Gearshift control for dry dual-clutch transmissions

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Abstract: - In this paper a smooth control algorithm for gear shifting is proposed for improving longitudinal dynamic performance during dry dual-clutch engagements while shifting up or down takes place based on measurements of engine speed and clutch speed, and on estimation of the dual-clutch engaging torque. Simulation effort is made by Matlab/Simulink for transient responses of the overall propulsion system with the dry dual-clutch transmission (DDCT) in consideration of twin clutch engagement rules for gear shifting. As a numerical example, the present control algorithm is used for the DDCT equipped vehicle to simulate overlapping shifting behaviors. By the numerical it is shown that the DDCT vehicle possesses of much better longitudinal dynamic performance in shifting by imposing proper control of engine and the dual-clutch.

Key-Words: - Dry dual-clutch transmission; Engaging force; Smooth control algorithm; Clutch control; Engine control; Shifting behaviors

1 Introduction

Dual-clutch Transmissions (DCT) for passenger vehicles are currently attracting the attention of automotive industries, in which a dual-clutch system between the engine and the gear transmission is used to alternate torque demands from one clutch to the other clutch without power interruption whenever the gear shifts take place during acceleration or deceleration [1]. It is well recognized that the DCT remain simple structure and high efficiency of manual transmissions and realize smooth shifting operations almost the same as conventional automatic transmissions (AT) [2, 3].

There are two types of dual clutches, wet and dry, adopted in the DCT system. In comparison with the dry dual clutches, the wet dual clutches in combination with a fully hydraulic control system for actuation and cooling is generally known to be more expensive [4]. Furthermore, the pump loss often leads to higher fuel consumption. For the reason, the dry DCT in which the dry dual-clutches are actuated by an electromechanical system of low cost in manufacturing may be an ideal option [5].

For a high shift quality, an effective control strategy of the transmission is required to regulate accurately the dual-clutch engagement forces acting on the friction discs. It is clear that the compromise treatment between the intensity and time of the engagement forces is accounted for in the control strategy. Recent studies have been focused on the modeling and control of automated manual transmission (AMT) [6], automatic transmission (AT) [7, 8] and continuous variable transmission (CVT) [9, 10]. These research works are intended to improve numerical modeling for accurate predictions of dynamic responses of the systems before development of the transmission prototype and optimization of the control system. As conventional automatic transmission technologies are well developed for carmakers, the trade-off technology of dual-clutch transmissions, especially the smooth control algorithm for dry DCT, is still at the emerging stage of development. The modeling and simulation of DCT are developed [11-13]. Among the control approaches for DCT gear shifting seen in the literature, some are unoptimizable strategies without the precise control of the dual-clutch and engine [14], whereas the other allows the tracking of wet DCT output torque trajectories [15]. Although former approaches cannot guarantee the high shift quality, the latter is not easy measuring the output torque directly at output axle.

The present research is intended to develop a shift control algorithm for optimization of the gearshift quality by reducing efficiency loss due to friction during the transition of sliding in the dry dual-clutch sliding. The presented controller for gear shifting is designed with feedback loops based on measurements of clutch and engine speeds, and on estimation of dual-clutch engaging torques. The shift control algorithm can regulate automatically the engaging forces of dual-clutch and engine throttle angle. Using the dual-clutch engaging forces and the engine throttle angle as control signals, the powertrain system model created in Matlab/Simulink can be used for numerical simulations of vehicle transmission performance.

2 The structure of the dry dual-clutch system

The dry dual-clutch system in the DCT is shown in Fig. 1, working modes of which are ordinarily open and engaged by pushing forward. In shift process of the DCT the overlapping power transmission is involved on one hand with the engagement of the oncoming clutch and on the other hand with the release of the offgoing clutch. Thus the twin clutches in DCT provide the powertrain with two routes from the engine to driving wheels. Because of some overlapping the twin clutches, no interruption of the power transmission in the driveline during shift turns out. Two independent electromechanical actuators are employed to operate meanwhile the oncoming and the offgoing clutches in the system. The prestressed spring acts on the other side of the lever. Between the two there is a movable support, whose longitudinal motion is generated via a ball screw driven by an electric motor. The pressure force of the clutch engagement may be determined by adjustment of the position of the moveable support from the prestressed spring and the lever ratio.

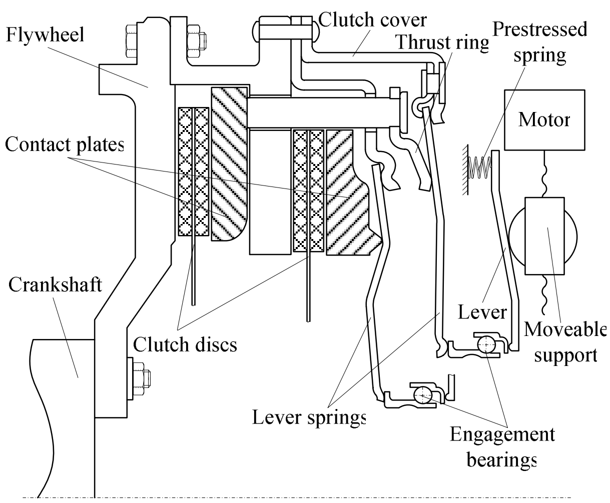


Fig. 1 Configuration of a dry dual-clutch system

Based on the clutch geometry and friction characteristics, the torque capacity T_{CL} at an engaging force F_{n} on the friction disc may be expressed as follows

