HIGH-LEVEL MODELING, SUPERVISORY CONTROL STRATEGY DEVELOPMENT, AND VALIDATION FOR A PROPOSED POWER-SPLIT HYBRID-ELECTRIC VEHICLE DESIGN

A Thesis

Presented in Partial Fulfillment of the Requirements for the Degree Master of Science in the Graduate School of The Ohio State University

By

Joseph M. Morbitzer, B.S.

* * * * *

The Ohio State University

2005

Master's Examination Committee:

Dr. Giorgio Rizzoni, Adviser
Dr. Yann G. Guennec
Dr. Eric R. Westervelt

Approved by

Adviser
Graduate Program in Mechanical Engineering
ABSTRACT

Over the last decade, hybrid-electric vehicles have progressed from a futuristic icon to a firm production reality for a growing number of automobile manufacturers. While the motivation for this trend may vary, hybrid-electric vehicles today symbolize a recognition of the necessity to evolve advanced automotive technologies in order to sustain a culture of freedom of mobility. The Challenge X program communicates this message towards academia and future automotive engineers with strong support from both government and industry.

The work of this thesis was aimed toward The Ohio State University's objectives as a participant in the Challenge X competition. As an initial task, the Ohio State team defined a set of vehicle technical specifications to steer and motivate the vehicle design and control strategy development. After an extensive decision-making process, a specific architecture emerged with the potential to meet the vehicle technical specifications. The chosen configuration is a charge-sustaining, power-split, hybrid-electric vehicle design. A downsized Diesel engine and integrated starter/alternator drive the front wheels through an automatic transaxle. A larger, tractive electric machine and single-speed gearbox exist on the rear drivetrain. Both electric machines and their respective inverters connect electrically to a single high-voltage battery pack.

The validation procedure for both the vehicle architecture and a control strategy involves use of a computer vehicle simulator. A quasi-static vehicle model acts as
a basis for a simulator to validate the design and control strategy with respect to energy management. A dynamic vehicle model establishes a foundation for eventual creation of a second simulator for drivability validation. Both simulators operate in a forward-moving fashion and contain three primary sections: (i) the driver, (ii) the hybrid-electric powertrain, and (iii) the vehicle. Both models are also highly nonlinear, but the main differentiating property is the relatively large system order of the dynamic model as compared to the quasi-static model.

The high-level supervisory control strategy strives to accomplish certain objectives. The initial task involves appropriately selecting the vehicle mode from those predefined as being advantageous to the particular architecture. The control strategy then calculates the driver power request and commands the powertrain actuators so as to meet that request. In certain and applicable vehicle modes, the torque split also aims to minimize fuel consumption. High-voltage battery pack state-of-charge management is both indirectly and inherently incorporated into the fuel consumption minimization approach. As a future task, drivability assurance may involve a final adjustment of control strategy commands so as to respect certain levels of several identified drivability metrics during the vehicle response.

Rapid prototyping with a rolling chassis apparatus provided a method of investigation into the practicality of solely utilizing the tractive electric machine and high-voltage battery pack for vehicle propulsion. Initial experimentation validates functionality of the electric machine and inverter and also indicates potential for the power electronics system to act alone in acceptably accelerating the vehicle inertia from a rest. More revealing analysis of the vehicle architecture and control strategy occurred via software-in-the-loop techniques using a simulator based upon the
quasi-static vehicle model. Simulation results verify expected fuel economy gains from conversion to a downsized Diesel engine, engine disablement at a vehicle rest, and regenerative braking. However, the simulator also demonstrates a reduced fuel economy from extended operation of the vehicle in a pure electric mode. Moreover, the simulator indicates a concern with the ability of the tractive electric machine and proposed high-voltage battery pack to sufficiently and solely power the vehicle in a pure electric mode. Further findings of the simulated vehicle in full hybrid-electric vehicle operation clearly reveal the control strategy's preference in exclusively relying upon the Diesel engine for most normal operation. Reasons for this behavior primarily result from the relatively high efficiency of the Diesel engine and ensuing lack of opportunity to improve overall system efficiency through engine load shifting. Still, the downsized engine necessitates some presence of power electronics for supplementation during large power requests. Therefore, for this particular vehicle architecture, the control strategy may be better suited to simply maintain sufficient charge of the high-voltage battery pack for supplemental power delivery as opposed to aggressive and frequent use of the electric machines. Reflection of these simulation results along with some certain intangible issues motivates several suggestions concerning a few particular potential vehicle architecture modifications for consideration and contemplation by the Ohio State Challenge X team.
Dedicated to my parents.
ACKNOWLEDGMENTS

I would like to thank:

- My mother and father for their unending support in all I do.

- My adviser, Dr. Giorgio Rizzoli, for his continuous enthusiasm, valuable guidance, and positive influence on both my undergraduate and graduate studies.

- Dr. Yann G. Guezenaec and Dr. Eric R. Westervelt for participating on my examination committee and for their timely and appreciated advice.

- Shawn Midlam-Mohier for his patience in the numerous situations in which he shared some of his knowledge for my benefit.

- Staff and students at the Center for Automotive Research for their approachability, assistance, and friendship.

- And finally, all the members I was fortunate enough to work with from the Ohio State FutureTruck and Challenge X teams for the comradeship, teamwork, friendships, and many laughs.
VITA

07 April 1981 .......................... Born - Willoughby, Ohio

August 2003 .................................. B.S. Mechanical Engineering
                                            The Ohio State University
                                            Columbus, Ohio

September 2003 - Present .................. Graduate Student
                                            The Ohio State University
                                            Columbus, Ohio

Major Field: Mechanical Engineering
TABLE OF CONTENTS

Abstract ......................................................... ii
Dedication ..................................................... v
Acknowledgments ............................................... vi
Vita ............................................................. vii
List of Tables ................................................. xi
List of Figures ................................................... xii
List of Symbols ................................................. xvii

Chapters:

1. INTRODUCTION .............................................. 1
   1.1 Energy and Oil Situation of the United States of America .... 1
   1.2 Challenge X Program ..................................... 6
   1.3 Brief Hybrid-Electric Vehicles Overview ................... 7
       1.3.1 Definition of a Hybrid Vehicle ..................... 7
       1.3.2 Classes of Hybrid Vehicles ....................... 7
       1.3.3 Advantages/Disadvantages of Hybrid Vehicles ....... 8
       1.3.4 Current State of the Hybrid Vehicle Market ....... 9

2. VEHICLE ARCHITECTURE SELECTION ..................... 11
   2.1 Vehicle Technical Specifications ........................ 11
   2.2 Fuel Selection .......................................... 11
   2.3 Initial Brainstorming .................................... 18
2.4 Hybridization Analysis ........................................... 20
2.5 Towing Analysis .................................................... 23
2.6 Mass Analysis ...................................................... 25
2.7 Decision Matrix ................................................... 27
2.8 Vehicle Architecture Selection ................................. 31
2.9 Component Selection ............................................. 33
2.10 Exhaust Aftertreatment Selection .............................. 38
2.11 Control System Selection ........................................ 42
2.12 Validation Approach ............................................. 43
2.13 Summary ............................................................ 44

