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A dry clutch control algorithm for AMT systems in a parallel hybrid electric bus

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Abstract

A clutch is a mechanical device which provides engine torque to the drive shaft of a vehicle. A dry clutch control for AMT (Automated Manual Transmission) systems has been an important issue to improve fuel economy and drivability in hybrid electric systems. In this paper, we propose a dry clutch control system to apply to the parallel hybrid electric bus. In order to analysis dynamic performance of the target vehicle, a vehicle dynamic model including engine, clutch, motor, transmission and vehicle is designed. For gear shifting simulation, the shifting maps for the hybrid electric bus are applied from the analysis results of DP (Dynamic Programming) theory that is one of the optimal control methods. The shifting maps consist of a pure electric mode and a hybrid electric mode calculated by using driving cycles for commercial vehicles. From vehicle dynamic equations, the control algorithm for a dry clutch is organized by using feedback loops based on the value of an engine, a clutch speed, a clutch release travel and an estimated clutch torque. Simulations are performed to analyze the dynamic performance of the proposed clutch control system during gear shifting. As a result, the vehicle model with the designed clutch controller compares to one with only the lockup controller in energy dissipation during gear shifting.

Keywords: Commercial, HEV, Transmission, Control system

1 Introduction

AMT (Automated Manual Transmission) systems have been widely adopted to offer easy handling and fuel efficiency for heavy duty vehicles. AMT is also a suitable system for parallel hybrid electric vehicles (HEVs). Therefore HEVs equipped with AMT for commercial vehicles have been developed all around the world.

Hybrid powertrain systems provide an additional control dimension, as they can control not only engine power but also motor power. It is possible to reduce fuel consumptions and improve emissions by power management control in HEVs. A dry clutch control for AMT systems is also an important issue to improve fuel economy and drivability of vehicles. However, one limitation of AMT is driving comfort reduction, caused by torque difference during gear shifting. The problem is serious in the case of heavy duty vehicles, where powertrain is relatively large.

There have been many researches about a dry clutch controller for AMT systems. Glielmo, et al. proposed optimization-based algorithm for AMT

systems with dry clutches [1]. Model predictive control for dry clutches was used in ref.[2, 3]. These researches for the clutch control were assumed that the engine torque is constant during time interval.
In this paper, vehicle dynamic model was designed to implement the dynamic performance of a hybrid electric bus. We proposed a dry clutch control algorithm to apply to the parallel hybrid electric bus. From vehicle dynamic equations, the control algorithm was organized by using feedback loops based on measurements of engine, clutch speed and clutch release travel and on an estimated clutch torque. Simulation was implemented on the clutch operation during gear shifting. Based on the results of simulation, energy loss for the hybrid electric bus with the proposed clutch control system was calculated and compared to the simulation results for the same model with only lock-up control system.

2 Modeling of the vehicle

The configuration of the target system considered as a parallel hybrid electric bus with AMT systems. In order to analysis the dynamic characteristics, the simulation program was consisted of plant module, CCU (clutch control unit) module, TCU (transmission control unit) module and driver module by using Matlab simulink and simdriveline as shown in Fig.1. The plant module, including an engine, a dry clutch, a motor, a transmission, battery systems and a vehicle, was designed by using performance data of components.