$$T_{\mathrm{CL}} = \begin{cases} 0 & \text{idle stroke} \\ n\mu F_{\mathrm{n}} r_{\mathrm{f}}\ \mathrm{sgn}(\omega_{\mathrm{in}} - \omega_{\mathrm{c}}) & \text{slipping stroke} \end{cases} \quad (1)$$

where n is the number of engaging surfaces, μ is the friction coefficient, ω_{in} and ω_{c} are the angular velocities of the flywheel and the friction disc, and r_{f} the friction radius of the following form as $r_{\mathrm{f}} = \frac{2}{3}\left(\frac{r_{\mathrm{b}}^3 - r_{\mathrm{a}}^3}{r_{\mathrm{b}}^2 - r_{\mathrm{a}}^2}\right)$ in which r_{b} and r_{a} outside and inside radii of the friction disc, respectively.

When the dual clutch is engaged, the engine is rigidly coupled to the driveline. As a result, consequently, the torque transmitted through the clutch is fixed in a range given below

$$-n\mu_0 F_{\mathrm{n}} r_{\mathrm{f}} \le T_{\mathrm{CL}} \le n\mu_0 F_{\mathrm{n}} r_{\mathrm{f}} \quad (2)$$

where μ_0 is the static friction coefficient of the clutch.

3 The dynamic model of powertrain

The dynamic model of the powertrain with dry DCT is presented in Fig. 2, which is comprised of an engine, a dry dual-clutch, a gearbox and a final reducer, as well as a wheel and vehicle mass.

As shown in Fig. 2, CL1 is refered to all the 1st, 3rd and 5th odd gears and CL2 is related all the 2nd, 4th, and 6th even gears, being both connected to the input shaft. Because gears of the next speed can be pre-engaged by synchronizer while the vehicle is driven at a current speed, the synchronization of even and odd gears has a very limited effect on shift quality. When a gear shift signal is initiated, the offgoing clutch is released and the oncoming clutch simultaneously engaged, which allows torque transfer without power interrupted during a gearshift.

Forces of engagement of the twin clutches are actually adjusted by moving the movable support with an electromotor actuator. As shown in Fig. 2, all gears and shafts in the gearbox are simplified to be moment inertias and torsional spring-damper assemblies, and two clutches are reduced to be friction elements with moments of inertia. The simulation model may be developed thereafter in the Matlab/Simulink for numerical predictions of the axle torque and speed responses.

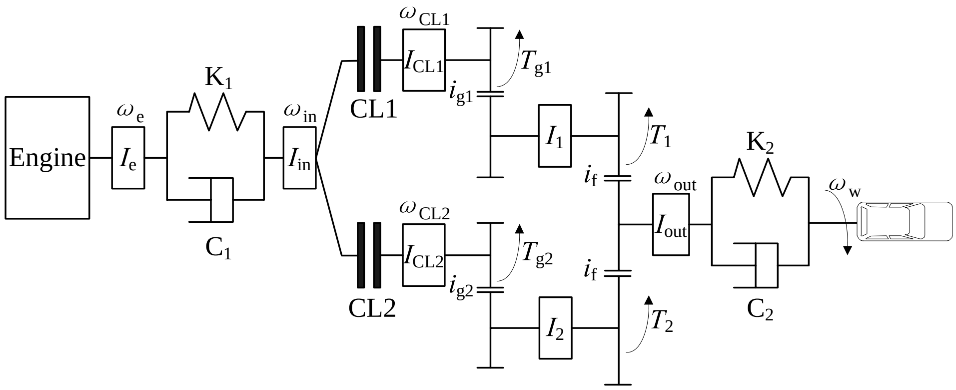


Fig. 2 The dynamic model of the powertrain with dry DCT

3.1 Engine

With change of the throttle angle, the output of the engine power can be characterized by the torque as a function of the engine speed. That means the engine may be mathematically modeled in terms of the throttle angle α_e and the engine angular velocity ω_e as follows

$$T_e = f(\alpha_e, \omega_e) \tag{3}$$

$$I_e \dot{\omega}_e = T_e - T_{in} \tag{4}$$

where T_e and T_{in} are torques of the engine and the flywheel, and I_e is the moment of inertia of the engine crankshaft.

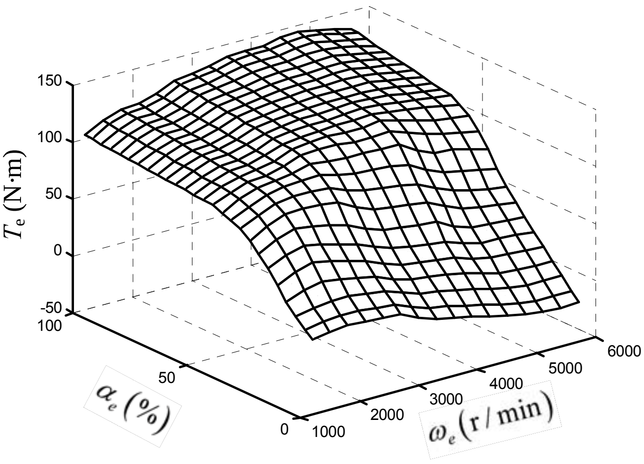


Fig. 3 Engine torque map

The engine torque modelled by equation (3) is shown in Fig. 3, with the throttle angle from 0 to 100 %.