3. QUASI-STATIC MODEL ............................................... 46

3.1 Driver ............................................................... 47
3.2 Hybrid-Electric Powertrain ....................................... 48
  3.2.1 Diesel Engine .................................................. 51
  3.2.2 Fuel Tank ...................................................... 52
  3.2.3 Integrated Starter/Alternator and Tractive Electric Machine ........ 52
  3.2.4 High-Voltage Battery Pack and Switchbox .................... 53
  3.2.5 Torque Converter .............................................. 58
  3.2.6 Automatic Transaxle and Single-Speed Gearbox .............. 63
  3.2.7 Brakes .......................................................... 64
  3.2.8 Wheels .......................................................... 65
3.3 Vehicle ............................................................... 66
3.4 Summary ............................................................ 68

4. DYNAMIC MODEL .................................................... 70

4.1 Driver ............................................................... 70
4.2 Hybrid-Electric Drivetrain ....................................... 71
  4.2.1 Diesel Engine and Integrated Starter/Alternator ............... 74
  4.2.2 Fuel Tank ...................................................... 74
  4.2.3 Torque Converter .............................................. 75
  4.2.4 Tractive Electric Machine .................................... 75
  4.2.5 High-Voltage Battery Pack and Switchbox .................... 75
  4.2.6 Automatic Transaxle and Single-Speed Gearbox .............. 76
  4.2.7 Half Shafts .................................................... 76
  4.2.8 Brakes .......................................................... 77
  4.2.9 Wheels .......................................................... 77
4.3 Vehicle ............................................................... 77
4.4 Summary ............................................................ 78
# LIST OF TABLES

<table>
<thead>
<tr>
<th>Table</th>
<th>Description</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>2.1</td>
<td>Vehicle Technical Specifications</td>
<td>12</td>
</tr>
<tr>
<td>2.2</td>
<td>Road Load Energy Sensitivities to Vehicle Parameters</td>
<td>25</td>
</tr>
<tr>
<td>2.3</td>
<td>Component Selection and Specifications</td>
<td>34</td>
</tr>
<tr>
<td>2.4</td>
<td>Tier 2 Bin 5 Light-Duty Emission Standards [12]</td>
<td>39</td>
</tr>
<tr>
<td>5.1</td>
<td>Vehicle Modes Description</td>
<td>84</td>
</tr>
<tr>
<td>5.2</td>
<td>Drivability Metrics [40]</td>
<td>100</td>
</tr>
<tr>
<td>6.1</td>
<td>cX-SIM Simulation Results, Tests # 1 - 8</td>
<td>121</td>
</tr>
<tr>
<td>6.2</td>
<td>cX-SIM Simulation Results, Test # 9</td>
<td>135</td>
</tr>
<tr>
<td>A.1</td>
<td>Estimated Weight Addition</td>
<td>148</td>
</tr>
<tr>
<td>A.2</td>
<td>Opportunities for Weight Reduction</td>
<td>149</td>
</tr>
<tr>
<td>B.1</td>
<td>Road Load Parameters for Chassis Dynamometer</td>
<td>153</td>
</tr>
<tr>
<td>B.2</td>
<td>cX-SIM Simulation Parameter Values (Continued)</td>
<td>154</td>
</tr>
</tbody>
</table>
# LIST OF FIGURES

<table>
<thead>
<tr>
<th>Figure</th>
<th>Description</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.1</td>
<td>2003 United States Energy Source and Usage (Statistics from [48])</td>
<td>2</td>
</tr>
<tr>
<td>1.2</td>
<td>2004 United States Oil Source and Usage (Statistics from [48])</td>
<td>3</td>
</tr>
<tr>
<td>1.3</td>
<td>World Industrial Growth Rate vs. Automobile Ownership (Statistics from [28])</td>
<td>4</td>
</tr>
<tr>
<td>1.4</td>
<td>United States Yearly Oil Reserves and Discoveries (Statistics from [49])</td>
<td>5</td>
</tr>
<tr>
<td>2.1</td>
<td>GREET Energy Analysis</td>
<td>15</td>
</tr>
<tr>
<td>2.2</td>
<td>GREET Greenhouse Gas Emissions Analysis</td>
<td>16</td>
</tr>
<tr>
<td>2.3</td>
<td>GREET NOx Emissions Analysis</td>
<td>17</td>
</tr>
<tr>
<td>2.4</td>
<td>PSAT Fuel Economy Analysis</td>
<td>21</td>
</tr>
<tr>
<td>2.5</td>
<td>PSAT Combined Fuel Economy Analysis</td>
<td>22</td>
</tr>
<tr>
<td>2.6</td>
<td>Towing Road Load Analysis</td>
<td>24</td>
</tr>
<tr>
<td>2.7</td>
<td>Road Load Energy Analysis</td>
<td>27</td>
</tr>
<tr>
<td>2.8</td>
<td>Architecture Selection Chart</td>
<td>28</td>
</tr>
<tr>
<td>2.9</td>
<td>Vehicle Architecture Diagram</td>
<td>32</td>
</tr>
</tbody>
</table>
2.10 Transmission Selection Chart ...................................... 36
2.11 Exhaust Aftertreatment System ................................. 41
2.12 Validation Procedure ............................................. 45
3.1 Quasi-Static Model: High-Level Structure .................. 47
3.2 Quasi-Static Model: Drivetrain Schematic .................... 49
3.3 Quasi-Static Model: Powertrain Component Level Structure . 50
3.4 Model Schematic of High-Voltage Battery Pack .............. 55
4.1 Dynamic Model: High-Level Structure ......................... 71
4.2 Dynamic Model: Drivetrain Schematic ........................ 72
4.3 Dynamic Model: Powertrain Component Level Structure ..... 73
5.1 Transaxle State and Gear Selection Logic ..................... 82
5.2 Vehicle Mode Selection Logic .................................... 85
5.3 Equivalent Fuel Consumption Usage ............................ 93
5.4 Equivalent Fuel Consumption Savings ......................... 94
5.5 HV Battery Pack Power Correction Factors .................. 97
6.1 Rolling Chassis Apparatus ...................................... 108
6.2 Rolling Chassis Test with CAN Control ....................... 110
6.3 Rolling Chassis Test with Analog Control .................... 111
6.4 cX-SIM Graphical User Interface ............................. 114
6.5 cX-SIM Hybrid-Electric Powertrain Structure ............... 115
6.6 Inputs and Outputs of Control Strategy ....................... 117
6.7 Control Strategy Progression ........................................ 118
6.8 Test # 2 and 3: FUDS Cycle ........................................ 122
6.9 Test # 2 and 3: FHDS Cycle ........................................ 122
6.10 Test # 2 and 3: 0-60 mph Acceleration ......................... 124
6.11 Test # 5: 0-60 mph Acceleration ............................... 125
6.12 Test # 5: Engine Operating Points .............................. 125
6.13 Test # 8: Engine Operating Points .............................. 128
6.14 Test # 8: Tractive Electric Machine Operating Points .......... 128
6.15 Electric Only Mode Investigation: HV Battery Pack Voltage ..... 130
6.16 Electric Only Mode Investigation: HV Battery Pack Capacity .... 130
6.17 $\bar{\eta}_{ch}$ and $\bar{\eta}_{dis}$ Effect on FUDS Fuel Economy .......... 131
6.18 $\bar{\eta}_{ch}$ and $\bar{\eta}_{dis}$ Effect on FHDS Fuel Economy .......... 132
6.19 $\bar{\eta}_{ch}$ and $\bar{\eta}_{dis}$ Effect on FUDS Final HV Battery Pack SOC .... 133
6.20 $\bar{\eta}_{ch}$ and $\bar{\eta}_{dis}$ Effect on FHDS Final HV Battery Pack SOC .... 134
6.21 $\zeta_{ch}$ ($s_{ch}$) and $\zeta_{dis}$ ($s_{dis}$) Adaptive Factors .......... 136
6.22 Test # 9: FUDS: Vehicle Velocity and HV Battery Pack SOC .... 138
6.23 Test # 9: FHDS: Vehicle Velocity and HV Battery Pack SOC .... 138
6.24 Test # 9: FUDS: Powertrain Actuators Torque Output ........... 139
6.25 Test # 9: FHDS: Powertrain Actuators Torque Output ........... 139
6.26 Test # 9: FUDS: Accelerator Pedal Position, EM Efficiency, and HV Battery Pack Current .......... 140
A.1 Decision Matrix .................................................. 150