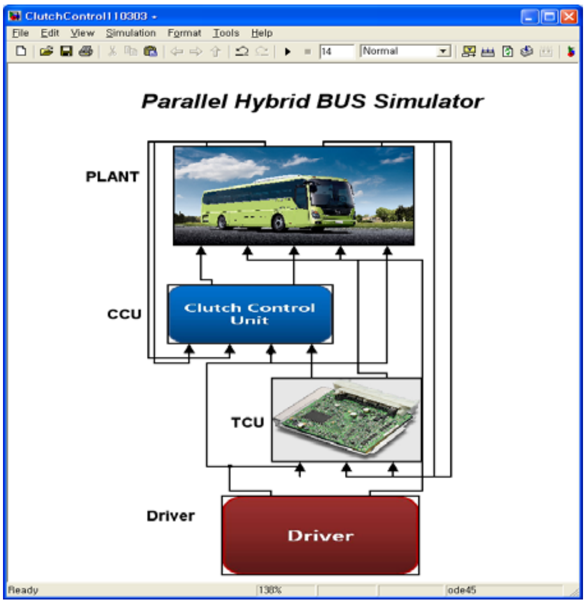


Figure1: Schematic view of simulation program

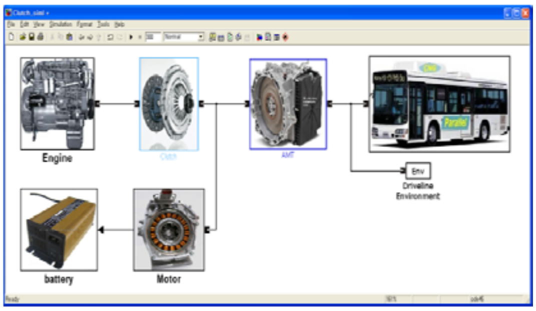


Figure 2: Construction of plant module

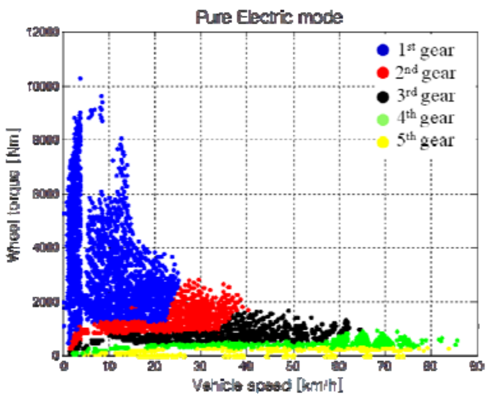
The construction of the plant module is shown in Fig. 2. The engine part in the vehicle modeling was generated by using torque map data respect to engine speed. Designed engine was considered as 6.4 liter CNG engine. The motor part in this model was also designed by using map data and considered as 60kW maximum powered motor. The transmission part was considered in 5 speed transmission. The battery performance data respect to SOC was used in the battery system. In the vehicle part, rolling resistance, aerodynamic resistance, and grading resistance was considered on the vehicle specifications. Power management strategy of the hybrid electric bus was constructed by the rule based strategy as in reference [4]. The rule based strategy was designed by the results of the backward simulations. Detailed management strategy of this system followed ref. [4].
The CCU module was designed for the dry clutch control algorithm, which are presented in the next section. Up-down shift patterns were applied to the TCU module. The driver model was constituted by accelerator signal and brake signal.

3 Development of shifting maps

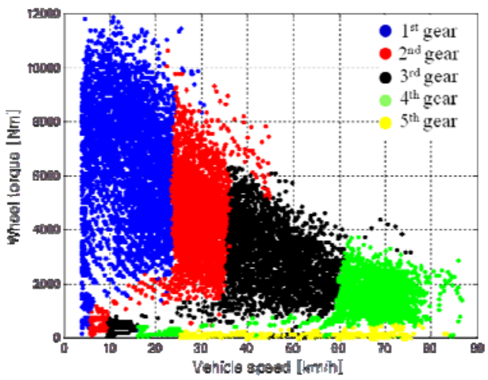
For gear shifting simulation, the shift maps for the hybrid electric bus were derived by the analysis results of DP (Dynamic Programming) theory that is one of the optimal control methods. Fig.3 shows five speed shift maps for pure electric and hybrid mode calculated by using 18 driving cycles among PSAT driving cycles. Used driving cycles are presented in Table 1.
The shift maps from DP results assumed the up-shift maps, because the results were derived from the optimal operating points of gear shifting. The down shift maps assumed -20 kph from the up-shift maps for hysteresis.

