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DEVELOPMENT OF DCT VEHICLE PERFORMANCE SIMULATOR TO EVALUATE SHIFT FORCE AND TORQUE INTERRUPTION

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ABSTRACT−This paper presents shift characteristics of a dual clutch transmission (DCT). To obtain the shift force, dynamic models of the DCT are constructed by using MATLAB/Simulink and considering the rotational inertia of every component and the target pre-select time. Dynamic models of the shift and clutch actuators are derived based on the experimental results of the dynamic characteristics test. Based on the dynamic model of the DCT synchronizer, control actuator and vehicle model, a DCT vehicle performance simulator is developed. Using the simulator, the shift force and speed of the relevant shafts are obtained. In addition, the torque and acceleration of actuators are calculated during the shift process by considering the engaging and disengaging dynamics of the two clutches. It is observed from the performance simulator that uninterrupted torque can be transmitted by proper control of the two clutches.

KEY WORDS : Dual clutch transmission (DCT), Clutch actuator, Shift actuator, Vehicle performance simulator

1. INTRODUCTION

The increasing demands to reduce fuel consumption and to improve efficiencies and driving comfort of an automobile are leading the automotive industry to further developments of the modern transmission concept. At present, manual transmission and automatic transmission are both well established. In the last few years, automated manual transmission (AMT) and continuously variable transmission (CVT) were introduced to the market, but they still have just a small market share (Grosspietsch *et al.*, 2001; Yeo *et al.*, 2004).

Through the automation of the engaging and coupling processes, the AMT has become closer automatic transmission with respect to comfort. An increase in the number of gears and thus, the increase in the spread improve the ride comfort. High efficiency, low costs, and favorable fuel consumption are all retained. However, torque interruption during the gear change has been a major problem because it degrades driving comfort.

Dual clutch transmission (DCT) is expected to be the next generation transmission, which will consist of a manual gear box, sensors, and actuators. DCTs have two separate transmission input shafts. The gears are preselected in the respective load-free branch of the gearbox. The gear steps change under full load by means of a controlled torque transfer from the first to the second clutch. The transmission efficiency close to that of the manual gearbox is achieved by intelligent mechatronic control of the two clutches and two input and output shafts. Precise control of the hydraulics and electronics enables the next higher gear to be permanently engaged and ready for activation. Unlike a typical automated manual transmission, elaborate clutch management provides no interruption in power transmission during shifting. Volkswagen introduced the first DCT to the market with Borg Warner, using the company's Dual Tronic wet-clutch and control system technology (Jost, 2003).

DCTs belong to powershift transmission based on manual synchromesh gearboxes (Grosspietsch *et al.*, 2001). Two dry plate or wet-type clutches are used to move the vehicle away from rest and for gearshifts. Downstream of each clutch is a multi-ratio synchromesh transmission. Automation and control of these systems are difficult but can be reasonably achieved with electronic control. In addition to all the functions required for a conventional automatic transmission, the starting clutch and the synchronizers also need to be controlled. Feedback from the synchronizers is required to recognize the end of the synchronizing phase (Schwab, 1999).

Dynamic characteristics of the actuators as well as the steady state characteristics are considered to be integral elements in the automation of the clutch and synchromesh activation and in the control system design. Steady

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state characteristics are determined from the design requirements such as actuator force, time etc. In the pre-select process, the actuator force depends on the synchronization time. A fast synchronization time needs a large actuator force; a slow synchronization needs a small actuator force. These demands may conflict with the driver's demand when the driver wants to change the shift gear (Moon *et al.*, 2005).

In this paper, dynamic models of a DCT powertrain are obtained, and a DCT performance simulator is developed. Using the simulator, shift characteristics in the pre-select process and DCT vehicle performance such as drivetrain torque and acceleration are investigated by considering the shift and clutch actuator characteristics.

