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Modeling and Design of Hybrid Control System for Dual Hybrid Electric Vehicle Drivetrains

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This paper describes the modeling of dual hybrid electric vehicle drivetrain and proposes a hybrid control system for controlling the drivetrains. In dual hybrid electric drivetrains, the energy from the engine passes through the planetary gear set and is split into the generator and motor paths. A complete dual hybrid electric drivetrain system model is developed. The modeling process is discussed for each of the major components of dual hybrid electric drivetrain, such as planetary gear transmission, gasoline engine, motor, generator and vehicle dynamics. Integrated nonlinear model and effects of parameter variations are also studied. The hybrid control system which is a discrete-event system interacting with a continuous-state system, is suitable for modeling and control of the systems that have state jumps and dynamics changes. In this paper, on/off state of engine is treated as a discrete state of HEV system and, velocities and torques, etc. as continuous states. The proposed hybrid control system consists of a continuous-state plant to be controlled, a continuous-state controller, an interface, and discrete-state controller. Simulation results are also presented comparing the performance of the proposed hybrid control system with that of the conventional HEV controller.

Keywords: Hybrid Electric Vehicle Drivetrain, Hybrid Control System

INTRODUCTION

A Hybrid Electric Vehicle (HEV) is a vehicle that has two sources of motive energy. Although there are many hybrid systems, it commonly refers to the system using gasoline engines in combination with batteries. HEVs have several advantages over conventional internal combustion engine (ICE) vehicles including improved fuel efficiency, while emissions are greatly decreased. Most auto manufacturer has been trying to develop efficient and cost-effective HEV drivetrain configurations. Some of those are parallel, series, and dual-type drivetrains.

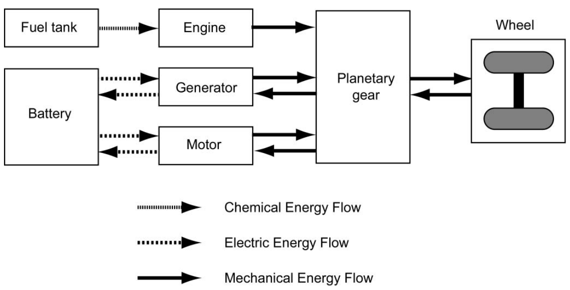

figure 1. Energy flow of dual hybrid electric vehicle drivetrain

Particularly, the dual-type HEV drivetrains use 2 electric machines together and can generate much torque than other configurations. Moreover, the dual-type HEV drivetrains have 2 degree-of freedom and we can control all of the output torque, wheel speed and energy flow simultaneously if there are the appropriate structures and controller. Hence, it is possible to provide Continuously-Variable-Transmission (CVT) ability without an additional transmission system. But the system and controller design become more complicated. Toyota has developed the world's first mass-produced, HEV, *Prius,* based on the dual-type HEV drivetrain.

In this paper, a new controller design methodology for dual-type HEV drivetrains based on Hybrid Control System (HCS) framework is proposed. For this purpose, a complete drivetrain system model is developed. Hybrid control system is a discrete-event system interacting with a continuous-state system and is suitable for the dual-type HEV drivetrain because it has discontinuous states and the change of the system dynamics.

MOELING OF DUAL HYBRID DRIVETRAIN

The structure and energy flow relationship of a dual-type hybrid electric vehicle drivetrain is shown in fig.1. It consists of a gasoline engine, two electric machines, a planetary gear set, and a battery unit, etc. The mechanical connection between these components is shown in fig.2.

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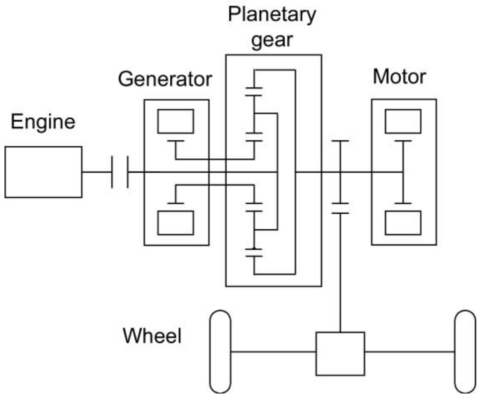

figure 2. Mechanical structure of dual hybrid electric vehicle drivetrain

The primary energy source is a fuel tank and the secondary energy source is a battery. The gasoline engine provides torque to drive the wheels. Two electric machines (a generator and a motor) and the engine are connected through a planetary gear set.