Table1: Used driving cycles in backward simulation

No.	Driving cycle	No.	Driving cycle
1	Bus TMB Line24N_1	10	Bus VH Turnhout_Empty_1
2	Bus TMB Line24N_2	11	Bus VH Turnhout_Empty_1
3	Bus TMB Line24S_1	12	Bus VH Turnhout_Full_1
4	Bus TMB Line24S_2	13	Bus VH Turnhout_Full_2
5	Bus TMB Line44E_1	14	Bus VH Turnhout_Medium_1
6	Bus VH Brusseles_Full_1	15	Bus VH Turnhout_Medium_2
7	Bus VH Brusseles Full_2	16	Bus VH Brusseles_Empty_1
8	Bus VH Brusseles Medium1	17	Bus VH Brusseles_Empty_2
9	Bus VH Brusseles Medium2	18	Decade Jumper_BCN_City_3



(a) EV mode



(b) HEV mode

Figure 3: Shifting maps for EV mode and HEV mode

4 Clutch control system

The driveline model for the clutch control system was designed as shown in Fig.4. Synchronizer model was omitted for a simplicity model.

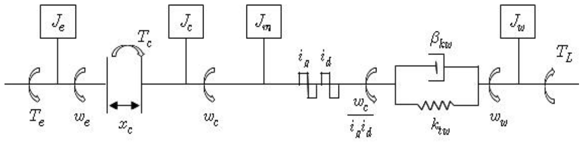


Figure 4: Driveline model

Based on this model, when the engine flywheel and the clutch disk are in the slipping operating conditions, the motion of the driveline can be described by the following equations:

$$J_e\dot{\omega}_e = T_e - T_c(x_c) \tag{1}$$

$$J_c\dot{\omega}_c = T_c(x_c) - k_{cm}\Delta\theta_{cm} - \beta_{cm}(\omega_c - \omega_m) \tag{2}$$

$$J_{eq}(i_g, i_d)\dot{\omega}_m = k_{cm}\Delta\theta_{cm} + \beta_{cm}(\omega_c - \omega_m) - \frac{1}{i_g i_d}\left[k_{tw}\Delta\theta_{mw} + \beta_{mw}\left(\frac{\omega_m}{i_g i_d} - \omega_w\right)\right] \tag{3}$$

$$J_w\dot{\omega}_w = k_{tw}\Delta\theta_{cw} - \beta_{cm}\left(\frac{\omega_m}{i_g i_d} - \omega_w\right) - T_L(x_w)$$

$$\Delta\dot{\theta}_{cm} = \omega_c - \omega_m \tag{4}$$

$$\Delta\dot{\theta}_{mw} = \frac{\omega_m}{i_g i_d} - \omega_w \tag{5}$$

In these equations, J means inertias; ω speed; T torque; x_c the throwout bearing position; and the subscripts e, c, m and w indicate engine, clutch, mainshaft, and wheels, respectively. Moreover, i_g is the gear ratio, i_d is the differential ratio, and T_L is the load torque, θ's are angles, k's are stiffness coefficients, and β's are friction coefficients. When the clutch is engaged, the engine speed ω_e and the clutch speed ω_e are equal. The engaged model can be obtained by simplified equations with the assumptions $\omega_e = \omega_c = i_g i_d \omega_w$ and $k_{tw} = \infty$. The clutch torque can be estimated by inverse of (1) as follows

$$\widetilde{T}_c = T_e - J_e\widetilde{\dot{\omega}}_c \tag{6}$$

From eq.(1), target engine torque can be calculated as follows

$$T_e^{ref} = J_e\dot{\omega}_e + T_c \tag{7}$$

AMT operating phases are possible to classify four different phases: disengagement, gear shifting, slipping, and engagement. Among these phases, this paper focused on slipping phase, which is engaging time after gear shifting. A clutch release

travel was controlled by the estimated clutch torque from the feedback loop based on vehicle dynamic equations. PI gain for the feedback loop can be obtained by the difference value between the clutch torque and the estimated clutch torque.
The constraint condition on the clutch engagement was considered to avoid the engine stall condition. For this condition, the engine speed was set above 500 rpm as the engine stall speed.
In addition, the wear of clutch pad is caused by frequent engagement and disengagement of a clutch. It is important to check the wear of a clutch because it leads to change the clamping force of a clutch.
The clutch contact point learning algorithm was also designed by using the changes of clutch torque along the clutch release travel. In the start up condition, as the clutch release travel is gradually moving, the clutch torque is going to be changed. When there is large change of clutch torque, then that point is the contact point on which flywheel disk and clutch pad met each other. Operating condition for the learning algorithm can be determined by the number of dis/engagement of clutch, and several clutch map data according to the degree of wear are added. As the clutch pad wears, proper clutch map would be selected. Variations of transmitted clutch torque according to the clutch release position are shown in Fig. 5.