2. MODELING OF DCT

2.1. Shift Mechanism of DCT for Pre-select

In Figure 1, the DCT investigated in this study is shown. It consists of 2 clutches C1 and C2, and 2 shafts S1 and S2. Clutch C1 is connected to shaft S1, which transmits power to the 1st, 3rd and 5th shift gear and the reverse gear, and clutch C2 is connected to shaft S1, which transmits power to the 2nd, 4th and 6th shift gear. The shift in the DCT is carried out as follows: when accelerating in the 1st gear, clutch C1 is engaged. When the next gearshift point is approached, the 2nd gear is pre-selected by actuating the synchronizer but is not active since C2 has not yet engaged. As soon as the ideal shift point is reached, C1 for the 1st gear disengages while C2 closes, activating 2nd gear. The gear change takes place under load, and consequently, power flow is maintained.

Figure 2 shows a double cone used in this study. Synchronizer is an integral element that generates the friction torque to reduce the synchronized side speed at upshift or to increase the synchronized side speed at downshift and changes the power flow (Kim *et al.*, 2004).

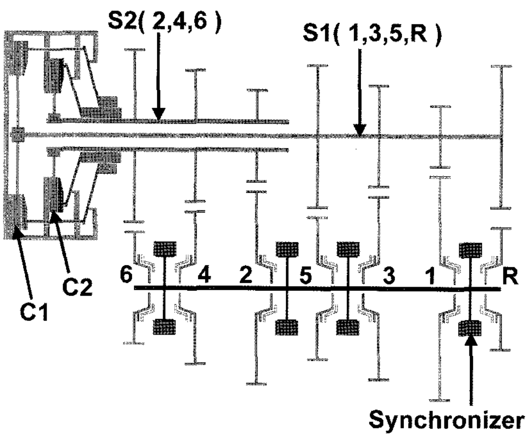

Figure 1. Schematic diagram of DCT.

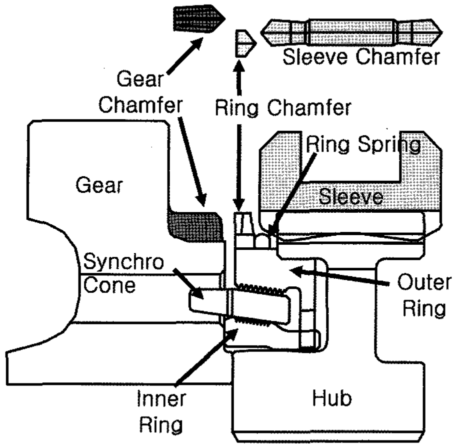

Figure 2. Synchronizer.

The double cone synchronizer in Figure 2 consists of the sleeve, gear, synchro cone, hub, outer ring and inner ring.

Figure 3 shows the friction torque generation mechanism. The friction torque T_{cone} is generated by the sleeve force F_{sleeve} as follows,

$$T_{cone} = \frac{\mu}{\sin\theta} R_{cone} F_{sleeve} \tag{1}$$

where μ is the cone friction coefficient of the cone, θ is the cone angle, R_{cone} is the cone mean radius, F_{sleeve} is the sleeve force.

The required friction torque can be derived in terms of the synchronization time by the following equation,

$$T_{cone} = \frac{\omega_i - \omega_f}{\Delta t} J_{ref} \tag{2}$$

where ω_i, ω_f are the initial and final velocity for the synchronization, respectively, J_{ref} is the reflected inertia, and t is the synchronization time.