A.2 Control System Architecture
   (Legend in Figure A.3) ........................................ 151

A.3 Control System Diagram Legend
   (for Figure A.2) ............................................... 152

B.1 Chevy Equinox Stock 3.4L Gasoline Engine Fuel Rate Map
   (Data Courtesy of GM [30]) .................................... 156

B.2 Chevy Equinox Stock 3.4L Gasoline Engine Efficiency Map ....... 156

B.3 Chevy Equinox Stock 3.4L Gasoline Engine Brake Power Map .... 157

B.4 Fiat 1.9L Diesel Engine Fuel Consumption Rate Map
   (Data Courtesy of Fiat [5]) ..................................... 157

B.5 Fiat 1.9L Diesel Engine Efficiency Map .......................... 158

B.6 Fiat 1.9L Diesel Engine Brake Power Map ....................... 158

B.7 Integrated Starter/Alternator Efficiency Map ..................... 159

B.8 Torque Converter Characteristic Curve .......................... 160

B.9 Torque Converter Efficiency Map ................................ 161

B.10 Transaxle Shifting Schedule ................................... 161

B.11 Delphi EV1 Tractive Electric Machine Efficiency Map
   (Data Courtesy of Delphi [4]) ................................ 162

B.12 Panasonic Prismatic Battery Module Characteristics
   (Data Courtesy of [8]) ........................................... 162

B.13 Test # 9: FUDS: Diesel Engine Operating Points ............... 163

B.14 Test # 9: FUDS: Integrated Starter/Alternator Operating Points .. 163
B.15 Test # 9: FUDS: Tractive Electric Machine Operating Points . . . . 164

B.16 Test # 9: FHDS: Diesel Engine Operating Points . . . . . . . . . . 164

B.17 Test # 9: FHDS: Integrated Starter/Alternator Operating Points . 165

B.18 Test # 9: FHDS: Tractive Electric Machine Operating Points . . . . 165
LIST OF SYMBOLS

Acronyms and symbols used in this thesis are listed and defined in the subsequent tables. Unless otherwise noted, the International System of Units (metric) is assumed for all dimensions in both the thesis text and equations. For other units as noted, the acronym definition appears in the list below.