The planetary gear is shown in fig3. Because the planetary gear has two degrees-of-freedom, we can control both of the wheel speed and the engine shaft speed using toque of two electric machine and engine. It means that the planetary gear and two electric machines can replace a conventional mechanical Continuously Variable Transmis -sion (CVT). A typical mechanical CVT controls both of the engine speed and the speed of wheels by changing the gear ratio continuously.

Moreover, the gasoline engine can be operated in the optimal speed regardless of the wheel speed or required power to drive the vehicle while the engine speed cannot remain at the optimal operating point in the conventional mechanical CVT as we will explain in the next section.

Optimal Engine Operation

Typical operating-performance graph of a gasoline engine is shown in fig 4. The contour level-curves (solid) are the equal-power-per-fuel operating points. *P0* and *P1* curves are the constant engine-output power lines, that is, the power-output of engine is the same (*P1*) at the operating points *A*, *G3*, *G4*. The optimal operating curve is the set of operating points where the output power per unit fuel consumption is maximum at the same power output. In the case of power output *P1* is required, the operating point *A* is the optimal operating point rather than *G3* or *G4*.

The speed ratio of wheels and engine shaft remains several discrete values in the drivetrain where the conventional transmission is installed. Four possible engine speed values at a specific fixed wheel speed are shown as dashed lines in the fig.4. When the power *P1* is required, we need to use the 3rd gear or the 4th gear and make the engine operating at the points *G3* or *G4* in this figure. CVT has infinite number of gear ratio and can

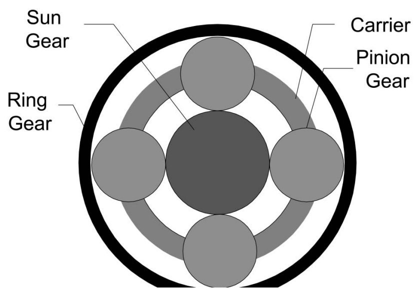

figure 3. Planetary gear set

make the engine operate at the operating point *A* which is more economic operating point than *G3* or *G4*.

Dual hybrid electric vehicle drivetrain with planetary gear and two electric machines can have another advantage over the conventional CVT. It can be seen that the operating point *O* is the optimum operating point in the whole range of operation. But, with the conventional CVT, we cannot make the engine at the point *O* when the power *P1* is required because the output power at the point *O* is *P0*, which is different from *P1*. In a dual hybrid electric vehicle drivetrain, however, it is possible. The redundant power *P1-P0* can be used charging the battery. If more power *P2* is required than *P0*, the additional power *P2-P0* comes from the battery.

COMPONENS MODELLING

Major components modeling are explained in detail below

Planetary Gear Modeling

A planetary gear set consists of a sun gear, a ring gear, a carrier, and pinion gears (see fig.3). Although there are several component connection configurations in the literature [1,2], the configuration in fig.2 are studied here. The sun gear and ring gear are connected to the electric machines respectively and the carrier is connected to the engine.

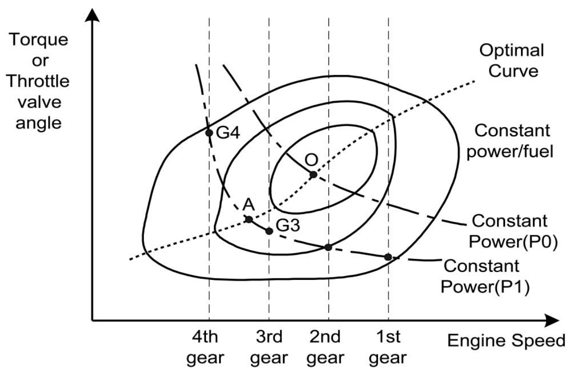

figure 4. Typical operation-performance graph of gasoline engine

Three operation modes are possible in the use of the planetary gear set in this configuration. In a normal mode, the gear has two degrees-of-freedom and ring gear and sun gear are freely rotating. In a fixed-carrier mode, a mechanical clutch prevents the carrier and the engine shaft from rotating. Similarly, The sun gear shaft does not rotate in a fixed-sun-gear mode. If we ignore the inertia and friction of the gear set, the dynamics equation of planetary gear set is

