

Development of a Dynamic Driveline Model for a Parallel-Series PHEV

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ABSTRACT

This paper describes the development and experimental validation of a Plug-in Hybrid Electric Vehicle (PHEV) dynamic simulator that enables development, testing, and calibration of a traction control strategy. EcoCAR 2 is a three-year competition between fifteen North American universities, sponsored by the Department of Energy and General Motors that challenges students to redesign a Chevrolet Malibu to have increased fuel economy and decreased emissions while maintaining safety, performance, and consumer acceptability. The dynamic model is developed specifically for the Ohio State University EcoCAR 2 Team vehicle with a series-parallel PHEV architecture. This architecture features, in the front of the vehicle, an ICE separated from an automated manual transmission with a clutch as well as an electric machine coupled via a belt directly to the input of the transmission. The rear powertrain features another electric machine coupled to a fixed ratio gearbox connected to the wheels. The model accounts for rotational dynamics and inertias of the torque generating components, gearboxes, and wheels. Additionally it considers the effects of stiffness and damping from the belt and shafts in the system. The simulator focuses on the drivetrain dynamics while actuator dynamics are either simplified or not considered. The vehicle simulator is validated by testing on the OSU EcoCAR 2 vehicle in both subsystem and full vehicle configurations.

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INTRODUCTION

Active vehicle dynamics controls serve several important purposes including increasing the range of conditions under which a vehicle behaves predictably as well as enhancing vehicle comfort and response [5]. Traction control particularly maximizes tractive and lateral forces between the vehicle's tires and the road for acceleration and steerability. This is accomplished using control inputs and available actuators to target an appropriate driven wheel slip and allows inexperienced drivers to perform as well as or better than experienced drivers on slippery roads. [5].

The test vehicle for this work was created as a part of the EcoCAR 2 Advanced Vehicle Technology Competition. EcoCAR 2 is a three-year competition sponsored by the US Department of Energy and General Motors. Fifteen universities from across the United States and Canada are challenged to redesign a stock Chevrolet Malibu in order to obtain higher fuel efficiency and lower emissions while maintaining or improving upon the safety, performance, and consumer acceptability. The test vehicle is the redesigned Malibu developed by the Ohio State EcoCAR 2 team.

The primary method used for control development for the Ohio State EcoCAR 2 vehicle involves model-based design. Mathematical models of the powertrain components are

created and utilized allowing control algorithms to be tested prior to implementation on the vehicle. The primary model used by the Ohio State EcoCAR 2 team for control development is an energy based hybrid vehicle simulator called EcoSIM which was developed by Ohio State teams over 15 years [3]. EcoSIM contains models of each powertrain component that are based on static maps, simple transfer functions, and simplified soft ECUs [3]. While very useful for its primary purpose of developing control algorithms that improve fuel economy, the model does not account for the powertrain dynamics including inertias and shaft stiffnesses causing it to be insufficient for traction control development. To accommodate traction control development, a new model was developed based on EcoSIM, expanding upon it to take the powertrain dynamics into account. The objective of this paper is to present a control oriented PHEV dynamic model that allows for the development of a traction control system for the unique powertrain architecture of the OSU EcoCAR 2 vehicle.

Computer models of HEVs have been extensively researched throughout the past few decades. Numerous models have been made that are quasi-static energy models such as those discussed in [8], [2], and [13]. These models are useful for predicting fuel economy and performance for different hybrid configurations, however they do not include the driveline dynamics or tire friction modelling that are imperative for the development of traction control. Autonomous is a hybrid

modeling program developed through funding from the department of energy that creates a plug-and-play environment for designing and modeling various hybrid powertrains from preconfigured component models [12]. While this is a useful platform for architecture development, its focus is primarily on energy consumption and thus it also does not contain the necessary dynamics for traction control development. HEV models containing dynamics have been developed, often for use in drivability studies, such as those in [11] and [10]. These models contain a high level of detail and complexity which makes them computationally heavy and sub-optimal for development of real-time traction control. Also, only one of these models included a tire friction model capable of simulating wheel slip. These models were created for parallel-through-the-road and common power-split configurations. The main contribution of this paper is the development of a dynamic model optimized for traction control development on a unique parallel-series extended range electric vehicle capable of mode switching.

