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Engine Clutch Pressure Command Control for a Parallel Hybrid Vehicle at Launching When Traction Motor Failed

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Abstract

A parallel type HEV (Hybrid Electric Vehicle) has a power transfer device that is constructed by wet clutches and the device is called ‘Engine Clutch’. The Engine Clutch is located between engine and electric traction motor so it transfers power of engine when the engine is started and clutches are engaged or cut the power when regenerative braking is occurred. When a traction motor of a parallel type HEV (Hybrid Electric Vehicle) could not generate torque to launch, the HEV should launch only by engine power. Because the engine should rotate at the stated idle speed to prevent the engine stop the relative rotational speed of clutch input and output shaft starts from engine idle speed and ends to zero when clutches engaged. In this study a method for control pressure command of engine clutch is proposed to satisfy desired clutch engagement time and decrease shock. A dynamic model of a parallel type HEV powertrain (including hydraulic clutch control components) is constructed to validate the method proposed.

Keywords: modelling, parallel HEV, powertrain, simulation

1 Introduction

A parallel type HEV (Hybrid Electric Vehicle) has wet clutches and one set of wet clutches is called ‘Engine Clutch’. The Engine Clutch is located between engine and electric traction motor so it transfers power of engine when the engine is started and clutches are engaged or cut the power when regenerative braking is occurred [1,2]. Typically a parallel type HEV starts to move by traction motor power. However, sometimes the vehicle should launch only with engine power when the traction motor could not generate enough power to move large mass of vehicle because of reasons like those too low SOC (Stat of Charge) of battery or physical defections of hardware. In this situation relative rotational speed of clutch input (engine side) and output (transmission side, it can be treated as wheel side after gear ratio) is equal as engine idle speed because the engine should maintain its idle speed to prevent engine stop.

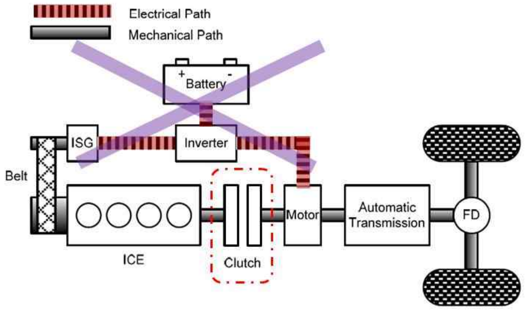

Figure1: power flow at launch start operation

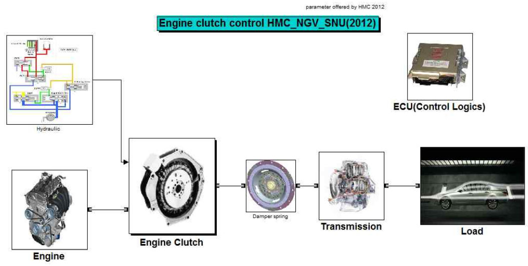

Figure 2: A parallel type HEV dynamic model

Because the relative rotational speed is quite large to engage clutches directly, the power of engine transfers to wheels and relative speed of clutches decreases by slipping clutches. The Engine Clutch is a kind of wet clutch devices which is controlled by hydraulic pressure. So the applied pressure to piston should be controlled properly to satisfy target engagement time performance and to decrease shock. In this study a control method of pressure command of Engine Clutch to satisfy engagement time performance is proposed. A dynamic model of a parallel type HEV powertrain was constructed by MATLAB/Simulink to validate the method proposed. The simulation result of dynamic model was also compared with experimental data of a test vehicle to validate the model.

2 Dynamic Modelling and simulation

A dynamic model of a parallel type HEV was constructed by MATLAB/Simulink to simulate 'Launch start operation'. As it mentioned in introduction, when a traction motor of a parallel type HEV could not generate torque to launch, the HEV should launch only by engine power.