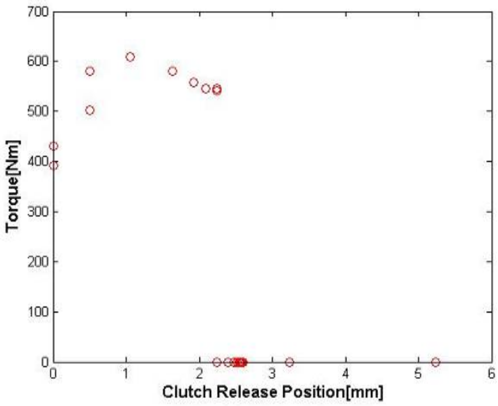
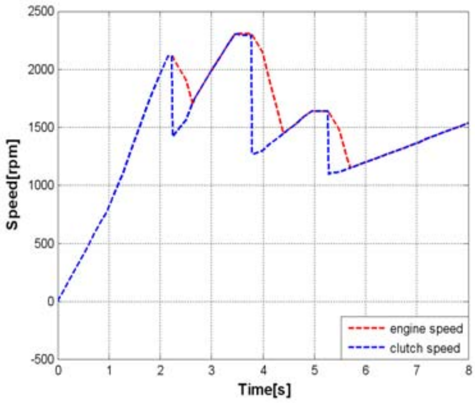


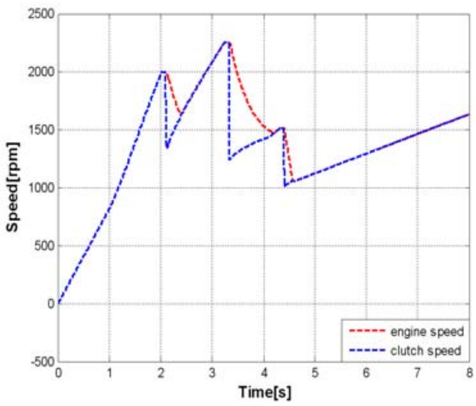
Figure 5: Variation of clutch torque on the contact point

5 Simulation results

Simulations were performed to analyze the dynamic performance of the proposed clutch control algorithm during shifting from 1st gear to 4th gear. Accelerator pedal signal in the driver module was set to the constant value 80%. In order to compare the simulation results of the hybrid bus with the clutch control, simulation of the one with only the clutch lock-up control was conducted. This condition called the uncontrolled model. Simulation results for the clutch control and the uncontrolled model were shown in Figs.5 and 6.
Fig.5 shows the results of the simulation comparison for a sequence of consecutive up-shift. As shown in Fig.5, the engine and the clutch speed followed well the engaged speed after gear shifting. The engine speed of the clutch uncontrolled model decreased more slow than that of the clutch controlled model after gear shift. Because the clutch controlled model has the engine throttle control. Slip speed of the controlled model is narrower than that of the uncontrolled model at the time of the engagement.
In addition, for a smooth clutch engagement, the clutch torque, which is controlled by release travel, operated based on the estimated clutch torque in advance as illustrated in Fig. 6(b).