$$\begin{aligned} T_S + \frac{\alpha}{\alpha+1} T_C &= 0 \\ T_R - T_L + \frac{1}{\alpha+1} T_C &= 0 \end{aligned} \tag{1}$$

where T_S, T_R, T_C, T_L are torques to the sun gear, the ring gear, the carrier, and the load torque, respectively. The dynamics equations when we consider the inertias of the gears are

$$\begin{pmatrix} \omega_S \\ \omega_R \\ \omega_C \end{pmatrix} = A \begin{pmatrix} T_S \\ T_R \\ T_C \end{pmatrix} \tag{2}$$

The inertia matrix A changes according to the operation mode of the planetary gear set.

$$\begin{aligned} A_{\text{normal}} &= \frac{1}{J_1 J_3 - J_2^2} \begin{pmatrix} J_3 & -J_2 & \frac{\alpha J_3 - J_2}{1+\alpha} \\ -J_2 & J_1 & \frac{J_1 - \alpha J_2}{1+\alpha} \\ \frac{\alpha J_3 - J_2}{1+\alpha} & \frac{J_1 - \alpha J_2}{1+\alpha} & \frac{\alpha^2 J_3 - 2\alpha J_2 + J_1}{(1+\alpha)^2} \end{pmatrix} \\ A_{\text{Carrier Fixed}} &= \frac{1}{\alpha^2 J_3 - 2\alpha J_2 + J_1} \begin{pmatrix} 1 & -\alpha & 0 \\ -\alpha & \alpha^2 & 0 \\ 0 & 0 & 0 \end{pmatrix} \\ A_{\text{Sun Gear Fixed}} &= \frac{1}{J_3} \begin{pmatrix} 0 & 0 & 0 \\ 0 & 1 & \frac{1}{1+\alpha} \\ 0 & \frac{1}{1+\alpha} & \frac{1}{(1+\alpha)^2} \end{pmatrix} \end{aligned} \tag{3}$$

where

$$\begin{aligned} J_1 &= \frac{\alpha^2}{(1+\alpha)^2}(J_C + M_P R_C^2) + \frac{\alpha^2}{(1-\alpha)^2} J_P + J_S \\ J_2 &= \frac{\alpha}{(1+\alpha)^2}(J_C + M_P R_C^2) - \frac{\alpha}{(1-\alpha)^2} J_P \\ J_3 &= \frac{1}{(1+\alpha)^2}(J_C + M_P R_C^2) + \frac{1}{(1-\alpha)^2} J_P + J_R \\ \alpha &= \frac{R_S}{R_R} \end{aligned}$$

Wheel and Vehicle Dynamics

The simplified wheel and vehicle dynamics are used for the control purpose [3,4]. When the deferential dynamics are ignored, the torque to the ring gear of the planetary gear set is

$$T_{out} = m(\frac{r}{R_d})^2 \frac{d\omega_R}{dt} + mg\sin\theta \frac{r}{R_d} + \frac{1}{2}\rho C_D S (\frac{r}{R_d})^3 {\omega_R}^2 + mgC_r \frac{r}{R_d} \tag{4}$$

where R_d is the final reduction gear ratio, r, tire radius, θ, slope angle, ω_R, angular speed of ring gear shaft, C_d, aerodynamic drag coefficient, m, mass of vehicle, C_r, rolling friction coefficient.

Each term in the above equation means the acceleration load, the slope load, the aerodynamic resistance, and the tire rolling friction.