From Equations (1) and (2), the sleeve force F_{sleeve} can

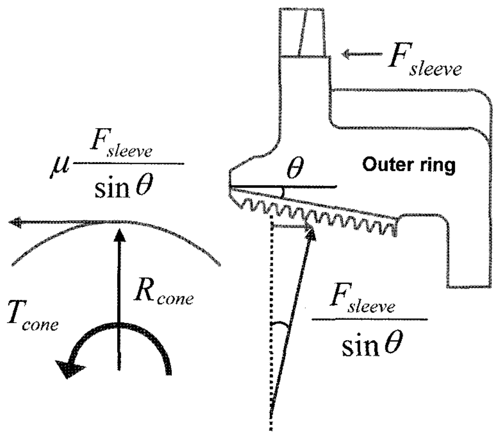

Figure 3. Cone torque.

be obtained (Bansbach, 1998).

$$F_{sleeve} = \frac{\omega_i - \omega_f}{\Delta t} \times J_{ref} \times \frac{\sin\theta}{\mu \cdot R_{cone}} \tag{3}$$

In Figure 4, a shift actuator linkage system is shown. The linkage system transmits the actuator force to the synchronizer sleeve. The synchronizer sleeve axial motion generates the friction force required for synchronization. The equation of sleeve motion V_{sleeve} is represented as

$$M_{sleeve}\dot{V}_{sleeve} = K_{fork}X_{fork} + B_{fork}(V_{control\_shaft} - V_{sleeve}) - B_{sleeve}V_{sleeve} - F_{reaction} \tag{4}$$

where M_{sleeve} is the sleeve mass, K_{fork} is the fork stiffness, X_{fork} is the relative displacement of the fork, B_{fork} is the fork damping coefficient, $V_{control\_shaft}$ is the velocity of control shaft, B_{sleeve} is the sleeve damping coefficient, $F_{reaction}$ is the reaction force applied to the sleeve.

2.2. Shift Actuator

For the shift actuation of the DCT in Figure 1, an electro-hydraulic actuator is used. In Figure 5, the hydraulic system of the shift actuator is shown. A proportional solenoid valve is used to supply the actuator pressure. The actuator pressure is controlled by the input voltage. Dynamic equation of the actuator pressure can be represented by

$$\frac{A_{act} + A_{act}X_{act}}{\beta}\frac{dP_{act}}{dt} = Q_{act} - A_{act}\dot{X}_{act} \tag{5}$$

where V_{act} is the actuator volume, A_{act} is the actuator piston area, X_{act} is the displacement of the actuator piston, β is the bulk modulus, P_{act} is the actuator pressure, and Q_{act} is the actuator flow rate.

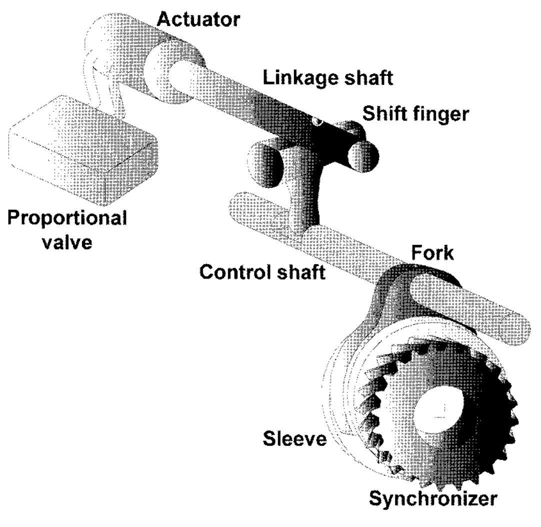

Figure 4. Linkage system.

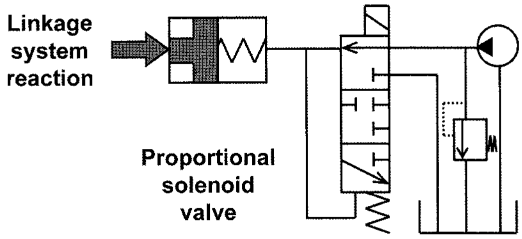

Figure 5. Shift actuator system.