3.2 Dual-clutch transmission

The equilibrium equation of motion in torsional vibrations for the input shaft of the dry dual-clutch may be written in the following form as

$$T_{in} = K_1(\theta_e - \theta_{in}) + C_1(\omega_e - \omega_{in}) \tag{5}$$

where θ_e and θ_{in} are the rotational angles of the engine crankshaft and the flywheel, and K_1 and C_1 are torsional rigidity and damping coefficient of the input shaft.

The torque transmitted through the dry dual-clutch can be expressed as

$$I_{in}\dot{\omega}_{in} = T_{in} - (T_{CL1} + T_{CL2}) \tag{6}$$

where T_{CL1} and T_{CL2} are the torques carried by the dry double clutches respectively, and I_{in} is the mass moments of inertia of the flywheel.

As shown in Fig. 2, furthermore, equilibrium equations of motion for the gearbox in torsional vibrations may be presented as follows

$$I_{CL1}\dot{\omega}_{CL1} = T_{CL1} - T_{g1} \tag{7}$$

$$I_1\dot{\omega}_1 = T_{g1}i_{g1} - T_1 \tag{8}$$

$$I_{CL2}\dot{\omega}_{CL2} = T_{CL2} - T_{g2} \tag{9}$$

$$I_2\dot{\omega}_2 = T_{g2}i_{g2} - T_2 \tag{10}$$

$$I_{out}\dot{\omega}_{out} = T_1 i_f + T_2 i_f - T_{out} \tag{11}$$

where I_{CL1}, I_{CL2}, I_1, I_2 and I_{out} are respectively the inertia of moments of CL1, CL2, odd gear, even gear and output shaft assemblies, ω_{CL1}, ω_{CL2}, ω_1, ω_2 and ω_{out} are respectively the angular velocities of CL1, CL2, odd gear, even gear and output shaft, i_{g1} and i_{g2} are the gear ratios of the odd and even gears, i_f is the final drive gear ratio, T_{out} is the torque at the output shaft, and T_{g1}, T_{g2}, T_1, and T_2 are torques as shown in Fig. 2.

The equilibrium equation of motion in torsional vibrations for the output shaft can be expressed in the following form as

$$T_{\mathrm{out}} = \mathrm{K}_2(\theta_{\mathrm{out}} - \theta_{\mathrm{w}}) + \mathrm{C}_2(\omega_{\mathrm{out}} - \omega_{\mathrm{w}}) \tag{12}$$

where θ_{out} and θ_{w} are the rotational angles of the output shaft and the wheel, ω_{out} and ω_{w} are the angular velocities of the output shaft and the wheel, K_2 is the stiffness of the output shaft, and C_2 is the damping coefficient of the out shaft.

3.3 Vehicle dynamics model

The torque T_{w} acting on the driving wheel generates the traction force to overcome the force of resistance and to accelerate the vehicle. The summation of the rolling and aerodynamic resistances constitutes the propulsion load for the vehicle. The dynamics model of the vehicle can be expressed, therefore, as follows

$$T_{\mathrm{out}} - T_{\mathrm{w}} = mR\frac{\mathrm{d}v}{\mathrm{d}t} \tag{13}$$

$$T_{\mathrm{w}} = (\xi mg + \frac{1}{2}\rho C_D A v^2)R \tag{14}$$

where R is the wheel rolling radius, m is the vehicle mass, v is the total velocity of the vehicle, ξ is the rolling resistance factor, ρ is air density, C_{D} is the aerodynamic resistance coefficient, and A is the front area of the vehicle.

4 Controller design

The main goal of the DCT controller is focused on improving shifting quality by balancing friction work down of the dual-clutch.

Unsmooth shift process yields torsional oscillations of the driveline, which should be prevented from the vehicle passengers comfort in the longitudinal dynamics. For evaluation of the driving smoothness of the vehicle, the term shock intensity J is defined the change rate of the vehicle's longitudinal acceleration a, which is recommended to satisfy $|\dot{a}| \le 10\ \mathrm{m/s^3}$, with respect to time of the following form as

$$J = \frac{\mathrm{d}a}{\mathrm{d}t} = \frac{\mathrm{d}^2 v}{\mathrm{d}t^2} \tag{15}$$

From Eq. (15), it is observed that smooth shift of the DCT vehicle can be achieved if the vehicle acceleration keeps constant ($\frac{\mathrm{d}v}{\mathrm{d}t} = c$) during a shift.