Gasoline Engine Modeling

A gasoline engine is a highly nonlinear system with multi input such as throttle valve angle, spark advance, angular speed of engine shaft, fuel flow, etc., and its modeling and validation are complicated. We used an experiment-based quasi-static model for a gasoline engine [5]. We assumed the torque output and the fuel consumption rate are dependent on the throttle angle valve and the angular speed of the engine shaft. Moreover, provided the decentralized engine controller keeping track of optimal engine torque line shown in fig.4., we have the following experiment-based dynamics equation.

$$\tau_E \dot{T}_E + T_E = f(\omega_E)$$

$$\tau_E \ddot{m}_E + \dot{m}_E = g(\omega_E)$$

$$\begin{aligned} f(\omega_E) &= a_4\omega_E^4 + a_3\omega_E^3 + a_2\omega_E^2 + a_1\omega_E + a_0 \\ g(\omega_E) &= c_4\omega_E^4 + c_3\omega_E^3 + c_2\omega_E^2 + c_1\omega_E + c_0 \end{aligned} \tag{5}$$

Electric Machines Modeling

Because dynamic of electric machines is even faster than that of engine, we assume that the torque output of the electric machines can follow the torque command immediately. There is also torque limit constraint dependent upon the angular velocity of the electric machine. The efficiency map in fig.5 is used to calculate the energy flow.

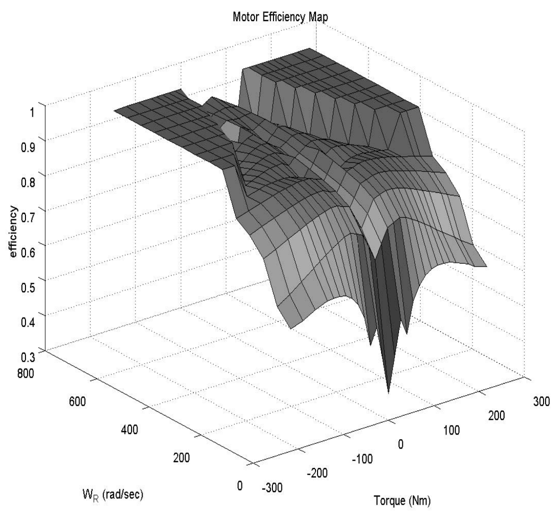

figure 3. Efficiency map of electric machine

CONTROLLER DESIGN USING HYBRID CONTROL SYSTEM FRAMEWORK

Overall dynamics of the dual hybrid electric drivetrain is highly nonlinear and, when the mode of planetary gear set changes, the dynamics equation also changes abruptly. In the literature [6,7], heuristic and experiment-based controller design methodology has been used. In this paper, more systemic and analytic controller design method which is based on the hybrid control system is presented.

The hybrid control system, a discrete-event system interacting with continuous-state system, is suitable for modeling and control of the systems that have state jumps and dynamics changes[8]. On/off state of engine and the mode of planetary gear set can be treated as a discrete state of HEV system and velocities and torques, etc. as continuous states. The framework for the hybrid control system design in [9] was modified and used here.

HYBRID CONROL SYSTEM COMPONENTS

The structure of hybrid control system is shown in fig.6.

Continuous State Plant

Continuous state plant is the system to be controlled, that is, the hybrid electric vehicle drivetrain in this paper. Previous sections described the modeling of continuous state plant in detail. There are two operation modes. Mode *0* is the engine-off mode. Engine shaft is fixed by a mechanical clutch and driving torque is generated from two electric machines in mode *0*. Mode *1* is engine-on mode. The planetary gear set has two degrees-of-freedom and torque is generated from both of the gasoline engine and electric machines

Continuous State Controller

Continuous state controller generates continuous control input *u* out of the continuous states from continuous plant and controller mode command *i* and the reference *r* from interface. Two continuous state controllers for mode *0* (engine off) and mode *1*(engine on) are selected according to the controller mode command *i.*

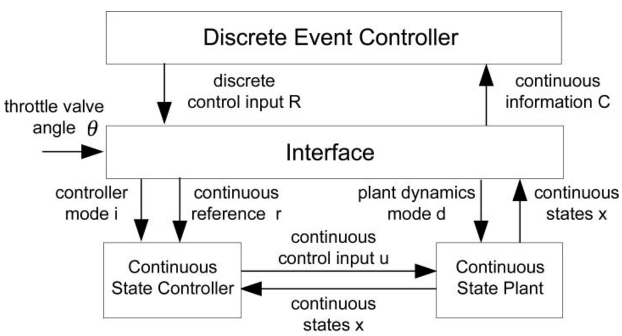

figure 4. Structure of hybrid control system for dual hybrid electric vehicle drivetrain

Interface between Continuous-Discrete Systems

Interface consists of two parts. One is continuous-discrete interface and the other is discrete-continuous interface.