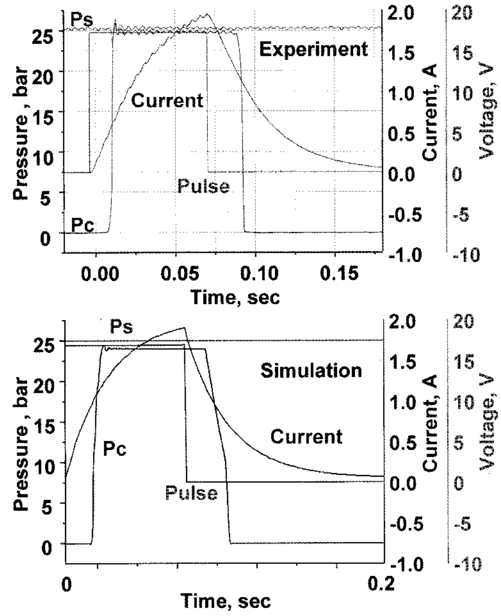

Figure 6. Experimental and simulation results of proportional valve.

In Figure 6, the simulation results of the shift actuator control valve are compared with the experimental results. As shown in Figure 6, the control pressure shows some response delay for a stepwise voltage input. This response delay is caused by the dynamic characteristic of the solenoid valve.

2.3. Clutch Actuator

In Figure 7, hydraulic system of the clutch actuator is shown. A flow control type solenoid valve supplies the hydraulic flow to the dry clutch. The clutch controller generates the control voltage to compensate the displacement error for a target displacement of the clutch release stroke.

Figure 8 shows the relationship between the clutch release stroke and release load used in this study.

2.4. Drivetrain

Figure 9 shows a schematic diagram of the DCT

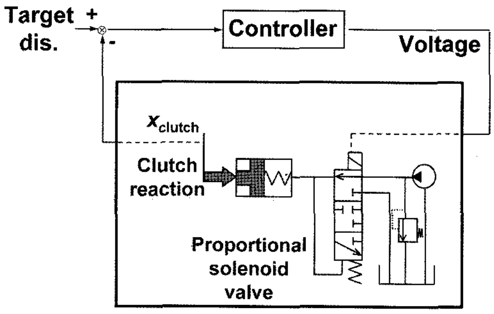

Figure 7. Clutch actuator system.

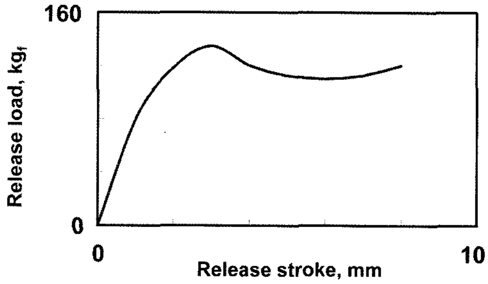

Figure 8. Clutch release load.

drivetrain investigated in this study. When the sleeve moves by the shift actuator, friction occurs between the rings and the gear cone and they become synchronized. During the synchronization, this friction force affects the rotational speed of the input and the output shaft.

The speed equation of the input shaft S1 and S2 for the 1–2 upshift preselect process is represented as

$$\dot{\omega}_{S1} = -\frac{b_{S1} + b_{damping}\left(\frac{1}{N_1^2} + \frac{1}{N_3^2} + \frac{1}{N_5^2} + \frac{1}{N_r^2}\right)}{J_{eq.S1}}\omega_{S1} \tag{6}$$

$$\dot{\omega}_{S2} = \frac{\frac{T_{cone}}{N_2} - \left\{b_{S2} + b_{damping}\left(\frac{1}{N_2^2} + \frac{1}{N_4^2} + \frac{1}{N_6^2}\right)\right\}\omega_{S2}}{J_{eq.S2}} \tag{7}$$

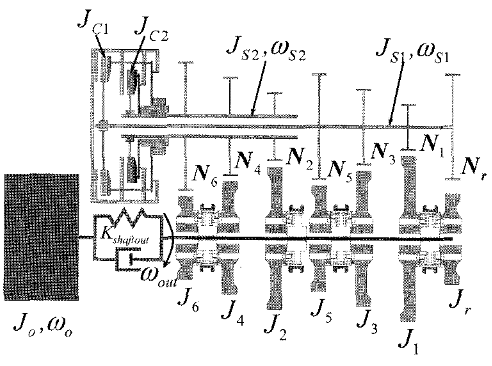

Figure 9. Drivetrain for DCT.