Then, angular accelerations of CL1, CL2, odd gear, even gear, output shaft and wheel are related to the vehicle acceleration and can be given in the following as

$$\dot{\omega}_{\mathrm{w}} = \dot{\omega}_{\mathrm{out}} = \frac{c}{R} \tag{16}$$

$$\dot{\omega}_1 = \dot{\omega}_2 = \frac{c i_{\mathrm{f}}}{R} \tag{17}$$

$$\dot{\omega}_{\mathrm{CL1}} = \frac{c i_{\mathrm{f}} i_{\mathrm{g1}}}{R} \tag{18}$$

$$\dot{\omega}_{\mathrm{CL2}} = \frac{c i_{\mathrm{f}} i_{\mathrm{g2}}}{R} \tag{19}$$

substitution equation (16)-(19) into equations (7)-(14) to obtain the relationship between T_{CL1} and T_{CL2} as following

$$T_{\mathrm{CL1}} = -\frac{i_{\mathrm{g2}}}{i_{\mathrm{g1}}}T_{\mathrm{CL2}} + \frac{i_{\mathrm{f}} c}{R}(I_{\mathrm{CL1}} i_{\mathrm{g1}} + \frac{I_1}{i_{\mathrm{g1}}} + \frac{I_{\mathrm{CL2}} i_{\mathrm{g2}}^2}{i_{\mathrm{g1}}} + \frac{I_2}{i_{\mathrm{g1}}}) + \frac{R(\xi mg + \frac{1}{2}\rho C_D A v^2 + mc)}{i_{\mathrm{f}} i_{\mathrm{g1}}} \tag{20}$$

thus, the engagement smoothness is related to the relationship between T_{CL1} and T_{CL2} and, hence, we will consider this quantity as a quantitative performance criterion.

By combination of Eqs (4), (6) and (20), together with inertia effects of the engine and flywheel, the desired engine torque required for torque transmission can be presented as follows

$$\hat{T}_{\mathrm{e}} = I_{\mathrm{e}}\dot{\omega}_{\mathrm{e}} + I_{\mathrm{in}}\dot{\omega}_{\mathrm{in}} + T_{\mathrm{CL1}} + T_{\mathrm{CL2}} \tag{21}$$

then the throttle angle can be calculated according to the desired engine torque, the current engine speed and a look-up table which has the engine map incorporated in it.

Another control objective which is to follows maintain as low as possible the friction heart dissipation of the clutch plates during the period of engagement can be written for different phases of shifting as

$$W = \int_0^t [T_{\mathrm{C1}}(\omega_{\mathrm{in}} - \omega_{\mathrm{CL1}}) + T_{\mathrm{C2}}(\omega_{\mathrm{CL2}} - \omega_{\mathrm{in}})]\mathrm{d}t \tag{22}$$

where t is the time change during shift. As obviously observed, from equation (22), the shift time should be as short as possible to reduce clutch plate overwear.

Owing to model inaccuracies or due to external disturbances unaccounted for (like wear of friction discs), differences between the speeds of engine and dual-clutch and desired reference signals will occur.

The main goal of the DCT controller is to achieve fast and accurate tracking of the desired engine and dual-clutch speed trajectories. Furthermore, the controller should also be robust for disturbances.

In our approach, the control objectives are met by ensuring that the engine and dual-clutch speeds track desired reference signals. The input signals for the controller are measurements of the engine and clutch speeds, and the estimation of the engaging torque during shift. The general architecture of the controlled system is given in Fig. 4. The gearshift controller output signals are the reference torques of engine and two clutches, and are generated from the gearshift controller on the basis of the reference torque of CL1 T_{CL1}^{ref} and the reference speeds ω_{e} and ω_{CL2}. The signal T_{e}^{ref} is actuated by the engine control unit, which is here assumed to be ideal, i.e., $T_{\mathrm{e}} = T_{\mathrm{e}}^{ref}$. The reference signal T_{CL1}^{ref} and T_{CL2}^{ref} actuated by the closed-loop electromechanical actuator approximated through the dual-clutch characteristic.