A schematic view of the HEV model is represented in Figure 2. Essential component of HEV powertrain i.e., engine, gearbox, and Engine Clutch, load, control logic has been considered in the model. Also equivalent hydraulic components were constructed to simulate wet clutch operation by using Simulink/Simhydraulics library.

Following assumptions was considered to simulate launch of the vehicle.

- Gear ratio is fixed as 1st step during Engine Clutch engaging process.
- Engine speed is controlled at its idle rpm to prevent engine stop by Engine Control.
- Accelerator pedal signal is fixed as 25% by a driver model.

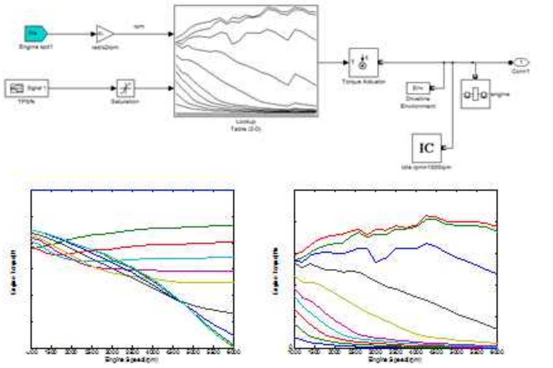

Figure 3: Engine model

2.1 Engine model

Engine model is constructed based on look up table model. Output torque performance data and friction torque loss data according to input throttle present and rotational speed are included in the engine model. Input signal of engine model is accelerator pedal present from driver's manipulation and feedback of engine speed. Engine model is presented in Figure 3.

2.2 Clutch model

Clutch transfer torque is determined by friction coefficient, friction area, effective radius, and normal pressure as seen in equation (1).

$$T_{cl} = \mu F_n = \mu NAPR_{eff},$$

$$\text{where } R_{eff} = \frac{2(R_0^3 - R_i^3)}{3(R_0^2 - R_i^2)} \qquad (1)$$

Where μ is friction coefficient, N is fiction face number, A is area, P is applied pressure, R_{eff} is radius [3~7]. The friction coefficient of a wet clutch changes with slip speed and temperature [3,4,5] Therefore friction coefficient is treated as a function of slip speed and temperature in this modelling. Friction characteristic of the engine clutch is represented in figure 4.

Wet clutch is controlled by hydraulic pressure. Hydraulic pressure command is determined by engine clutch control logic. In the launch start operation, the control logic calculates required engine torque and then torque that should be transferred by engine clutch is determined. Next, required pressure is calculated by equation (1) from engine clutch torque. Finally pressure command is transferred to hydraulic pressure control devices and actual pressure applying to clutches is determined by hydraulic components.

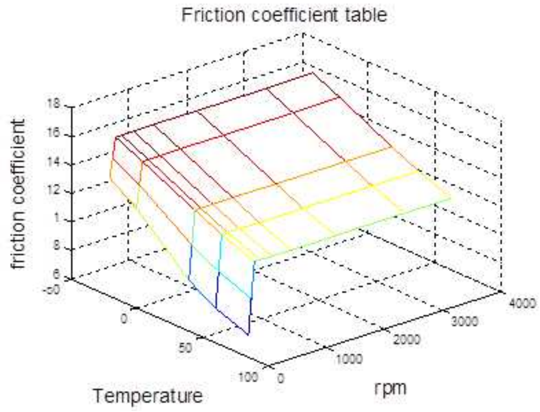

Figure 4: Friction characteristic of wet clutch

2.3 Hydraulic components

As it mentioned in 2.2, wet clutches are controlled by hydraulic pressure. Actual pressure applying to clutches is determined by hydraulic components. Pressure command signal that is determined by engine clutch control logic and actually applying pressure have difference because of hydraulic characteristic. Hydraulic oil performance is affected by temperature [6, 7]. Dynamic response to command signal can be delayed because of change of viscosity. Therefore dynamic analysis model of hydraulic components is required. Hydraulic model and result of dynamic analysis are represented in Figure 5.