<table>
<thead>
<tr>
<th>Acronym</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>AC</td>
<td>Alternating Current</td>
</tr>
<tr>
<td>AMT</td>
<td>Automated-Manual Transmission</td>
</tr>
<tr>
<td>Ah</td>
<td>Amp-Hour</td>
</tr>
<tr>
<td>B20</td>
<td>20% BioDiesel (and 80% low-sulfur Diesel blend)</td>
</tr>
<tr>
<td>BTU</td>
<td>British Thermal Unit</td>
</tr>
<tr>
<td>°C</td>
<td>(degrees) Celsius</td>
</tr>
<tr>
<td>CAN</td>
<td>Controller Area Network</td>
</tr>
<tr>
<td>CAR</td>
<td>(The Ohio State University) Center for Automotive Research</td>
</tr>
<tr>
<td>CIDI</td>
<td>Compression-Ignition Direct-Injection</td>
</tr>
<tr>
<td>CO</td>
<td>Carbon Monoxide</td>
</tr>
<tr>
<td>CVT</td>
<td>Continuously-Variable Transmission</td>
</tr>
<tr>
<td>cX-SIM</td>
<td>Challenge X (quasi-static vehicle) SIMulator</td>
</tr>
<tr>
<td>cX-DYN</td>
<td>Challenge X DYNamic (vehicle simulator)</td>
</tr>
<tr>
<td>DC</td>
<td>Direct Current (electricity)</td>
</tr>
<tr>
<td>DPF</td>
<td>Diesel Particulate Filter</td>
</tr>
<tr>
<td>E85</td>
<td>85% Ethanol (and 15% reformulated gasoline blend)</td>
</tr>
<tr>
<td>EM</td>
<td>Electric Machine (refers to rear, tractive electric machine)</td>
</tr>
<tr>
<td>EVT</td>
<td>Electronically-Variable Transmission</td>
</tr>
<tr>
<td>Symbol</td>
<td>Definition</td>
</tr>
<tr>
<td>--------</td>
<td>------------</td>
</tr>
<tr>
<td>(°) F</td>
<td>(degrees) Fahrenheit</td>
</tr>
<tr>
<td>FCV</td>
<td>Fuel Cell Vehicle</td>
</tr>
<tr>
<td>FHDS</td>
<td>Federal Highway Driving Schedule</td>
</tr>
<tr>
<td>FUDS</td>
<td>Federal Urban Driving Schedule</td>
</tr>
<tr>
<td>GHG</td>
<td>GreenHouse Gas (emissions)</td>
</tr>
<tr>
<td>GM</td>
<td>General Motors (Corporation)</td>
</tr>
<tr>
<td>GREET</td>
<td>Greenhouse gases, Regulated Emissions, and Energy use in Transportation</td>
</tr>
<tr>
<td>GUI</td>
<td>Graphical User Interface</td>
</tr>
<tr>
<td>H₂</td>
<td>Hydrogen</td>
</tr>
<tr>
<td>HCCI</td>
<td>Homogeneous-Charge Compression-Ignition</td>
</tr>
<tr>
<td>HEV</td>
<td>Hybrid-Electric Vehicle</td>
</tr>
<tr>
<td>HIL</td>
<td>Hardware-In-the-Loop</td>
</tr>
<tr>
<td>HV</td>
<td>High-Voltage</td>
</tr>
<tr>
<td>hp</td>
<td>Horsepower</td>
</tr>
<tr>
<td>ICE</td>
<td>Internal Combustion Engine</td>
</tr>
<tr>
<td>IMA</td>
<td>Integrated Motor Assist</td>
</tr>
<tr>
<td>ISA</td>
<td>Integrated Starter/Alternator</td>
</tr>
<tr>
<td>IVM</td>
<td>Initial Vehicle Movement</td>
</tr>
<tr>
<td>lbs</td>
<td>Pounds</td>
</tr>
<tr>
<td>LNT</td>
<td>Lean NOₓ Trap</td>
</tr>
<tr>
<td>mi</td>
<td>Mile</td>
</tr>
<tr>
<td>mpgge</td>
<td>Miles Per Gallon, Gasoline Equivalent</td>
</tr>
<tr>
<td>mph</td>
<td>Miles Per Hour</td>
</tr>
<tr>
<td>NMOG</td>
<td>Non-Methane Organic Gas</td>
</tr>
<tr>
<td>NiMH</td>
<td>Nickel-Metal Hydride</td>
</tr>
<tr>
<td>NOₓ</td>
<td>Oxides of Nitrogen</td>
</tr>
<tr>
<td>OPEC</td>
<td>Oil Producing Exporting Countries</td>
</tr>
<tr>
<td>OSU</td>
<td>(The) Ohio State University</td>
</tr>
<tr>
<td>PID</td>
<td>Proportional-Integral-Derivative</td>
</tr>
<tr>
<td>Symbol</td>
<td>Description</td>
</tr>
<tr>
<td>--------</td>
<td>-------------</td>
</tr>
<tr>
<td>$A$</td>
<td>Gain of Lyapunov potential function</td>
</tr>
<tr>
<td>$A_f$</td>
<td>Projected frontal area of vehicle $m^2$</td>
</tr>
<tr>
<td>$b_{hs,f}$</td>
<td>Angular damping coefficient of front half shafts $N \cdot m \cdot s/rad$</td>
</tr>
<tr>
<td>$b_{hs,r}$</td>
<td>Angular damping coefficient of rear half shafts $N \cdot m \cdot s/rad$</td>
</tr>
<tr>
<td>$b_{veh}$</td>
<td>Lumped translational damping coefficient of tires, vehicle suspension, etc. $N \cdot s/m$</td>
</tr>
<tr>
<td>Symbol</td>
<td>Units</td>
</tr>
<tr>
<td>--------</td>
<td>---------------</td>
</tr>
<tr>
<td>$C_1 - C_0$</td>
<td>$kg \cdot m^2/rad^2$</td>
</tr>
<tr>
<td>$C_d$</td>
<td>-</td>
</tr>
<tr>
<td>$C_r$</td>
<td>-</td>
</tr>
<tr>
<td>$D_{dr}$</td>
<td>$s^2/m$</td>
</tr>
<tr>
<td>$E_{be}$</td>
<td>$A \cdot s$</td>
</tr>
<tr>
<td>$F_{rd}$</td>
<td>$N$</td>
</tr>
<tr>
<td>$F_{trac}$</td>
<td>$N$</td>
</tr>
<tr>
<td>$F_{veh}$</td>
<td>$N$</td>
</tr>
<tr>
<td>$F_{wh,f}$</td>
<td>$N$</td>
</tr>
<tr>
<td>$F_{wh,r}$</td>
<td>$N$</td>
</tr>
<tr>
<td>$g$</td>
<td>$m/s^2$</td>
</tr>
<tr>
<td>$I_{dr}$</td>
<td>$1/m$</td>
</tr>
<tr>
<td>$I_{isa}$</td>
<td>$N \cdot m/rad$</td>
</tr>
<tr>
<td>$i_{bt}$</td>
<td>$A$</td>
</tr>
<tr>
<td>$i_{bt,nn}$</td>
<td>$A$</td>
</tr>
<tr>
<td>Symbol</td>
<td>Definition</td>
</tr>
<tr>
<td>--------</td>
<td>------------</td>
</tr>
<tr>
<td>$i_{ul, mx}$</td>
<td>Assumed maximum electrical current output of HV battery pack</td>
</tr>
<tr>
<td>$i_{th,p,mn}$</td>
<td>Theoretical electrical current input of HV battery pack for theoretical maximum HV battery pack power input</td>
</tr>
<tr>
<td>$i_{th,p,mz}$</td>
<td>Theoretical electrical current output of HV battery pack for theoretical maximum HV battery pack power output</td>
</tr>
<tr>
<td>$J_{em}$</td>
<td>Rotational inertia of EM rotor $\text{kg} \cdot \text{m}^2$</td>
</tr>
<tr>
<td>$J_{ice}$</td>
<td>Rotational inertia of ICE crankshaft $\text{kg} \cdot \text{m}^2$</td>
</tr>
<tr>
<td>$J_{isa}$</td>
<td>Rotational inertia of ISA rotor $\text{kg} \cdot \text{m}^2$</td>
</tr>
<tr>
<td>$k_{hs,f}$</td>
<td>Rotational stiffness of front half shafts $\text{N} \cdot \text{m/\text{rad}}$</td>
</tr>
<tr>
<td>$k_{hs,r}$</td>
<td>Rotational stiffness of rear half shafts $\text{N} \cdot \text{m/\text{rad}}$</td>
</tr>
<tr>
<td>$k_{veh}$</td>
<td>Translational lumped stiffness of tires, vehicle suspension, etc. $\text{N/\text{m}}$</td>
</tr>
<tr>
<td>$m_{em,x}$</td>
<td>Reflected rotational inertia of EM rotor to translational inertia at vehicle level $\text{kg}$</td>
</tr>
<tr>
<td>$m_f$</td>
<td>Mass of Diesel fuel used by ICE over elapsed time period $t_e$ $\text{kg}$</td>
</tr>
<tr>
<td>$\dot{m}_f$</td>
<td>Mass flow rate of Diesel fuel into ICE $\text{kg/s}$</td>
</tr>
<tr>
<td>$\dot{m}_{f, leq}$</td>
<td>Equivalent mass flow rate of Diesel fuel of HV battery pack $\text{kg/s}$</td>
</tr>
<tr>
<td>$\dot{m}_{f, eq}$</td>
<td>Total equivalent mass flow rate of Diesel fuel by ICE and HV battery pack $\text{kg/s}$</td>
</tr>
<tr>
<td>Symbol</td>
<td>Unit</td>
</tr>
<tr>
<td>--------</td>
<td>------</td>
</tr>
<tr>