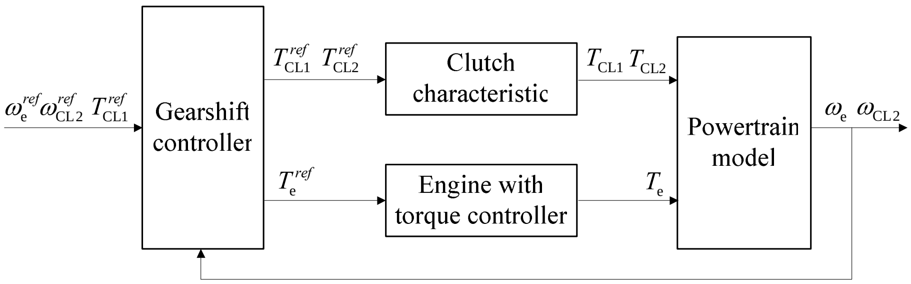


Fig. 4 General closed-loop scheme during shift

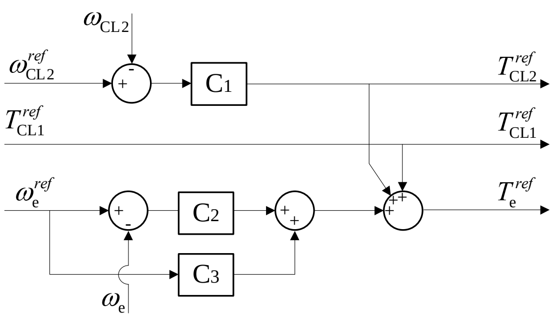


Fig. 5 Block diagram of the DCT gearshift controller

A block diagram of the gearshift controller is shown in Fig. 5. The feedback loop on the CL2 clutch speed provides the reference clutch torque T_{CL2}^{ref} . The controller C_3 realizes a feedforward action obtained by computing the left-hand side of (5) after replacing the actual engine speed with the corresponding reference signal. The loop on the engine speed, together with the corresponding feedforward compensation, provides the desired difference between the engine torque and the clutch torque. This scheme can be viewed as a decoupling controller scheme and thus the two controllers C_1 and C_2 are PID designed by applying to the model (5)–(16). The controller parameters P, I and D have been tuned manually.

5 Simulation results

The powertrain model and the controller are implemented in Matlab/Simulink environment for the shift control and simulation. According to speeds of engine and dual-clutch friction discs, the shift control algorithm regulates the engine and dual-clutch. Using engine throttle angle and the engagement forces on friction discs as control signals, vehicle responses during transmission 1-2 upshift and 2-1 downshift are studied. Table 1 shows technical parameters of a midsize vehicle equipped with a dry dual-clutch transmission. The input and output shafts damping constants are estimated via Rayleigh’s dissipation term [16].

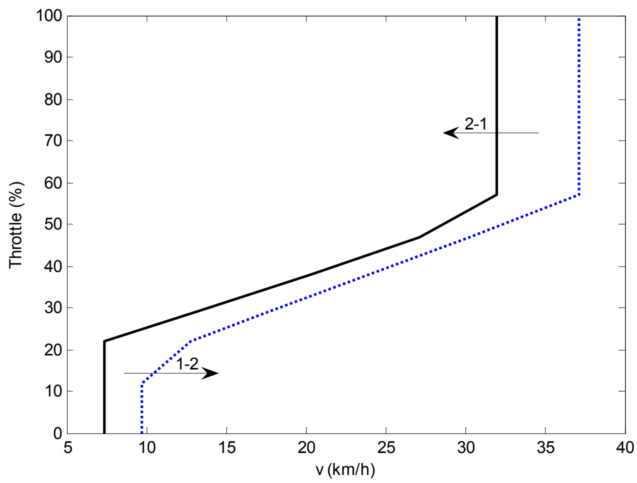


Fig. 6 Gear shift schedule

The transmission control system controls the shift schedule based on current vehicle speeds and throttle angles. As shown in Fig. 6, the curve at the right represents the threshold for the 1-2 upshift, and that on the left for the 2-1 downshift.

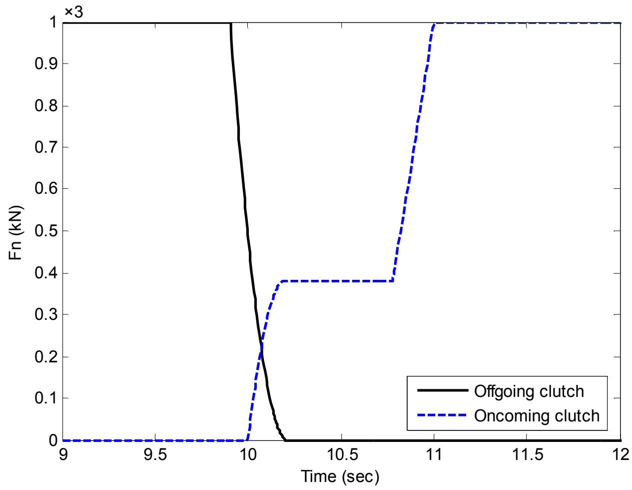
Table 1 Vehicle and transmission parameters

Iterms	Value
Vehicle mass	1660 kg
Tire radius	0.32 m
Transmission gear ratios	First 4.2, second 2.5, third 1.6, fourth 1.2, fifth 1.0, sixth 0.9
Final drive gear ratios	3.07
Odd gear's moment of inertia	0.002 kg m^2
Even gear's moment of inertia	0.002 kg m^2
Engine moment of inertia	0.33 kg m^2
Input shaft moment of inertia	0.1 kg m^2
Input shaft spring constant	12000 N m/rad
Input shaft damping constant	100 N m s/rad
Output shaft moment of inertia	0.1 kg m^2
Output shaft spring constant	15000 N m/rad
Output shaft damping constant	360 N m s/rad
CL1 moment of inertia	0.1 kg m^2
CL2 moment of inertia	0.1 kg m^2
Tire moment of inertia	1.15 kg m^2
The constant friction coefficient	0.3
The number of engaging surface	2
The friction radius of CL1	97 mm
The friction radius of CL2	94 mm

