u_{on} is set to T_{c2}, and u_{off} is set to T_{c1} Since two intermediate shafts for clutch 1 and 2 are coupled with each other, T_J is bounded as some value with respect to the inertia and the shaft acceleration.

Therefore, u_{on} and u_{off} are constrained as a nondecreasing function and a nonincreasing function, respectively. Also, this makes a system to be constrained into the admissible set $\mathcal{A}$. ■

In this formulation, the control input $\mathbf{u}(t)$ is bounded and the admissible set $\mathcal{A}$ is defined to represent the constrained state-space for gear shifting management system. The convex combination of two control inputs can be thought of as weighted average transmission torque to a vehicle during gearshift.

In other words, the constraints of the off-going clutch can be applied in compensation for the oncoming clutch, and vice versa. Therefore, the transmitted torque quantity on the shaft remains within a certain amount of torque. These are mainly from the unique feature of dual clutch transmissions that have two clutches as multiple inputs. A gearshift operation is simultaneously performed with the engagement of the on-coming clutch and the disengagement of the off-going clutch. Thus, this strategy aims at maximizing such an advantage of DCTs in conjunction with the control formulation. Fig. 5 shows the schematic of the control concept of the proposed strategy.

C. Engine Control

During gearshift process, there is abrupt change of vehicle driveline speed so that the drive shaft torque is also dropped unexpectedly (i.e. torque phase). This is mainly due to interrupting transferred torque in a moment. When the gear shifting is completed, the drive shaft torque is recovered to its desired value corresponding to the gear ratio of newly engaged speed (i.e. inertia phase). Generally, some overshoot can occur because of variation in transferring torque in the inertia phase. It may lead to a vehicle jerk, which is recognized as drivers discomfort.

To solve this problem, engine torque reduction must be considered incorporating with gear shifting control considered in the preceding subsection. For example, when the clutch is controlled in order to achieve an upshift process, the engine torque controller defines the target engine speed at which the clutch engagement is completed. If the target engine speed is too low compared with the nominal value, it is possible that the engine is stalled. On the other hand, too high target engine speed may deteriorate the shift quality. Therefore, the target engine speed for engine torque reduction must be chosen appropriately.

The target engine speed can be determined by two clutch speed measurements and the desired gear shifting duration. Since the gear shifting is performed in a short period of time, it is assumed that the engine speed changes linearly with time. In 1-2 upshift case, the target speed is defined as:

$$\omega_{c2d}(t) = \omega_{c2a}(t) + \frac{\partial \omega_{c1a}(t)}{\partial t}\left(\frac{i_{t2}}{i_{t1}}\right)(t_i - t_c) \tag{22}$$

where, the time derivative of the $\omega_{c1a}(t)$ is numerically obtained from the measurement signal in $t \in [t_c, t_i]$ to avoid noise problem. Accordingly, the engine torque controller is simply designed as

$$T_e = -K_{Pe}\left(\omega_{c2}(t) - \omega_{c2d}(t)\right) - \int_{t_c}^{t_i} K_{Ie}\left(\omega_{c2}(\tau) - \omega_{c2d}(\tau)\right) dt \tag{23}$$

where, K_{Pe} and K_{Ie} denote the feedback gains. Once it is defined appropriately, the engine torque is also controlled indirectly by feedback of a speed tracking error. Note that the desired engine speed is taken as the maximum value of the prescribed engine speed and the idle speed. The block diagram of this scheme is shown in Fig. 6. This approach easily provides an effective way for active damping of driveline oscillation.

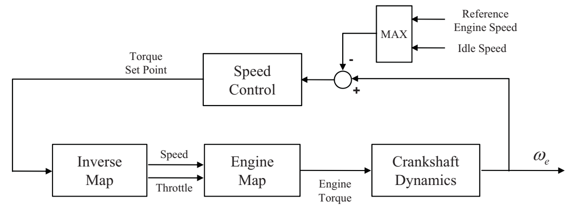

Fig. 6. Block diagram of the engine speed/torque control

V. Simulation Results

In this section, some simulation results are presented to illustrate the proposed gear shifting management strategy. Fig. 7 shows the speed trajectory of the engine, clutch 1, and clutch 2 during 1-2 upshift case. Speed synchronization between two clutches is performed from 1.0 to 1.5 sec. As shown in Fig. 8, the engine torque is reduced right before the engagement of the second clutch in order to reduce shifting shocks. Fig. 9 shows that oscillation of output shaft torque is reduced during the period after the second clutch is engaged.