A discrete-continuous interface generates controller mode command *i* and the reference *r* out of the driver's throttle valve angle command and discrete control input *R* from discrete event controller.

Discrete Event Controller

Discrete event controller coordinates the overall system behaviors and decides the mode of the continuous state controller.

CONTROLLER DESIGN

Mode 0 Continuous State Controller Design

In mode 0, the actuators for generating torque is two electric machines. Hence, the design objective for mode 0 continuous state controller is to find an optimal torque dependent on the vehicle speed and driving torque command that minimize electric power consumption under the torque limit constraints that depends on the angular speed of electric machines. Result of the computational optimization using practical parameter information was shown in fig.7.

Approximate optimal motor torque equation is

$$T_M(\omega_R, T_{OUT}) = 1.0127 T_{OUT} + 0.4086\omega_R + 0.0014 \quad (6)$$

which is very similar to the motor control law in [6].

Mode 1 Continuous State Controller Design

The objective of the continuous state controller in mode *1* is tracking the reference torque command. Robust linear control law for systems with parameter uncertainties [10] is used in this paper. Gasoline and vehicle dynamics are time-varying nonlinear system and robust design is required for guaranteeing stability of the system. First, continuous state plant is linearized at a nominal operating

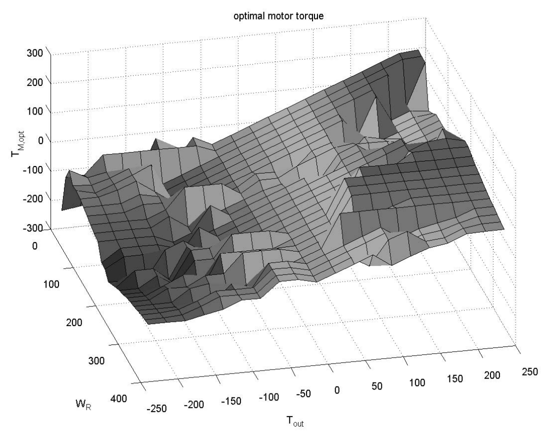

figure 5. Optimal motor torque map

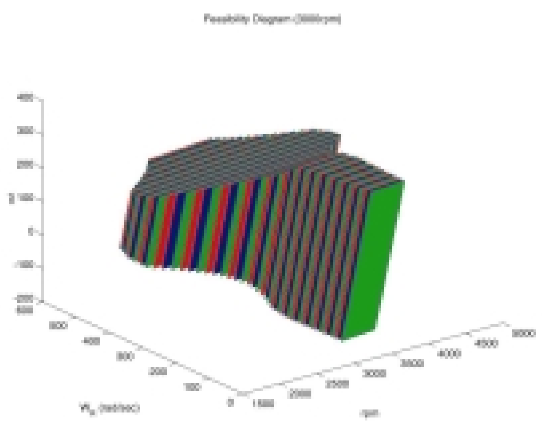

figure 6. An example of feasible reference region (rpm = 3000)

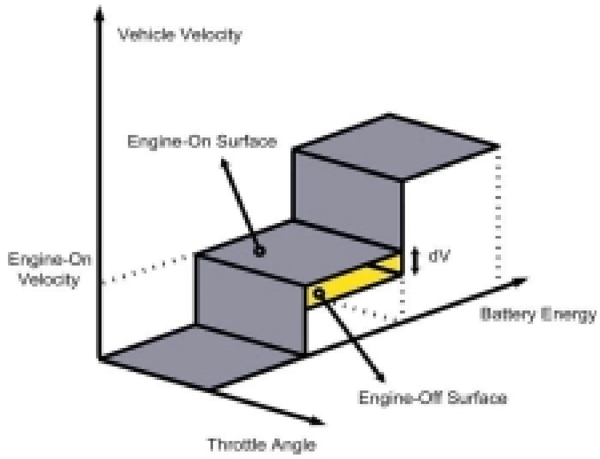

figure 7. Event-generate-surface

point and uncertainty bounds for the parameters are obtained.