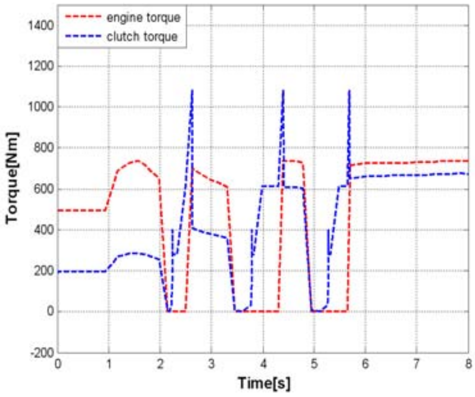


(a) The clutch uncontrolled model

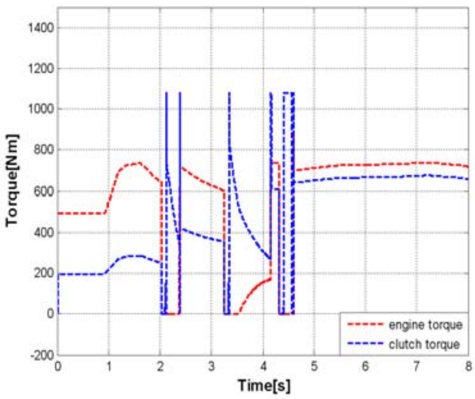


(b) The clutch controlled model

Figure 5: Simulation comparison of the engine and the clutch speed



(a) The clutch uncontrolled model



(b) The clutch controlled model

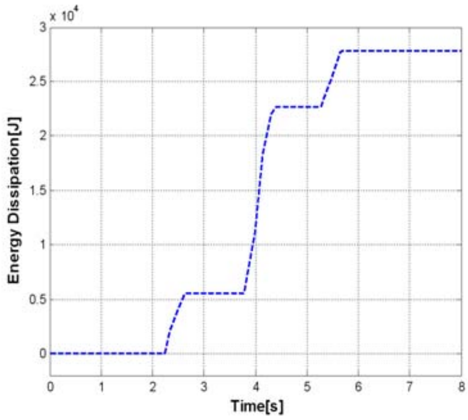
Figure 6: Simulation comparison of the engine and the clutch torque

In order to confirm the energy improvement, the energy dissipation, which was calculated during gear shift operations, was measured. The energy loss dissipated during the engagement, which can be written as

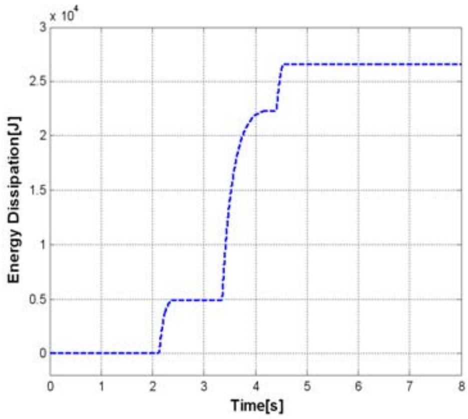
$$E_L = \int_0^t \omega_{sl}(t) T_c(t) dt$$

The slipping speed ω_{sl} is an important factor for driver comfort. Calculated energy loss during simulation is shown in Fig. 7. Table 1 shows the dissipated energy value during gear shifting. Although the dissipated energy of the clutch controlled model is larger than that of the uncontrolled model during 2-3 gear shifting, total dissipated energy is small.

As a result, the designed clutch control model reduced the energy dissipation, which affects the fuel efficiency, in the amount of 4.5% compared with the uncontrolled model.



(a) The clutch uncontrolled model



(b) The clutch controlled model

Figure 7: Results of the dissipated energy

Table1: Energy dissipation during gear shifting

Energy dissipation	Uncontrolled model	Controlled model
1-2 gear shift (J)	5510	4890
2-3 gear shift (J)	17124	17386
3-4 gear shift (J)	5175	4328
Total dissipated energy (J)	27809	26604
Improvement of energy loss	-	4.5%

6 Conclusions

In this paper, a dry clutch control of AMT systems was presented to apply to the hybrid electric bus. The clutch control system using the engine throttle and the estimated clutch torque was proposed. This control algorithm was designed by the feedback loop based on the motion of the driveline. It was tested in vehicle simulation by using Matlab simulink and simdriveline.

Simulation results show the effectiveness of the proposed clutch control system during gear shifting. Performance was evaluated in terms of duration of gearshift, dissipated energy. The clutch controlled model reduced the dissipated energy in the amount of 4.5% compared with the uncontrolled model. To obtain exact gain and the estimated clutch torque, it is suggested to investigate the various effectiveness of the proposed control in further research.

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