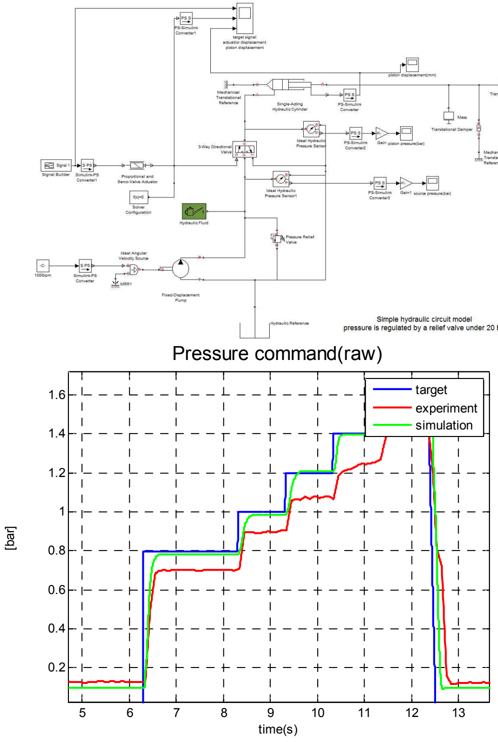

Figure 5: Hydraulic component modelling

Dynamic model of hydraulic components is constructed by Simulink /Simhydraulic library.

2.4 Driving Resistance (Load)

In this dynamic model motion of vehicle is assumed as one direction i.e., longitudinal motion. Therefore rolling resistance and aerodynamic resistance, gravitational resistance are applied while a vehicle is driving by following equations (2).

$$
\begin{aligned}
F_{roll} &= fmg\cos(\theta) \\
F_{aero} &= \frac{1}{2}\rho C_d A_F V^2 \\
F_{grav} &= mg\sin(\theta)
\end{aligned}
\qquad (2)
$$

Where f is rolling resistance coefficient, θ is degree of slope, A_f is projection area, ρ is air density, V is vehicle speed [1,2]

2.5 Validation

The simulation result of the dynamic model was compared to experimental data of test vehicle in launch start operation. Total engaging time and speed of engine clutch input and output shaft are well matched.

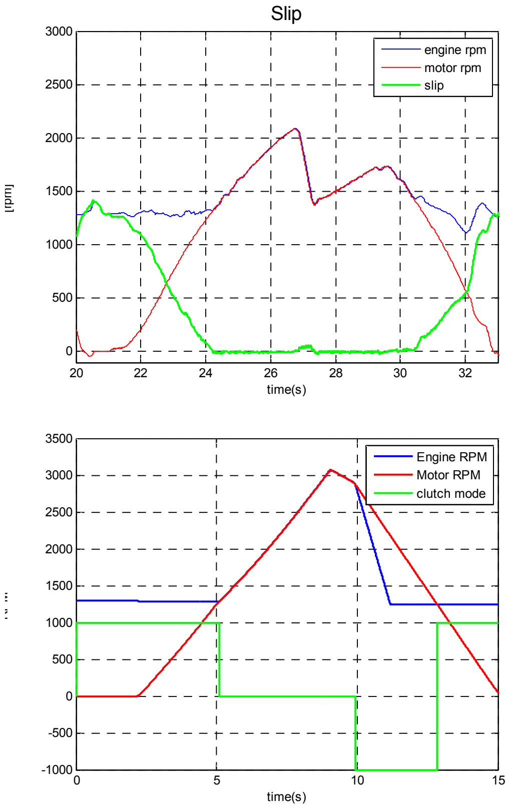

Figure 6: Friction characteristic of wet clutch

Table 1: Target engagement time of Engine Clutch

	Target engagement time(sec)	Rotational speed at engagement
APS 0%	4s	1300rpm
APS 25%	2s	1300rpm
APS 50%	1.2s	1300rpm
APS 100%	1s	1300rpm