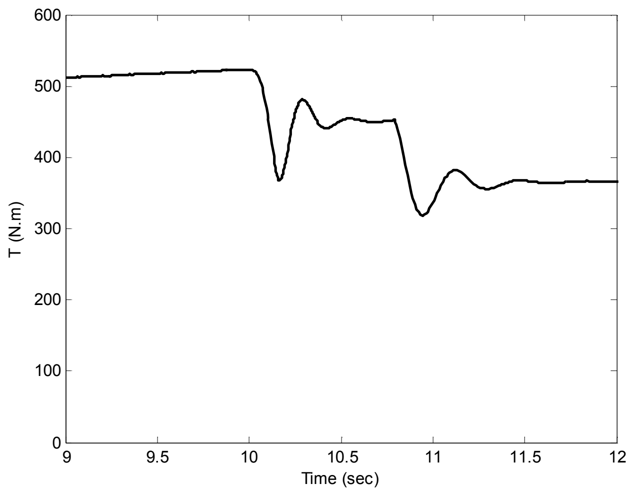
5.1 Upshift

Numerical results of the force of engagement F_n for the two friction discs and the engine throttle angle are illustrated in Fig. 7, using the control algorithm for the case of 1-2 upshift. In the beginning of the upshift, the force of engagement of the offgoing CL1 is reduced by the control strategy. When the force engagement of the offgoing CL1 is reduced to the value of where the clutch starts to slip, the force engagement of the oncoming CL2 is increased. For the synchronous speeds of the target gear and the engine, the engine throttle angle should be depressed within the inertia phase, according to Eq. (21). At the end of the inertia phase the force of engagement of the oncoming CL2 is sharply reached to its total engagement.

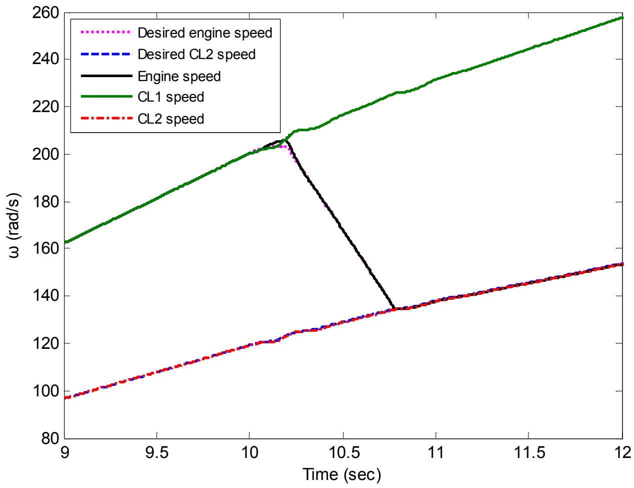
In the case of an upshift from 1st to 2nd gears, simulation results for dynamic responses of output torque T_{out}, angular velocity ω of both friction discs including engine, and shock intensity J are also presented in Fig. 7. In this situation, for 1-2 upshift the vehicle is driven on a straight lane at the speed of 17.51 km/h determined according to the current throttle angle and a look-up table which has the shift schedule incorporated in it.



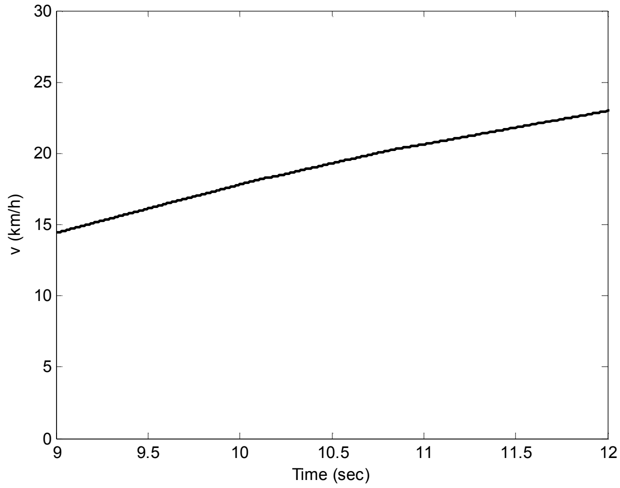
(a) The engaging forces on two friction discs