For the linear system and uncertainties, robust linear control law is calculated. Last, feedforward torque correction is added to compensate the stead-state error.

Interface Design

Major role of interface part is to generate reference command and to pass the information between the discrete event controller and the continuous controller–system.

Generating appropriate reference command is important because too large values for the reference may cause instability problem due to the torque limit constraints of electric machines. As one of the solution to avoid this problem, reference modification method is proposed in this paper. For a step reference command, the controlled system shows that the time response of the control input is about 20 times faster than those of states. It means that saturation of the control inputs occurs immediately if the reference torque command has too large value. Hence we can calculate the region of the reference command that does not cause the control input saturate. An example of feasible reference command region is shown in fig. 8. Reference modification method changes reference command if it goes out of the feasible region to prevent the instability caused by the control input saturation.

Discrete Event Controller Design

Discrete event controller coordinates the discrete state of the hybrid system. Whether the engine is operating or not is the discrete state for a dual hybrid electric vehicle. An event that the change of discrete state occurs when some of the continuous states such as the vehicle velocity, battery energy, and throttle valve angle meet a specific conditions. These sets of continuous states are called an event-generate-surface. The event-generate-surface used in this paper is shown in fig. 9.

Engine begins to operate when the vehicle velocity reaches engine-on speed or battery energy becomes too low.

The engine-off velocity is lower than the engine-on velocity by *dV*. Successive engine on-off phenomenon arises if the engine-off velocity equals to the engine-on velocity because the vehicle decelerates when the engine is turned on. Design parameters of the discrete event controller are the engine-on velocity and the velocity difference *dV*. If the driving velocity profile is given, the optimal engine-on velocity can be determined easily by using computational optimization, but, this is not the general case.

We generate a large number of different velocity profiles for city driving and calculate fuel consumption, battery energy, and the number of engine turn-on through computer simulations. One of the result is shown in fig.10.

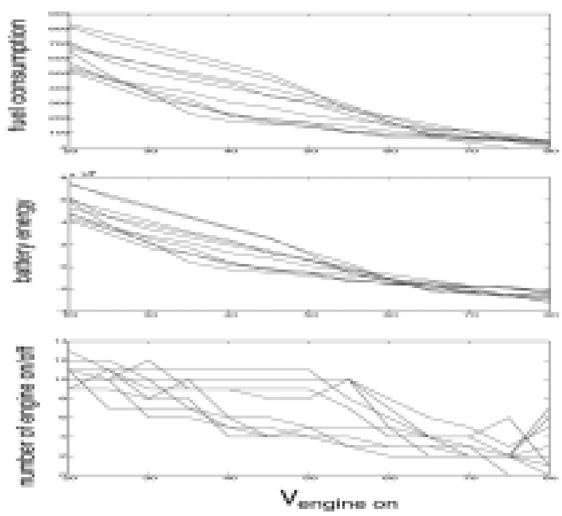

figure 8. Simulation result of fuel consumption, battery energy, and the number of engine-on/off vs. engine-on velocity. (when *dV = 10 km/h*)

We can determine the engine-on velocity from the simulation result. As a rule of thumb, to maintain the battery energy at a nominal level may be the control objective of control because we can reduce the capacity of battery and, therefore, the weight of vehicle. We have engine-on velocity of *50 km/h* for the vehicle used in this simulation.

CONCLUSION

A complete dual hybrid electric drivetrain system model is developed. The modeling process is discussed for each of the major components of dual hybrid electric drivetrain. The integrated nonlinear model and the effects of parameter variation are also studied. A new hybrid control system which consists of the continuous-state plant to be controlled, a continuous-state controller, an interface, and a discrete-state controller is proposed. For each control component and overall controller, its performances and stabilities are proved through computer simulations. The proposed design procedure shows a new controller design methodology for a dual-type hybrid electric vehicle drivetrain.

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