3 Pressure Command Control Method

As it mentioned in 2.2 and 2.3, actual pressure applied to engine clutch plates is determined by dynamic response of hydraulic components according to pressure command and it can be changed by driving condition (i.e., temperature and etc.). Controllable parameter to control clutch behaviour is limited by pressure command. Thus pressure command should be determined to satisfy target engagement performance presented in Table 1. Pressure command baseline is determined according to driver's torque demand. After a launch operation pressure command is modified to decrease errors defined as the difference between target speed profile which is calculated by table1 and dynamic response of the HEV model. Five base points of pressure were selected to modify after a launch operation which is presented in Figure 7. The gain of modified pressure point is determined by sensitive analysis with various gain of each point. If performance fails to meet target performance though pressure command was modified, the pressure command should be modified by learning difference between target and actual performance [6].

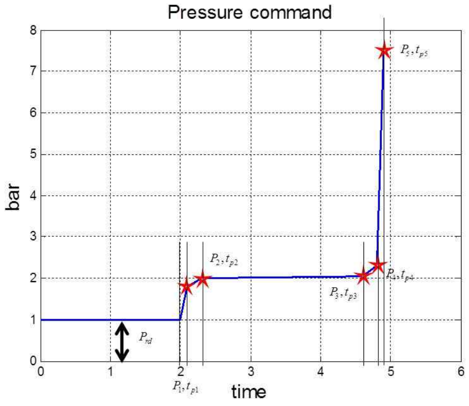

Figure 7: Base pressure line and five base point to be modified

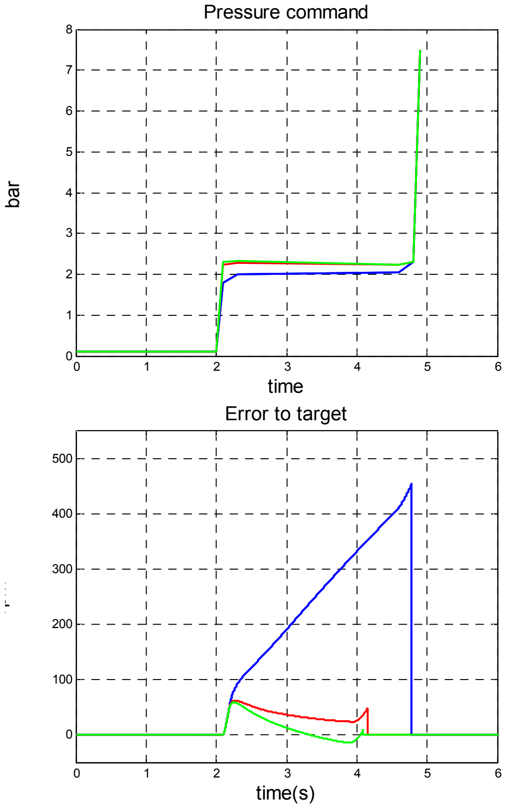

Figure 8: (a) Modified pressure command after 1^{st} and 2^{nd} launch and (b) speed errors decreased.

4 Simulation results

HEV dynamic model simulation results of launch operation after pressure command tuning is represented in Figure 8. Blue line in (a) is base pressure line and red line is modified pressure command after 1st launch. Also green line is modified pressure command after 2nd launch. Each line in (b) is corresponding speed error of output shaft speed to the target. As seen in (b) averaged error decreased and engagement time also decreased after modify pressure command.

5 Conclusion

In this study pressure command control method of Engine Clutch in a parallel type HEV was proposed. A dynamic model of a HEV powertrain was constructed by MATLAB/Simulink and simulations of launch operation only with engine power were performed. Five points in pressure command were selected and after a launch operation the points were modified to decrease speed error. Thus the target engagement time was satisfied

Acknowledgments

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