<td>$m_{NO_x}$</td>
<td>kg/s</td>
</tr>
<tr>
<td>Mass flow rate of ICE engine-out NO_x</td>
<td></td>
</tr>
<tr>
<td>$m_{f,NO_x,eq}$</td>
<td>kg/s</td>
</tr>
<tr>
<td>Equivalent mass flow rate of Diesel fuel associated with ICE engine-out NO_x mass flow rate</td>
<td></td>
</tr>
<tr>
<td>$m_{veh}$</td>
<td>kg</td>
</tr>
<tr>
<td>Curb mass of vehicle</td>
<td></td>
</tr>
<tr>
<td>$m_{veh,c}$</td>
<td>kg</td>
</tr>
<tr>
<td>Effective mass of vehicle</td>
<td></td>
</tr>
<tr>
<td>$n_{bt,p}$</td>
<td>-</td>
</tr>
<tr>
<td>Number of battery module strings in parallel in HV battery pack</td>
<td></td>
</tr>
<tr>
<td>$n_{bt,s}$</td>
<td>-</td>
</tr>
<tr>
<td>Number of battery modules in series per string in HV battery pack</td>
<td></td>
</tr>
<tr>
<td>$P_{bt}$</td>
<td>W</td>
</tr>
<tr>
<td>Electrical power output/input of HV battery pack</td>
<td></td>
</tr>
<tr>
<td>$P_{bt,ma}$</td>
<td>W</td>
</tr>
<tr>
<td>Maximum electrical power input of HV battery pack</td>
<td></td>
</tr>
<tr>
<td>$P_{bt,ma,cor}$</td>
<td>W</td>
</tr>
<tr>
<td>Corrected maximum electrical power input of HV battery pack</td>
<td></td>
</tr>
<tr>
<td>$P_{bt,mx}$</td>
<td>W</td>
</tr>
<tr>
<td>Maximum electrical power output of HV battery pack</td>
<td></td>
</tr>
<tr>
<td>$P_{bt,mx,cor}$</td>
<td>W</td>
</tr>
<tr>
<td>Corrected maximum electrical power output of HV battery pack</td>
<td></td>
</tr>
<tr>
<td>$P_{dc-de, e}$</td>
<td>W</td>
</tr>
<tr>
<td>Electrical power from HV battery pack to DC-DC converter</td>
<td></td>
</tr>
<tr>
<td>$P_{dr}$</td>
<td>s/m</td>
</tr>
<tr>
<td>Proportional coefficient of PID driver model</td>
<td></td>
</tr>
<tr>
<td>$P_{em,e}$</td>
<td>W</td>
</tr>
<tr>
<td>Electrical power output/input of EM inverter</td>
<td></td>
</tr>
<tr>
<td>$P_{isa}$</td>
<td>N·m·s/rad</td>
</tr>
<tr>
<td>Proportional coefficient of PID-modeled control for ISA in Engine Start mode</td>
<td></td>
</tr>
<tr>
<td>Symbol</td>
<td>Description</td>
</tr>
<tr>
<td>--------</td>
<td>-------------</td>
</tr>
<tr>
<td>( P_{\text{isa,e}} )</td>
<td>Electrical power output/input of ISA inverter</td>
</tr>
<tr>
<td>( P_{\text{rd,nn}} )</td>
<td>Maximum mechanical power input of drivetrain at road</td>
</tr>
<tr>
<td>( P_{\text{rd,mx}} )</td>
<td>Maximum mechanical power output of drivetrain at road</td>
</tr>
<tr>
<td>( P_{\text{rd,mx,e}} )</td>
<td>Maximum mechanical power output of ISA and EM at road</td>
</tr>
<tr>
<td>( P_{\text{rd,rg}} )</td>
<td>Requested mechanical power output/input of drivetrain at road</td>
</tr>
<tr>
<td>( Q_{\text{hvd}} )</td>
<td>Lower heating value of Diesel fuel</td>
</tr>
<tr>
<td>( Q_{\text{hvg}} )</td>
<td>Lower heating value of gasoline fuel</td>
</tr>
<tr>
<td>( R_{\text{ch}} )</td>
<td>Internal electrical charging resistance of single battery module from HV battery pack</td>
</tr>
<tr>
<td>( R_{\text{dis}} )</td>
<td>Internal electrical discharging resistance of single battery module from HV battery pack</td>
</tr>
<tr>
<td>( r_{\text{wh}} )</td>
<td>Total radius of wheel and tire assembly</td>
</tr>
<tr>
<td>( s_{\text{bt}} )</td>
<td>State-of-charge of HV battery pack</td>
</tr>
<tr>
<td>( \dot{s}_{\text{bt}} )</td>
<td>Threshold state-of-charge of HV battery pack for transition into/out of Electric Only mode</td>
</tr>
<tr>
<td>( s_{\text{bt,0}} )</td>
<td>Assumed initial state-of-charge of HV battery pack</td>
</tr>
<tr>
<td>( \dot{s}_{\text{bt,1}} )</td>
<td>Specified lower state-of-charge limit of HV battery pack</td>
</tr>
<tr>
<td>Symbol</td>
<td>Units</td>
</tr>
<tr>
<td>--------</td>
<td>-------</td>
</tr>
<tr>
<td>$s_{bt,nom}$</td>
<td>-</td>
</tr>
<tr>
<td>$\hat{s}_{bt,u}$</td>
<td>-</td>
</tr>
<tr>
<td>$T_{brk,f}$</td>
<td>N·m</td>
</tr>
<tr>
<td>$T_{brk,\max}$</td>
<td>N·m</td>
</tr>
<tr>
<td>$T_{brk,r}$</td>
<td>N·m</td>
</tr>
<tr>
<td>$T_{em}$</td>
<td>N·m</td>
</tr>
<tr>
<td>$T_{em,br,mn}$</td>
<td>N·m</td>
</tr>
<tr>
<td>$T_{em,br,\max}$</td>
<td>N·m</td>
</tr>
<tr>
<td>$T_{em,mn}$</td>
<td>N·m</td>
</tr>
<tr>
<td>$T_{em,\max}$</td>
<td>N·m</td>
</tr>
<tr>
<td>$T_{em,net}$</td>
<td>N·m</td>
</tr>
<tr>
<td>$T_{em,rq}$</td>
<td>N·m</td>
</tr>
<tr>
<td>$T_{gb}$</td>
<td>N·m</td>
</tr>
<tr>
<td>$T_{hs,f}$</td>
<td>N·m</td>
</tr>
<tr>
<td>Symbol</td>
<td>Units</td>
</tr>
<tr>
<td>--------</td>
<td>-------</td>
</tr>
<tr>
<td>$T_{hs,r}$</td>
<td>$N \cdot m$</td>
</tr>
<tr>
<td>Torque through rear half shafts</td>
<td></td>
</tr>
<tr>
<td>$T_{\text{ICe}}$</td>
<td>$N \cdot m$</td>
</tr>
<tr>
<td>Brake output torque of ICE</td>
<td></td>
</tr>
<tr>
<td>$T_{\text{ICe},rq}$</td>
<td>$N \cdot m$</td>
</tr>
<tr>
<td>Requested brake output torque of ICE</td>
<td></td>
</tr>
<tr>
<td>$T_{\text{ICe},mx}$</td>
<td>$N \cdot m$</td>
</tr>
<tr>
<td>Maximum brake output torque of ICE</td>
<td></td>
</tr>
<tr>
<td>$T_{\text{ISA}}$</td>
<td>$N \cdot m$</td>
</tr>
<tr>
<td>Brake output/input torque of ISA</td>
<td></td>
</tr>
<tr>
<td>$T_{\text{ISA},mn}$</td>
<td>$N \cdot m$</td>
</tr>
<tr>
<td>Maximum brake input torque of ISA</td>
<td></td>
</tr>
<tr>
<td>$T_{\text{ISA},mx}$</td>
<td>$N \cdot m$</td>
</tr>
<tr>
<td>Maximum brake output torque of ISA</td>
<td></td>
</tr>
<tr>
<td>$T_{\text{ISA},rq}$</td>
<td>$N \cdot m$</td>
</tr>
<tr>
<td>Requested brake output/input torque of ISA</td>
<td></td>
</tr>
<tr>
<td>$T_{\text{TC},p}$</td>
<td>$N \cdot m$</td>
</tr>
<tr>
<td>Input/output torque of torque converter pump</td>
<td></td>
</tr>
<tr>
<td>$\dot{T}_{\text{TC},p}$</td>
<td>$N \cdot m$</td>
</tr>
<tr>
<td>Threshold output torque of torque converter pump at transition between torque multiplication and torque coupling modes of torque converter</td>
<td></td>
</tr>
<tr>
<td>$T_{\text{TC},t}$</td>
<td>$N \cdot m$</td>
</tr>
<tr>
<td>Output/input torque of torque converter turbine</td>
<td></td>
</tr>
<tr>
<td>$T_{\text{Tr}}$</td>
<td>$N \cdot m$</td>
</tr>
<tr>
<td>Output/input torque of automatic transaxle (post gear reduction)</td>
<td></td>
</tr>
<tr>
<td>$T_{\text{WH},f}$</td>
<td>$N \cdot m$</td>
</tr>
<tr>
<td>Output/input torque of front wheels</td>
<td></td>
</tr>
<tr>
<td>$T_{\text{WH},r}$</td>
<td>$N \cdot m$</td>
</tr>
<tr>
<td>Output/input torque of rear wheels</td>
<td></td>
</tr>
<tr>
<td>$t$</td>
<td>$s$</td>
</tr>
<tr>
<td>Time</td>
<td></td>
</tr>
</tbody>
</table>