(b) Output torque

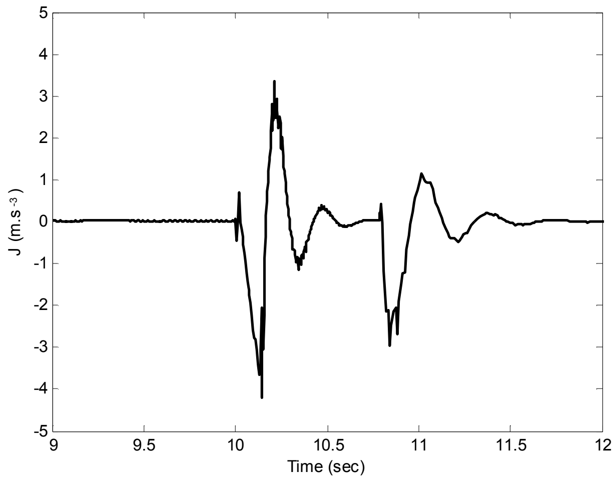


(c) The angular velocities of both clutches including engine

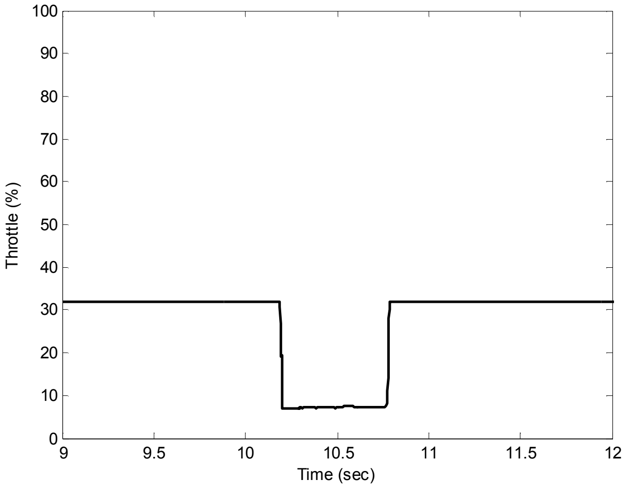


(f) Vehicle speed

Fig. 7 Simulation results of a 1-2 upshift



(d) Shock intensity

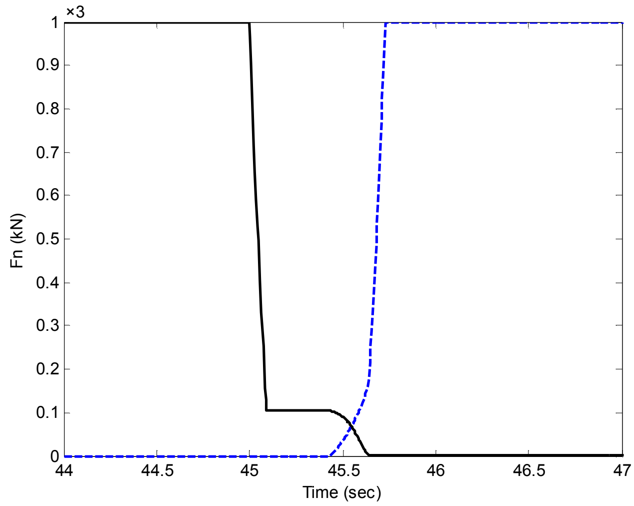


(e) Throttle angle

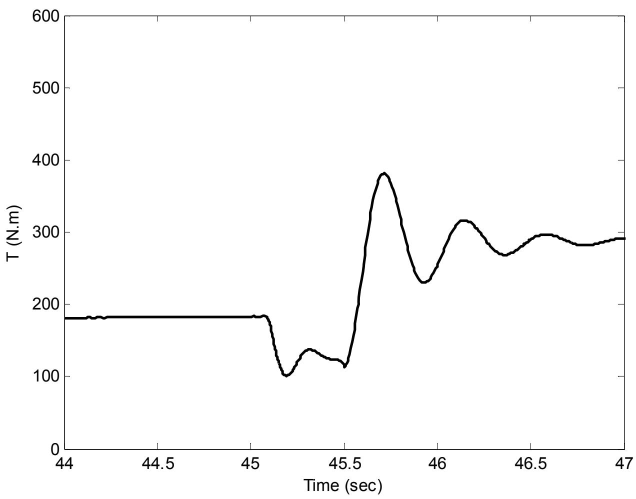
During the upshift, closed-loop tracking of reference speeds of engine and CL2 is achieved through manipulation of engaging force at the CL2 and engine torque. As shown in Fig. 7 from the output torque and shock intensity, the output torque the transition is reduced gradually without impacts from the torque to the inertia phases. The shock is acceptable, because the shock intensity of the vehicle is less than 5 m/s^3. Then, the DCT vehicle enables the smooth shift from the 1^{st} to the 2^{nd} gears. Before the shift, however, the offgoing clutch is engaged to allow the engine torque to transmission directly to the driveline. In the torque phase the engine torque is transferred by both the clutches simultaneously and the two clutches are partially closed. When the offgoing CL1 is total released, the 2^{nd} gear is ready to shift and the inertia phase starts. Within the inertia phase, the engine speed must be slow down to approach to the speed of synchronization with the target gear.

5.2 Downshift

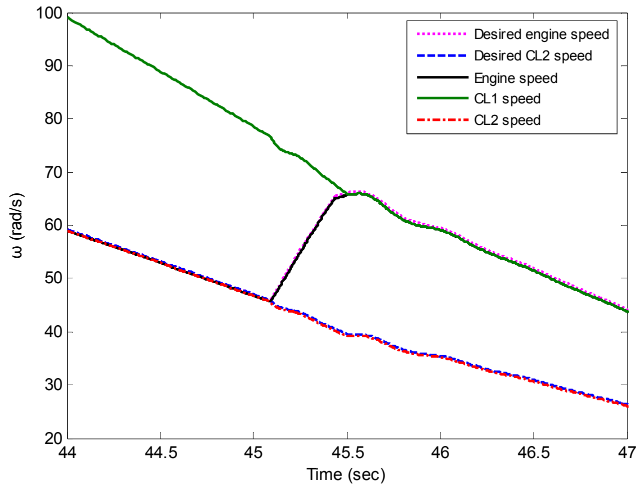
On contract to the upshift, the downshift operated by the inertia phase in advance, and then followed by the torque phase. Fig. 8 shows a simulation result for an upshift from 1^{st} to 2^{nd} gear, using the proposed control algorithm. In the beginning of the downshift, the force of engagement acting on the friction disc of the offgoing CL2 is sharply reduced in a similar way to that in the torque phase of the upshift process. For the synchronous speed of the target gear within the inertia phase of the downshift, the engine speed is accelerated by increasing the throttle angle, according to Eq. (21). By the end of the inertia phase, the oncoming CL1 is increased whereas the force of engagement of the offgoing CL2 is reduced in association with decrease of the throttle angle.