EXPERIMENTAL VEHICLE PLATFORM

The experimental vehicle is a commercially available 2013 Chevrolet Malibu that was converted to a PHEV at the Ohio State University Center for Automotive Research. The conversion was motivated by the EcoCAR 2 competition, an Advanced Vehicle Technology Competition with the primary goals of increasing fuel economy and decreasing emissions while maintaining a consumer acceptable, showroom ready vehicle.

The powertrain architecture of the converted vehicle is a parallel-series PHEV as shown in Figure 1. Vehicle components relevant to dynamic model development are:

- 1.8L Honda high compression ratio engine that can be coupled or decoupled from the transmission input via a clutch
- 82 kW permanent magnet electric machine (EM) belted directly to the transmission input
- 6-speed automated manual transmission to drive the front axle
- 82 kW permanent magnet EM connected to the rear axle through a fixed-ratio gearbox
- 18.9 kWh battery pack powering both electric machines

The vehicle configuration allows the vehicle to operate in several different modes including charge depleting, charge sustaining series, and charge sustaining parallel operation.

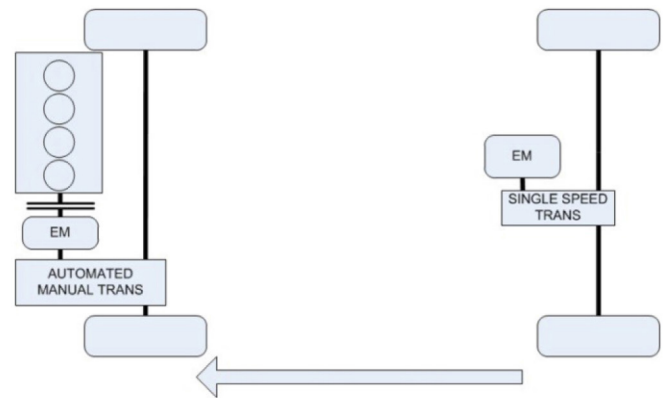


Figure 1. Experimental Vehicle Architecture

MODEL REQUIREMENTS

EcoCAR 2 Design Process

The EcoCAR 2 competition functions on a three year vehicle development process (VDP). A simplified version of the VDP is given in Figure 2. The first year consists of architecture selection and then component selection and packaging design based upon the chosen architecture. The second and third years consist of component integration and controls optimization. While shown as separate processes on the simplified diagram, in reality both are ongoing and intertwined processes throughout years two and three. The competition is currently in the early stages of year three. The Ohio State vehicle is currently in the late stages of fabrication. Once the vehicle hardware is fully realized, validation testing as described below will be used to fully identify all parameters. Prior to validation, parameters used are estimated based upon manufacturer data or similar sized components for which data is available.