*xxv*
\begin{tabular}{|l|l|}
\hline
$t_e$ & $s$ \\
Elapsed time & \\
\hline
$t_f$ & $s$ \\
Elapsed time over life of vehicle & \\
\hline
$V_{cc}$ & $V$ \\
Closed-circuit voltage of HV battery pack & \\
\hline
$V_{oc}$ & $V$ \\
Open-circuit voltage of single battery module from HV battery pack & \\
\hline
$V_f$ & $m^3$ \\
Volume of Diesel fuel used by ICE over elapsed time period $t_e$ & \\
\hline
$V_{f,eq}$ & $m^3$ \\
Equivalent volume of gasoline fuel used by ICE over elapsed time period $t_e$ & \\
\hline
$v_{des}$ & $m/s$ \\
Driver-desired translational velocity of vehicle & \\
\hline
$v_{veh}$ & $m/s$ \\
Translational velocity of vehicle & \\
\hline
$v_{veh,f}$ & $m/s$ \\
Translational velocity of front of vehicle & \\
\hline
$v_{veh,l}$ & $m/s$ \\
Threshold translational velocity of vehicle for transition into Engine Start mode & \\
\hline
$v_{veh,r}$ & $m/s$ \\
Translational velocity of rear of vehicle & \\
\hline
$v_{veh,u}$ & $m/s$ \\
Threshold translational velocity of vehicle for transition into Electric Only mode & \\
\hline
$x_{veh}$ & $m$ \\
Translational displacement of vehicle over elapsed time period $t_e$ & \\
\hline
$\alpha$ & - \\
Accelerator pedal position & \\
\hline
$\dot{\alpha}$ & - \\
Threshold accelerator pedal position for transition into Engine Start mode & \\
\hline
\end{tabular}
<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\beta$</td>
<td>Brake pedal position</td>
</tr>
<tr>
<td>$\gamma_{ice}$</td>
<td>State of ICE (1 = on, 0 = off)</td>
</tr>
<tr>
<td>$\gamma_{NO_x}$</td>
<td>Transformation factor between $\dot{m}<em>{f,NO_x,eq}$ and $\dot{m}</em>{NO_x}$</td>
</tr>
<tr>
<td>$\gamma_{tc}$</td>
<td>Torque converter clutch lock-up command (1 = lock-up on, 0 = lock-up off)</td>
</tr>
<tr>
<td>$\gamma_{tr}$</td>
<td>Automatic transaxle gear command (1 = 1st gear, ..., 6 = 6th gear)</td>
</tr>
<tr>
<td>$\gamma_{veh}$</td>
<td>Effective vehicle mass factor</td>
</tr>
<tr>
<td>$\gamma_{4WD}$</td>
<td>4WD initiation command (1 = 4WD on, 0 = 4WD off)</td>
</tr>
<tr>
<td>$\Delta_{oh}$</td>
<td>Change in state-of-charge of HV battery pack over elapsed time period $t_c$</td>
</tr>
<tr>
<td>$\zeta_{oh,cm}$</td>
<td>Correction factors on HV battery pack maximum input power</td>
</tr>
<tr>
<td>$\zeta_{oh,dis}$</td>
<td>Correction factors on HV battery pack maximum output power</td>
</tr>
<tr>
<td>$\zeta_{brk,f}$</td>
<td>Braking proportionality for front brake calipers</td>
</tr>
<tr>
<td>$\zeta_{brk,r}$</td>
<td>Braking proportionality for rear brake calipers</td>
</tr>
<tr>
<td>$\zeta_{ch}$</td>
<td>Adaptive tuning factor for average combined charging efficiency</td>
</tr>
<tr>
<td>$\zeta_{dis}$</td>
<td>Adaptive tuning factor for average combined discharging efficiency</td>
</tr>
<tr>
<td>$\zeta_{em}$</td>
<td>Gain factor for adjustment of EM regenerative braking</td>
</tr>
<tr>
<td>Symbol</td>
<td>Description</td>
</tr>
<tr>
<td>--------</td>
<td>-------------</td>
</tr>
<tr>
<td>$\zeta_{m,f}$</td>
<td>Proportion of curb mass in front of vehicle</td>
</tr>
<tr>
<td>$\zeta_{m,r}$</td>
<td>Proportion of curb mass in rear of vehicle</td>
</tr>
<tr>
<td>$\eta_{be}$</td>
<td>Electrical efficiency of HV battery pack</td>
</tr>
<tr>
<td>$\bar{\eta}_{be}$</td>
<td>(Assumed) average electrical efficiency of HV battery pack</td>
</tr>
<tr>
<td>$\bar{\eta}_{ch}$</td>
<td>Average combined efficiency for HV battery pack charging condition</td>
</tr>
<tr>
<td>$\bar{\eta}_{ch,0}$</td>
<td>Nominal average combined efficiency for HV battery pack charging condition</td>
</tr>
<tr>
<td>$\bar{\eta}_{dis}$</td>
<td>Average combined efficiency for HV battery pack discharging condition</td>
</tr>
<tr>
<td>$\bar{\eta}_{dis,0}$</td>
<td>Nominal average combined efficiency for HV battery pack discharging condition</td>
</tr>
<tr>
<td>$\eta_{em}$</td>
<td>Electrical $\leftrightarrow$ mechanical combined efficiency of EM and inverter</td>
</tr>
<tr>
<td>$\eta_{fe}$</td>
<td>Fuel economy efficiency of vehicle</td>
</tr>
<tr>
<td>$\eta_{ice}$</td>
<td>Chemical $\rightarrow$ mechanical efficiency of ICE</td>
</tr>
<tr>
<td>$\bar{\eta}_{ice}$</td>
<td>(Assumed) average chemical $\rightarrow$ mechanical efficiency of ICE</td>
</tr>
<tr>
<td>$\eta_{isa}$</td>
<td>Electrical $\leftrightarrow$ mechanical combined efficiency of ISA and inverter</td>
</tr>
<tr>
<td>$\bar{\eta}_{isa}$</td>
<td>(Assumed) average electrical $\leftrightarrow$ mechanical combined efficiency of ISA and inverter</td>
</tr>
<tr>
<td>Symbol</td>
<td>Definition</td>
</tr>
<tr>
<td>-------</td>
<td>------------</td>
</tr>
<tr>
<td>$\eta_{gb}$</td>
<td>Mechanical efficiency of EM gearbox</td>
</tr>
<tr>
<td>$\eta_{tc}$</td>
<td>Mechanical efficiency of torque converter</td>
</tr>
<tr>
<td>$\eta_{tr}$</td>
<td>Mechanical efficiency of automatic transaxle</td>
</tr>
<tr>
<td>$\mu$</td>
<td>Coefficient of static friction between tires and ground</td>
</tr>
<tr>
<td>$\rho_{\text{air}}$</td>
<td>Mass density of air at standard temperature and pressure $\left(\text{kg/m}^3\right)$</td>
</tr>
<tr>
<td>$\rho_d$</td>
<td>Mass density of Diesel fuel $\left(\text{kg/m}^3\right)$</td>
</tr>
<tr>
<td>$\rho_g$</td>
<td>Mass density of gasoline fuel $\left(\text{kg/m}^3\right)$</td>
</tr>
<tr>
<td>$\tau_{em}$</td>
<td>Time constant of torque response of EM $\left(\text{s}\right)$</td>
</tr>
<tr>
<td>$\tau_{isa}$</td>
<td>Time constant of torque response of ISA $\left(\text{s}\right)$</td>
</tr>
<tr>
<td>$\tau_{gb}$</td>
<td>Single-speed ratio of EM gearbox (input:output)</td>
</tr>
<tr>
<td>$\tau_{tc}$</td>
<td>Speed ratio of torque converter (turbine:pump)</td>
</tr>
<tr>
<td>$\tau_{tr}$</td>
<td>Threshold speed ratio of torque converter (turbine:pump) at transition between torque multiplication and torque coupling modes of torque converter</td>
</tr>
<tr>
<td>$\tau_{tr}$</td>
<td>Speed ratio of automatic transaxle (input:output)</td>
</tr>
<tr>
<td>$\Upsilon$</td>
<td>Magnitude of Lyapunov potential function</td>
</tr>
<tr>
<td>$\phi$</td>
<td>Slope of road with respect to horizontal $\left(\text{rad}\right)$</td>
</tr>
<tr>
<td>Symbol</td>
<td>Unit</td>
</tr>
<tr>
<td>--------</td>
<td>------------</td>
</tr>
<tr>
<td>$\omega_{em}$</td>
<td>rad/s</td>
</tr>
<tr>
<td>$\omega_{gb}$</td>
<td>rad/s</td>
</tr>
<tr>
<td>$\omega_{hs,f}$</td>
<td>rad/s</td>
</tr>
<tr>
<td>$\omega_{hs,r}$</td>
<td>rad/s</td>
</tr>
<tr>
<td>$\omega_{ice}$</td>
<td>rad/s</td>
</tr>
<tr>
<td>$\dot{\omega}_{ice}$</td>
<td>rad/s</td>
</tr>
<tr>
<td>$\dot{\omega}_{ice, dn}$</td>
<td>rad/s</td>
</tr>
<tr>
<td>$\dot{\omega}_{ice, up}$</td>
<td>rad/s</td>
</tr>
<tr>
<td>$\omega_{t}$</td>
<td>rad/s</td>
</tr>
<tr>
<td>$\omega_{tr}$</td>
<td>rad/s</td>
</tr>
<tr>
<td>$\omega_{wh,f}$</td>
<td>rad/s</td>
</tr>
<tr>
<td>$\omega_{wh,r}$</td>
<td>rad/s</td>
</tr>
</tbody>
</table>
CHAPTER 1