where ω_{S1} is the speed of shaft S1, ω_{S2} is the speed of shaft S2, $J_{eq.S1}$ and $J_{eq.S2}$ are the equivalent values of inertia for input shafts S1 and S2, respectively, and b_{S2} are the rotational damping coefficients. Output shaft speed equation for 1–2 upshift pre-select is obtained by

$$J_o \cdot \dot{\omega}_o = -b_o\omega_o - K_{shaftout}\theta_{shaftout} - b_{shaftout}(\omega_o - \omega_{out}) \tag{8}$$

$$\dot{\omega}_{out} = \frac{1}{J_{eq.out}}\{K_{shaftout}\theta_{shaftout} + b_{shaftout}(\omega_o - \omega_{out}) - N_2T_{cone} - b_{out}\omega_{out}\} \tag{9}$$

$$\dot{\theta}_{shaftout} = \omega_o - \omega_{out} \tag{10}$$

where $J_{eq.out}$ is the equivalent inertia viewed from the output shaft, J_o is the equivalent inertia of the vehicle, K_{out} is the stiffness of the output shaft, and b is the damping coefficient.

3. SIMULATION RESULTS

Figure 10 shows a DCT simulator developed in this study by using MATLAB/Simulink. The DCT simulator is developed based on the dynamic models of DCT, synchronizer, linkage system, actuator and drivetrain. The DCT simulator consists of an engine, clutch, drive-train and vehicle dynamic module. System parameters used in the simulation are shown in Table 1.

In Figure 11, simulation results for 1–2 upshift pre-select are shown. In the simulation, the target synchronization time is selected as $\Delta t = 0.2$ seconds. For the given pre-select time of 0.2 seconds, the required shift force is calculated as 650 N from Equation (3). The shift actuator force is determined as 900 N by considering the lever ratio of the linkage system. The actuator pressure, which is equivalent to the actuator force, is obtained as 25 bars. Correspondingly, the control valve input voltage, which is required to generate the target pressure, can be determined as 18 volt from the control valve characteristic, which is shown in Figure 11(a). In Figure 11, the dynamic response of the actuator pressure (b) and actuator force (c) are shown. The actuator pressure (b) follows the target value after some time delay. The actuator force (c) shows

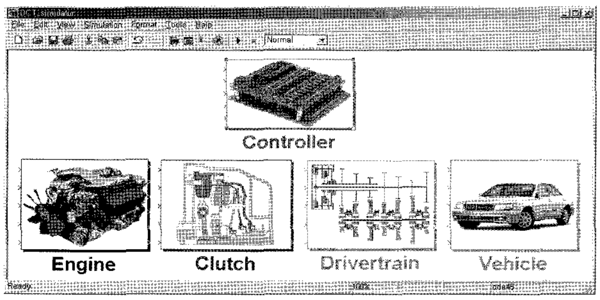

Figure 10. DCT simulator.

Table 1. Parameters used in simulation.

Parameter	Value
First stage gear ratio	3.636
Second stage gear ratio	2.206
Third stage gear ratio	1.516
Fourth stage gear ratio	1.143
Fifth stage gear ratio	0.903
Sixth stage gear ratio	0.756
Engine	1600 cc
Vehicle mass	1300 kg

a similar response. The sleeve reaction force shows a high peak at the initial stage of the contact and follows the reference force with some oscillation.