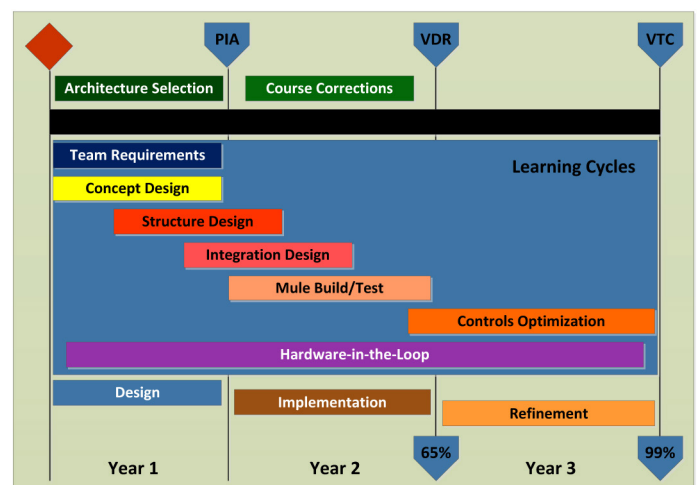


Figure 2. EcoCAR 2 Vehicle Development Process

Traction Control Philosophy

Because this model is intended for traction control development, the intended traction control strategy is an important factor to consider in model development. The Ohio State team intends to take advantage of the electric motors to control the torque to the wheels during a traction control event. The competition rules prohibit interfering with or changing the operation of the ABS system during braking so the traction control strategy will focus on traction control during acceleration and cruise. The control strategy of the vehicle utilizes a charge depleting mode for heavy acceleration in which the electric motors accelerate the vehicle while the engine is off and disconnected from the system. This is when an acceleration traction control event is most likely to occur. If a traction control event occurs while the vehicle is using the charge depleting mode, the vehicle will be held in a fixed gear that is determined to be most appropriate at the time and traction control will be applied. This improves the performance of the traction control by eliminating the disruption of the shifting event. If the battery state-of-charge is too low to allow the vehicle to function in charge depleting mode, the vehicle will enter the charge sustaining series mode for traction control. The series mode leaves the front axle with no propulsive torque input from the powertrain and the rear electric motor to control the torque transmitted to the rear wheels in order to maintain control of the vehicle.

Requirements

Based on the background given above, the requirements for the model can be described:

- High fidelity of the model to capture high frequency vibrations is not required as only lower frequency dynamics will be important for traction control
- The model must be capable of modeling slip on all four wheels and demonstrating differential action between the two wheels on a given axle
- The model must be able to run in real time for HIL simulation
- The model must be available for SIL/HIL work prior to mechanical realization of the vehicle

MODEL DESCRIPTION

Assumptions

The following assumptions were used in the development of the dynamic model, driven by the requirements listed above.

- Only longitudinal vehicle dynamics considered
- Torsional stiffness of shafts and gears except half shafts are assumed to be infinitely large
- Friction between the tires and road surface are assumed to be uniformly acting on all tires
- Environmental factors such as temperature, pressure, and humidity are not taken into consideration
- Drivetrain losses are represented by lumped efficiency and

friction models

Model Overview

A diagram of the powertrain dynamic model is given in Figure 3.

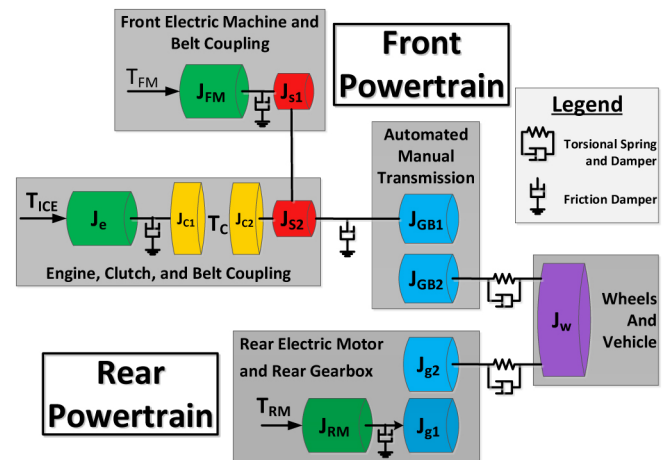


Figure 3. Dynamic Model of the Driveline

The supervisory control strategy developed for the test vehicle converts a percentage-based torque request from the pedals into actual torque requests for each of the three torque generating components based on the optimization of fuel economy and emissions. The dynamic model presented here focuses on translating the torque requests into the torque realized by the engine and motors and how it propagates through the powertrain to the wheels. The following sections will describe each of the model components in detail.