INTRODUCTION

1.1 Energy and Oil Situation of the United States of America

The energy situation of the United States is overwhelming: Americans, at under 5% of the world’s population in 2003, used 98.1 quadrillion (10^{15}) BTUs of energy (25% of the world’s total) during that same year [48]. As relatively large numbers such as these are difficult to put into perspective, a few equivalent actions are listed below.

98.1 Quadrillion BTUs of Energy Equates to:

- A “Little Boy” atomic bomb (Hiroshima) detonating every 21 seconds for one year.

- Every person in the U.S. continuously driving a Chevrolet Equinox at approximately 20 mph for one year.

- Every person in the U.S. bench weightlifting 1877 lbs at one repetition a second for one year.

- Warming about half of Lake Erie (57 mi^3) from ice at 32°F to liquid at 75°F (latent and specific heat).
• 1151.6 kg of mass instantly “transformed” into energy ($E = mc^2$).

As of 01 January 2005, the U.S. led the world in coal reserves and stood 6th with respect to natural gas but ranked 11th in the world in oil reserves [48]. Still (as in Figure 1.1), the United States relied most heavily on oil to support its energy expenditure as over a quarter of the total energy was utilized for transportation purposes. In total, a projected 173.3 billion gasoline-equivalent gallons of fuel powered American vehicles in 2003 with over 99.7% of this volume either gasoline or Diesel gasoline-equivalent fuel [47]. Therefore, it is reasonable to assume that effectively all the energy used for vehicle transportation came from oil with a negligible amount from natural gas or renewable sources.

![U.S. Energy Source and Use in 2003](image)

Figure 1.1: 2003 United States Energy Source and Usage (Statistics from [48])
From January to October of 2004, America consumed oil at a rate of 20.4 million barrels (1 barrel = 42 gallons) per average day (as in Figure 1.2); equivalently, each person in the U.S. "used" 2.9 gallons everyday by themselves. The U.S. imported 58% of this oil whereas, by comparison, it imported 35% of its oil at the time of the 1973 Middle Eastern Oil Producing Exporting Countries (OPEC) nations oil embargo upon the United States. America’s over 500,000 oil wells supported 42% of its total oil consumption, but the great majority of these are marginal wells that reap only a few barrels of oil a day [48]. Approximately 70% of the United State’s daily oil usage facilitated transportation in the form of gasoline oil, Diesel fuel oil, or jet fuel oil. Specifically, the oil powers the almost 800 vehicles per 1000 people in the United States; by comparison, India owns approximately 13 automobiles per 1000 inhabitants and China possesses only 10 (Figure 1.3).

Figure 1.2: 2004 United States Oil Source and Usage (Statistics from [48])
As in Figure 1.4, the United States has been continually and constantly depleting its oil supply reserves (21.9 billion barrels as of 01 January 2005, down 31% from 1977). Annual discoveries have been relatively constant over the past 30 years and amounted to under 20% of America’s total oil usage in 2003. A similar situation exists on a global basis: world oil discovery peaked in the mid-1960s and, as of 2003, 80% of the world oil production at that time was extracted from caches discovered before 1973 [33].

Considering the previous statistics, saying that the U.S. oil future looks ominous may be an understatement. The United States continues to rely heavily on oil for energy while only discovering nominal new amounts each year, resulting in a gradual but real downward trend in current oil reserves. The crippling effect of the 1973 OPEC oil embargo evidenced the dependence of the U.S. lifestyle on oil, yet the United States
Figure 1.4: United States Yearly Oil Reserves and Discoveries (Statistics from [49])

increases its precariousness by importing a significantly larger percentage of its oil today. The advent of the oil “rollover,” or when the world oil demand permanently exceeds the world’s oil supply, is being accelerated by the rapid development of 3rd world countries. Clearly, the United States needs to seriously address the oil issue in order to sustain a culture of freedom of mobility for the future. Short of major lifestyle changes (which may soon become a reality anyway), there exist only a few ways to impede the depleting trend of the U.S. oil reserves: increase dependence on foreign nations for oil, use alternative fuels in place of oil, and/or embrace technologies that decelerate the rate of oil consumption.
1.2 Challenge X Program

