Computer Model for a Parallel Hybrid Electric Vehicle (PHEV) with CVT

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1. Introduction
A complete dynamic vehicle model for a Parallel Hybrid Electric Vehicle (PHEV) equipped with a Continuously Variable Transmission (CVT) is developed and presented. The PHEV has a post-transmission electric motor. The model is intended to represent the dominant dynamics relating to the control of the overall system. Both analytical and empirical techniques are used in developing the model. The modeled components include: internal combustion engine, engine clutch, CVT, electric motor, lead-acid battery, vehicle driveline, hydraulic brakes, and vehicle longitudinal and tire dynamics. A vehicle computer control module as well as component controllers, sensors and control mechanisms are also modeled. The model is implemented in the Matlab/Simulink environment. This platform, coupled with a modular model design, offers the potential to evaluate the effects of using different engines, motors, clutches, etc. on the vehicle behavior without redesigning the entire model. Simulation results are presented.

The model is a dynamic, modular, forward-type simulation. The model consists of a driver sub-model trying to follow a predetermined velocity profile, a dynamic vehicle model, and a PHEV control module that acts as the interface between the driver and vehicle models. The relationships between these components are shown in Figure 1. The PHEV control module controls the vehicle such that the driver power demands are met. It should be noted that the emphasis of this paper is on the vehicle modeling (PHEV with CVT block in Figure 1).

![Figure 1: Major Simulation Components](image)

2. Vehicle Dynamic Model
The vehicle being modeled is a post-transmission parallel hybrid vehicle [1]. In other words, the electric motor is connected to the opposite side of the transmission as the engine. This vehicle configuration is shown in Figure 2.

![Figure 2: Post-Transmission Parallel Hybrid](image)

2.1 Engine Dynamic Model
The key exogenous control variable to the SI ICE engine system is the throttle angle (or mechanical equivalent). Control variables such as spark advance, Exhaust Gas Recirculation (EGR), and Air-to-Fuel ratio (A/F), are assumed to be in the default configuration as pre-calibrated emission control variables.

The engine model [2] contains representations of the throttle body, engine pumping phenomena, induction process dynamics, fuel transient behavior, intake and exhaust breathing lags, individual cylinder constituent estimation, and rotating inertia. The induction process is somewhat complex in this instance in that it consists of a throttle body embedded between and upstream single channel plenum with air filter (or air measuring restriction) and a downstream intake plenum with a local port region near each intake valve. Simulation of the breathing apparatus will accommodate forward and reverse flow for both the air filter system and the throttle body allowing the dynamic model to be exercised continuously at or near Wide Open Throttle (WOT).

2.2 CVT Dynamic Model
The continuously variable transmission consists of two pulleys connected by a metal belt, as shown in Figure 3 [3]. The primary pulley is driven by the clutch, while the secondary pulley connects to the intermediate drive gear assembly. The ratio is varied by applying force to the pulleys, thereby changing their effective diameters. The belt is free to slide along the pulley sheaves. As with fixed gears, the CVT ratio is defined by the speed ratio, which is in this case the primary speed divided by the secondary speed.
The significant dynamic difference between the CVT and the fixed ratio transmission is apparent when taking the derivative of equation 1. Because the CVT gear ratio is now a continuous function of time, we get the following equation.

\[
\dot{\omega}_p = g_{\text{CVT}} \omega_s + \left( \frac{d}{dt} g_{\text{CVT}} \right) \cdot \omega_s
\]  

(2)

The CVT is assumed to be 100% efficient. By conservation of energy, the primary torque and secondary torque are related through the gear ratio.

\[
g_{\text{CVT}} = \frac{T_c}{T_p}
\]

(3)

It should be noted that due to the efficiency assumption, the CVT does not slip. This means that the rotational speeds of the two pulleys are related at all times through the gear ratio, which itself is a controlled variable. This means that from a modeling point of view, there is only a single rotating mass, and therefore we model the CVT with a single state variable. The CVT model outputs are the rotational speeds of both the primary and secondary.

2.3 Driveline and Vehicle Dynamics

The details of the driveline are shown in Figure 4 [1]. The CVT secondary is modeled as rigidly attached to the vehicle driveshaft by way of the transmission final drive ratio, \( g_{\text{fd}} \). Driveshaft speed is calculated from the speed of the CVT secondary. Torque is transmitted from the driveshaft to the two drive wheels by means of a differential gearing system with unity gear ratio.

2.4 Electric Motor Model

The electric motor modeled here is a 56 kW, 3-phase, AC induction motor. Torque is produced through two electric fields (one on the stator, the other on the rotor) that are rotating at different speeds. The AC induction motor dynamics have the form given in equation (4) for positive torque and equation (5) for negative torque. The positive torque delivered is the minimum of the requested torque and the maximum deliverable torque at the current motor speed, multiplied by a first-order lag. The negative torque delivered is the maximum of the requested torque and the minimum positive (maximum negative) deliverable torque at the current motor speed. The eigenvalue, \( \lambda_m \), is typically quite large (fast) relative to the motor and vehicle dynamics.

\[
T_m = \min \left( T_{\text{m, request}} - \frac{\lambda_m}{s + \lambda_m}, T_{\text{m, max}}\omega_n \cdot \frac{\lambda_m}{s + \lambda_m} \right)
\]

(4)

\[
T_m = \max \left( T_{\text{m, request}}, T_{\text{m, max}}\omega_n \cdot \frac{\lambda_m}{s + \lambda_m} \right)
\]

(5)

2.5 Lead-Acid Battery Model

The battery model is a lumped parameter dynamic characterization of a lead acid battery [1]. The model is based on the following battery model diagram.

From Figure 5, the following two equations characterize the dynamic battery behavior.

\[
R_b C_p \dot{\tilde{e}}_{ib} = \tilde{e}_{ib} - e_{ib} - R_b i_{ib}
\]

(6)

\[
\tilde{R} C_p \dot{\tilde{e}}_{ib} = V_{oc} - \left( \frac{R_b + \tilde{R}}{R_b} \right) \tilde{e}_{ib} + \frac{\tilde{R}}{R_b} e_{ib}
\]

(7)

At an average discharge current of 75A and a nominal electrolyte temperature of 20°C, the open circuit voltage is a function of the battery state of charge.

\[
V_{oc} = 338.8\left[0.94256 + 0.05754(SOC)\right]
\]

(8)

3. Simulation Results

This set of simulation runs consists of following a desired ramp velocity profile from 0 to 60 mph and then decelerating back to 0 mph. Simulation data for a 15% road grade is contained in Figure 6 and Figure 7 for the PHEV with CVT. The effect of the CVT can be seen clearly looking at the engine torque (\( \text{Engine Tq (Nm)} \)) and RPM plots for these two cases. From 10 to 23.5 seconds in the simulation, the engine is engaged and RPM is held constant at 4000 RPM and the throttle angle is held at 49 degrees. This is the operating point at which the engine most efficiently produces a high level of power. Through the use
of the changing CVT gear ratio (CVT gear ratio), the engine can be operated at this point over the speed range of approximately 20 to 52 MPH. This results in a smoother halfshaft torque profile (Halfshaft Tq (Nm)) meaning better driveability, while allowing for better control of A/F transients during this period.

4. References


Figure 4: Parallel Hybrid Powertrain with CVT
Figure 5: Parallel Hybrid Powertrain with CVT

Figure 6: Simulation Results, CVT, 15% Grade - 1
Figure 7: Simulation Results, CVT, 15% Grade - 2