Engine

A detailed thermodynamic model describing engine dynamics does not provide substantial additional information for the validation of a traction control oriented dynamic vehicle model, particularly because the engine will not be used to transmit torque to the wheels in the intended traction control strategy as described previously. Such detailed models are often used for the design of engine control systems, however for this work a simplified representation of the engine model is used to reflect the dynamics between torque request and resulting engine speed. The engine is modeled as an ideal torque actuator with a constant inertia connected through the crankshaft to the rest of the powertrain. The resulting crankshaft dynamics are given by:

$$\dot{\omega}_e = \left(\frac{1}{J_e + J_{c1}} \right) T_e - T_{e,fr} - T_c \quad (1)$$

Where

$\dot{\omega}_e$ = engine angular acceleration

T_e = engine indicated torque

$T_{e,fr}$ = engine friction torque

T_c = torque transmitted by the clutch

J_e = lumped engine inertia

J_{c1} = inertia of the clutch flywheel attached to the engine

The engine friction is modeled by a lookup table constructed from experimental data. A delay between torque request and torque generation is included in the model to account for the induction-to-power stroke transport delay.

Front Electric Machine

Because the pulleys on which the belt is mounted are coupled directly to the electric machine and transmission, they are modeled as lumped inertias with those components. The primary pulley, denoted by the subscript s1, is coupled to the output shaft of the electric machine. Belt dynamics are ignored. Taking into account these assumptions, the dynamics of the front electric machine are given by:

$$\dot{\omega}_{FM} = \left(\frac{1}{J_{FM} + J_{s1}} \right) T_{FM} - T_{FM,fr} - T_{belt} \quad (2)$$

Where

$\dot{\omega}_{FM}$ = Front electric machine angular acceleration

T_{FM} = Front electric machine torque

$T_{FM,fr}$ = Front electric machine internal friction torque

T_{belt} = Torque transmitted by the belt

J_{FM} = Front electric machine inertia

J_{s1} = Inertia of primary belt pulley

The front electric machine friction term is modeled as the combined friction of all components at the node of the front electric machine, the belt and pulleys, and the transmission primary shaft when the engine clutch is open.

Transmission

The clutch disk and secondary belt pulley are coupled directly to the input shaft of the transmission so they are modeled as a lumped inertia. A highly detailed dynamic model of the transmission would not provide significant useful information for the development of traction control because the vehicle will be kept in a fixed gear during a traction control event so a simplified model was chosen to prevent unnecessary model complexity and maintain an efficient model based design. The

simplified model represents the torque on the input and output shafts of the transmission by a static multiplication by the gear ratio as shown in (3).

The dynamics of the transmission input are given by

$$\dot{\omega}_{GB1} = \left(\frac{1}{J_{c2} + J_{s2} + J_{GB1}} \right) T_c + B \cdot T_{belt} - T_{GB1} \quad (3)$$

Where

$\dot{\omega}_{GB1}$ = Transmission input angular acceleration

B = Belt ratio

T_c = Torque transmitted by the clutch

T_{GB1} = Torque transmitted to transmission primary shaft

J_{c2} = Clutch disk inertia

J_{s2} = Inertia of belt pulley attached to transmission input

J_{GB1} = Equivalent transmission primary shaft inertia

The dynamics of the transmission output are given by

$$\dot{\omega}_{GB2} = \left(\frac{1}{J_{GB2}} \right) R \cdot T_{GB1} - T_{ax,f} \quad (4)$$

Where

$\dot{\omega}_{GB2}$ = Transmission output angular acceleration

$T_{ax,f}$ = Torque transmitted to the front wheels

J_{GB2} = Equivalent transmission secondary shaft inertia

R = Transmission gear ratio

Rear Electric Machine and Gearbox

Similar to the engine and front electric machine, the rear electric machine is modeled as a lumped inertia, ideal torque actuator. The rear powertrain dynamics can be described with a lumped model that reflects all inertias to the rear electric machine output shaft.

$$\dot{\omega}_{RM} = \left(\frac{1}{J_{RM} + J_{g1} + \frac{J_{g2}}{r^2}} \right) T_{RM} - T_{RM,fr} - \frac{T_{ax,r}}{r} \quad (5)$$