The pre-select procedure can be divided into four steps, as described by the dotted line. In step 1, the sleeve moves forward, clearing the gaps between the sleeve and the outer ring. In step 2, the sleeve chamfer maintains contact with the outer ring chamfer. In this step, the actuator pushes the linkage system to generate the friction torque (e) required to increase the synchronized side (S2) speed for the synchronization. Correspondingly, the speed of the shaft S2 (f) increases. As shown in the S2 speed response (f), most of the synchronization process is accomplished in the step 2. In addition, the outer ring chamfers are aligned for the sleeve chamfers to pass through the outer ring chamfers. Once the outer ring (gear) speed becomes higher than the output shaft speed, the outer ring rotates relative to the sleeve and the sleeve moves forward. Therefore, the sleeve reaction force (d) decreases and shows zero value when the outer ring is ready to allow the sleeve to pass between the outer ring chamfers. In step 3, the sleeve moves past the outer ring until it impacts the gear. Since no reaction force is applied to the sleeve (d) while the sleeve is moving, no friction torque (e) is generated. In this step, the impact may occur between the sleeve and the gear. During step 2, the linkage mechanism has been compressed, storing the energy in the system. Since the sleeve is released at the end of step 2 by the stored energy, the sleeve impacts the gear in the step 3. The spike in the sleeve force (d) in the step 3 is caused by the impact. Step 4 is the region where the synchronization is finished completely.

Using the DCT vehicle simulator, performance simulations are carried out. In Figure 12, simulation results are shown. In Figure 12(a), throttle pedal opening profile is used as an input for the simulation. In addition, the clutch displacement profiles of C1 and C2 (b) are used for the start and 1–2 upshift. The engine torque (c) is generated according to the throttle opening. The engine speed (d) increases with the throttle opening and decreases as the clutch C1 is engaged. The engine speed increases again with the 1st gear until the gear ratio is shifted to the 2nd gear. The C1 and C2 clutch displacements (e) follow the target displacement (b) by the clutch actuation. The engine torque is transmitted to the clutch side, showing uninterrupted torque shape (f) due to the C1 and C2 clutch control. Damper spring torque (g) shows some oscillation due to the damper clutch dynamics. The small peak in the dotted circle is caused by the rotational inertia change during the pre-select process. The vehicle velocity (h) increases without showing any hesitation or even decreases, a behavior which can be normally found in MT or AMT vehicle.

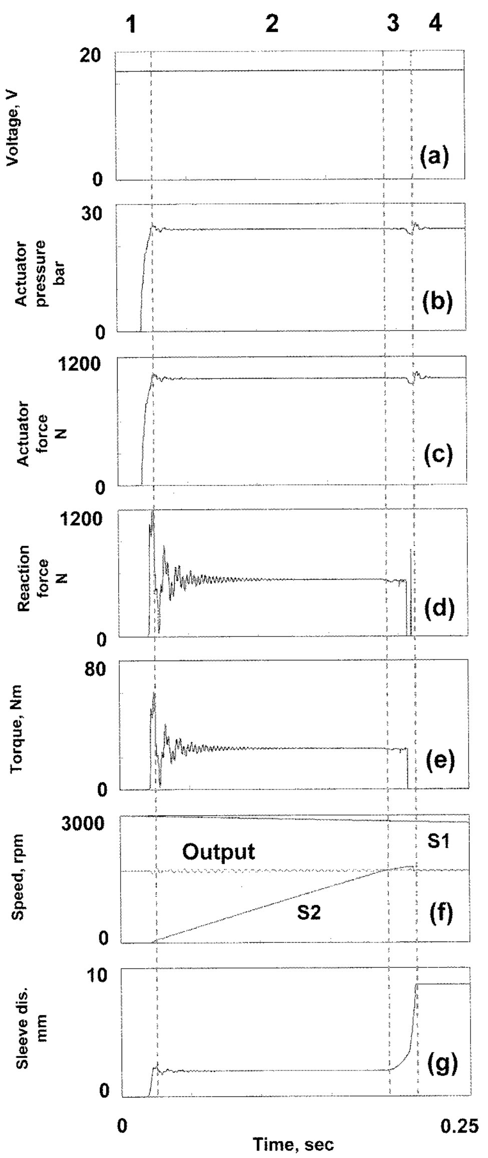

Figure 11. Simulation results of 1–2 upshift preselect.