According to Fig. 10, it is easily verified that the proposed method has the advantage presented in the control strategy. The clutch normal force of the oncoming clutch is composed of feedback and feedforward control while that of the off-going clutch is derived by a feedforward control only. At 1.1 sec, it can be observed that the off-going clutch torque is held while the oncoming clutch torque is commanded negative torque rate by the baseline feedback controller. Thus, the proposed control method can prevent undesirable actuator effect like dead-zone nonlinearity.

VI. Conclusions

In this paper, a novel control strategy for gear shifting of DCT vehicles is proposed to utilize the unique characteristics of dual clutch systems. The resulting trajectories of the on-coming and the off-going clutch torques are formulated as a

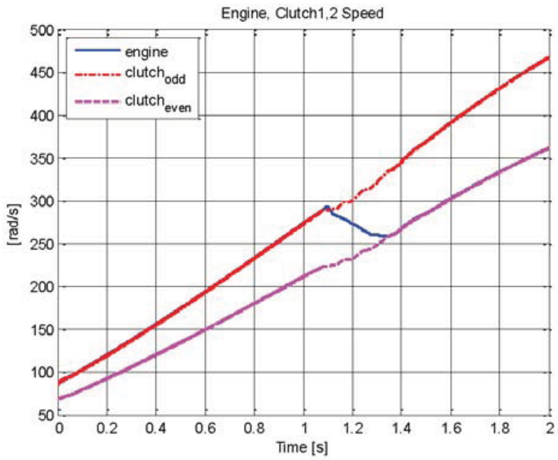

Fig. 7. Engine and clutch speeds during 1-2 upshift

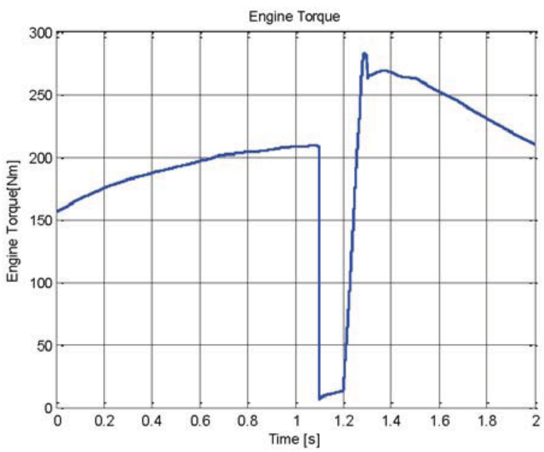

Fig. 8. Engine torque in 1-2 upshift

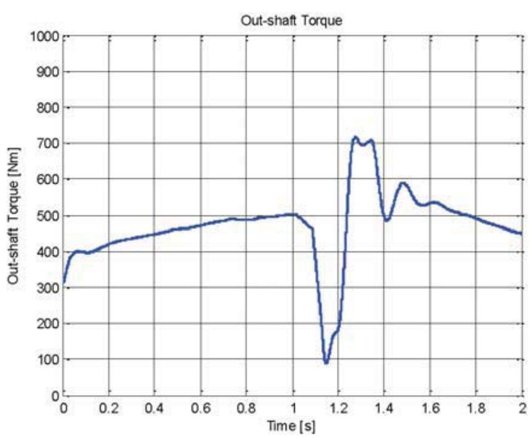

Fig. 9. Output shaft torque in 1-2 upshift

nondecreasing and a nonincreasing function, respectively. As a result, the control inputs are made as convex combination

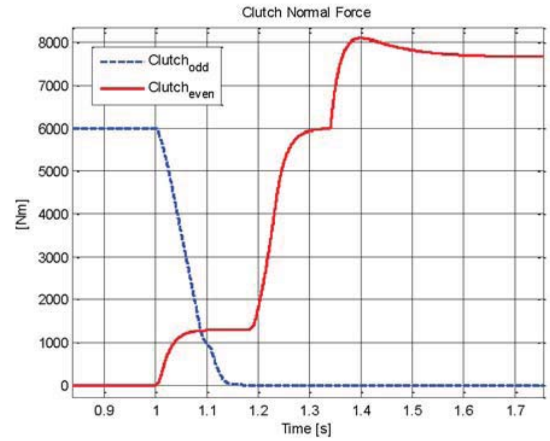

Fig. 10. Clutch torque in 1-2 upshift

with a finite torque split. In order to analyze and validate the proposed control scheme, simulations are performed for a vehicle model equipped with a DCT. The results show that the proposed controller ensures good drivability of a vehicle in spite of fast gear shifting action. In future works, robustness issues and several driving modes should be taken into account.

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