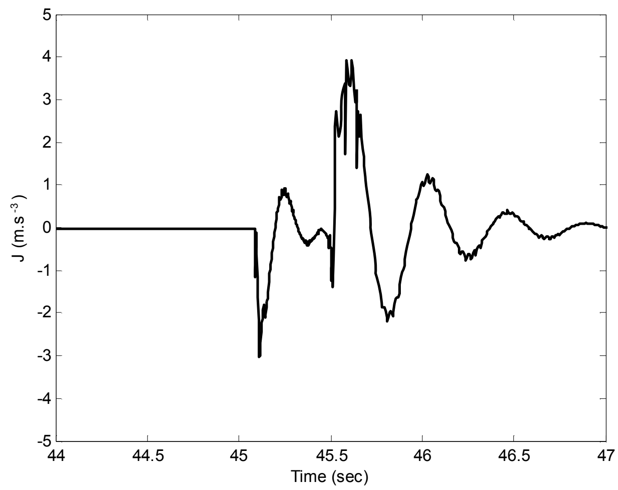
(a) The engaging forces of two friction discs



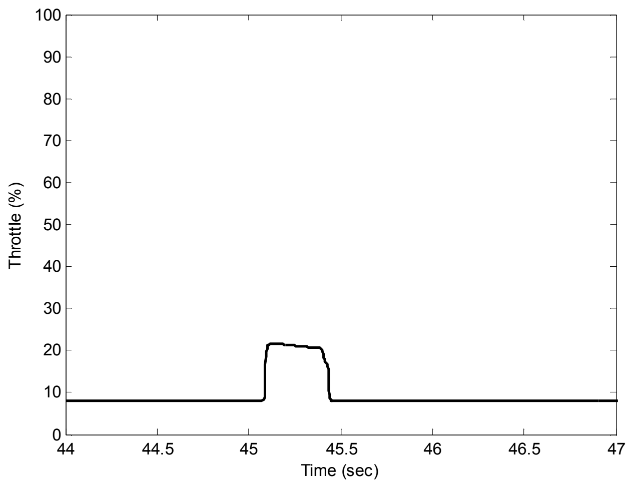
(b) Output torque



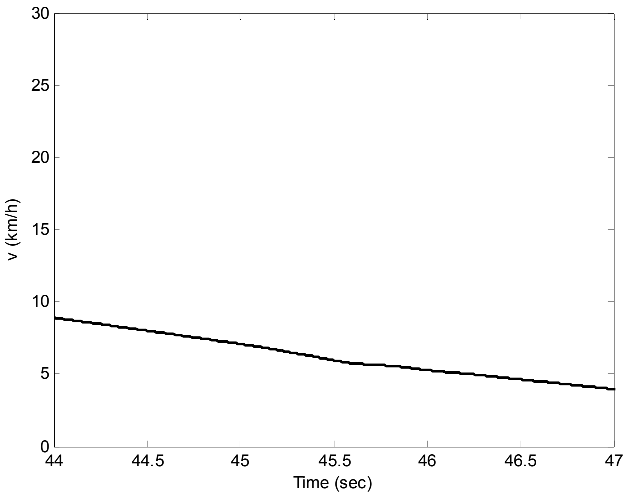
(c) The angular velocities of both clutches including engine



(d) Shock intensity



(e) Throttle angle



(f) Vehicle speed

Fig. 8 Simulation results of a 2-1 downshift

In the test, the speed in the case of 2-1 downshift is 7.069 km/h for simulation. It can be seen that the tracking performances of speed reference trajectories are good. The smooth output torque is obtained during the downshift process. In the beginning of the downshift, difference of speeds between the offgoing CL2 and the engine occurs due to reduction of the force of engagement and friction slippage increases. Meanwhile, the engine and the oncoming CL1 disc are initially engaged and the downshift process is alternated to the torque phase.

6 Conclusion

The dynamic and mathematical modeling of the dry DCT system with incorporation of the dual clutch is presented for numerical simulations of transient responses of a vehicle during gear shifting processes. For the purpose of improving the shift quality and gearshift performance, the optimal control algorithm is proposed based on decoupled speed and torque control loops to compromise torque impacts and friction energy dissipation of the dry dual-clutch. The mathematical model for the propulsion system is developed by using the software package Matlab/Simulink, in which engine throttle angle and dual-clutch engaging forces as control signals to regulate the DCT shifting behaviors. Simulation results are produced and compared. It has been shown that the effectiveness of the control algorithm during upshift and downshift is proved. Useful numerical predictions are provided for the smooth shift control strategy to promote the objective judgment on the DCT shift rules before the dry DCT prototyping is developed.

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