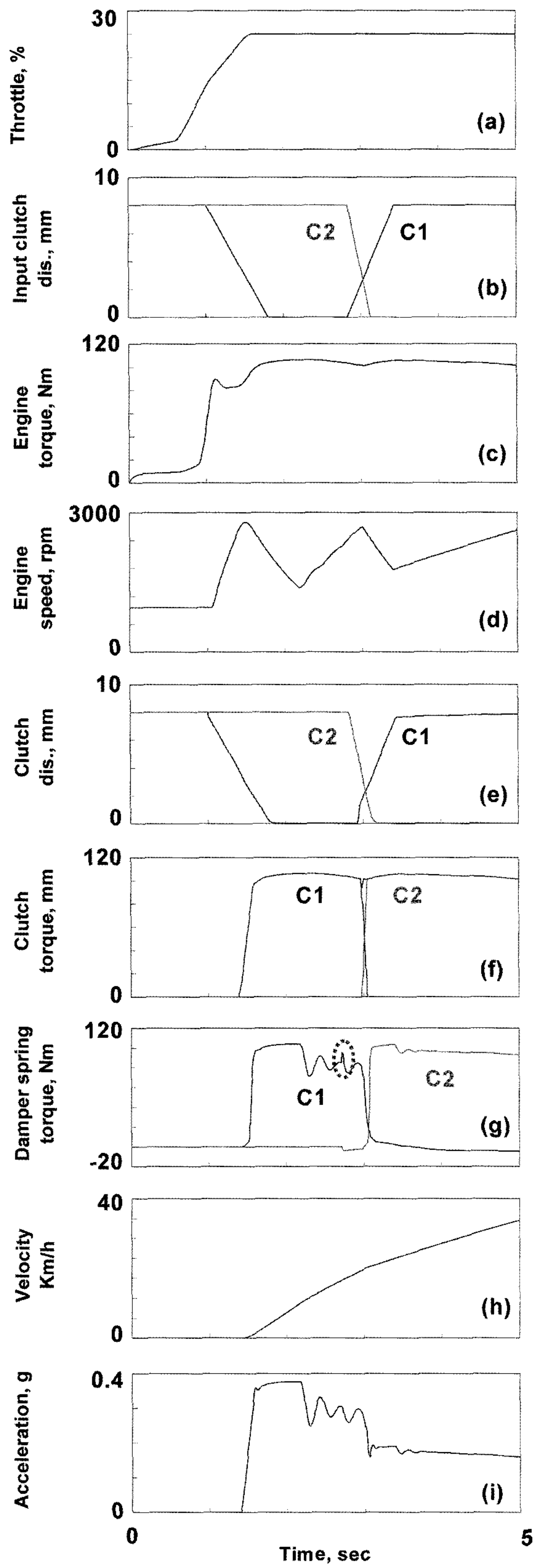

Figure 12. Simulation results of DCT vehicle performance.

The acceleration (i) decreases when the gear ratio is shifted from the 1st gear to the 2nd gear. The negative acceleration, which can be normally experienced in MT or AMT, does not occur due to the DCT's two clutch operation.

4. CONCLUSION

Shift performance for a DCT is investigated. To obtain the shift force, dynamic models of the DCT are constructed using MATLAB/Simulink and by considering the rotational inertia of every component. Shift force is obtained from a target preselect time. Dynamic models of the shift and clutch actuators are derived based on the test results of the control value. Based on the dynamic model of the DCT synchronizer, control actuator, and vehicle model, a DCT vehicle performance simulator is developed. Using the simulator, the shift force and speed of the relevant shafts are obtained. In addition, the torque and acceleration are calculated during the shift process by considering the engaging and disengaging dynamics of the two clutches. The simulator showed that uninterrupted torque transmission can be obtained by proper control of the two clutches.

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