Among other objectives, the Challenge X program strives to emphasize the use of alternative fuels and advanced technologies that decrease the dependence on oil for mobility. Officially started in May of 2004, Challenge X is a three-year series among seventeen North American universities with the goal of re-engineering a 2005 Chevrolet Equinox for improved fuel economy and reduced emissions while maintaining performance, utility, safety, and consumer acceptability. Headline sponsored by the U.S. Department of Energy and General Motors Corporation (GM), the program also attracts sponsorship from many other industry and government leaders. Teams achieve the goals of the competition by utilizing advanced hybrid powertrains, novel control strategies, alternative fuels, lightweight materials, and innovative emission control techniques. Challenge X emphasizes the development and implementation of these technologies by following a progression that is representative of GM's Global Vehicle Development Process. Therefore, the first year of the competition stresses the validation of the chosen hybrid systems in an out-of-vehicle environment. The goal is to then have properly engineered systems ready for integration into the actual Equinox vehicle in years two and three. Overall, the Challenge X program is a joint effort of government, industry, and academia to address and endorse solutions to energy and environmental issues related to the automobiles of today. Further information on Challenge X can be found at http://www.challengex.org.
1.3 Brief Hybrid-Electric Vehicles Overview

1.3.1 Definition of a Hybrid Vehicle

A hybrid vehicle is defined as a vehicle that incorporates two or more energy storage devices (at least one irreversible and at least one reversible) from which energy can be extracted and transformed into mechanical power for vehicle propulsion purposes. These energy storage devices may include a:

- Fuel tank (gasoline, Diesel fuel, ethanol, hydrogen, compressed natural gas, etc.).
- Electrical battery or supercapacitor (numerous chemistries available).
- Flywheel battery.
- Hydraulic accumulator.

1.3.2 Classes of Hybrid Vehicles

Hybrid vehicles have been traditionally defined as residing in one of two classes: series or parallel. A series hybrid implies a configuration in which energy flows combine and interact on a level other than mechanical. For instance, a series vehicle may include an internal combustion engine that turns an electrical generator which connects to the same high-voltage bus as an electrical battery. Electrical power is then supplied to or drawn from an electric machine mechanically coupled to the road. A parallel hybrid configuration is one that combines power flows on a mechanical level on the drivetrain. For example, a parallel vehicle may include an electric machine mechanically coupled to the engine crankshaft. Power to the road may then originate from the engine, electric machine, or both. Certain hybrid configurations
cannot be classified as purely series or parallel; for instance, architectures with innovative transmissions or other means of coupling two or more electric machines can potentially act as a series hybrid, parallel hybrid, or any combination in between. Although the term originated solely in association with a particular transmission design, any series-parallel hybrid configuration can be generally referred to as a power-split design.

1.3.3 Advantages/Disadvantages of Hybrid Vehicles

The practicality of hybrid vehicles stems from several advantages as compared to conventional automobiles; notably:

- The primary power converter (e.g., internal combustion engine) can be down-sized to typically result in a higher efficiency at a given operating point. During large and intermittent power requests, the secondary power converter (e.g., electric machine) can supplement the power lost from the downsizing of the primary power converter.

- The secondary power converter can add or subtract power to or from the drivetrain to shift the operating point of the primary power converter to that of a higher efficiency.

- The secondary energy source (e.g., electrical battery) can facilitate a disablement of the primary energy converter (e.g., internal combustion engine) during vehicle idle or other low power request situations (e.g., engine start/stop to mitigate engine idling losses).
A percentage of the vehicle's kinetic energy can be recuperated and stored in the secondary energy storage device for use during future tractive situations.

Hybrid vehicles also have a few disadvantages. The addition of a second energy storage system and power converter usually results in a significant mass increase of the vehicle as well as consuming space normally reserved for cargo. The inherent complexity of hybrid vehicles necessitates extensive control strategies to realize the advantages of the hybrid system while managing the power flows to be as transparent to the driver as possible. Further, the complexity of the system results in complex problems; hybrids typically cannot be fixed in the garage of the common consumer and even dealerships need special training to properly educate their mechanics on hybrid diagnostics.

1.3.4 Current State of the Hybrid Vehicle Market

The concept of hybrid vehicles themselves are nothing radically new: the last decade of the 1800s and the first of the 20th century spawned ideas and productions related to hybrids from several locations across the globe. Notably, American engineer H. Piper filed a patent for a gasoline-electric hybrid powertrain in 1905 with the sole hope of increasing acceleration (bettering fuel economy seemed not to be a priority). Though the idea of hybrids never completely disappeared throughout the 20th century, it was perhaps slightly ahead of its time as the light-duty automotive market was effectively dominated by gasoline-powered conventional vehicles. However, the surging technology of the past few decades (digital control, computer systems, battery improvements, etc.), combined with public concern over intermittent oil crises and
resulting government initiative, worked to bring hybrids to a permanent level of production reality. Now in the 21st century, every new model year seems to yield at least one new hybrid vehicle. Most of these are conversions from their conventional stock counterparts and all of them are hybrid-electric vehicles (HEVs) with a fuel tank and electrical battery as the energy storage devices. Hybrid-electric vehicles tend to be (and deservingly so) slightly more expensive than equivalent conventional automobiles. However, most consumers who purchase hybrids ignore the slight initial cost and instead focus on the reduced trips to the fuel station, whether or not the savings in fuel ever expend the difference. Most other HEV drivers are technologically savvy consumers pursing state-of-the-art products, tend to be motivated by current environmental and energy issues, or both. Either way, it is evident that hybrid-electric vehicles currently control a share of the market that can only grow as the technology becomes more common to the consumer and as the bleak oil situation incites increasing fuel prices. However, it should be noted that HEVs alone only reduce the rate of oil consumption and, in effect, only prolong the presence of the oil issue. Therefore, certain automotive industry leaders are considering hybrid-electric vehicles as a gateway between conventional vehicles and true sustainable mobility, where these automobiles of the future will be powered by alternative fuels or a renewable resource.
CHAPTER 2

VEHICLE ARCHITECTURE SELECTION

The Ohio State University (OSU) Challenge X team (hereafter referred to as the Ohio State team) engaged in an extensive selection process to decide upon a vehicle configuration for development in the Challenge X program. This chapter thoroughly describes the analysis and thought process that resulted in the chosen hybrid-electric architecture with emphasis on those parts exclusively completed by the author.

2.1 Vehicle Technical Specifications

Consistent with the Challenge X vehicle development procedure based upon GM’s Global Vehicle Development Process, the Ohio State team first defined a set of vehicle technical specifications (VTS). These specifications serve as vehicle goals that motivate and steer the design and development of the vehicle’s architecture and control strategy. Ohio State’s final VTS are listed in Table 2.1.

2.2 Fuel Selection

Challenge X teams were given the option of choosing from four fuels for use in the competition: reformulated gasoline, bioDiesel, ethanol, and gaseous hydrogen. Prior
<table>
<thead>
<tr>
<th>Description</th>
<th>Stock Equinox</th>
<th>Competition Goal</th>
<th>OSU VTS</th>
</tr>
</thead>
<tbody>
<tr>
<td>IVM - 60 mph</td>
<td>~ 8.9 s</td>
<td>≤ 9.0 s</td>
<td>≤ 10.0 s</td>
</tr>
<tr>
<td>50 - 70 mph</td>
<td>~ 6.8 s</td>
<td>≤ 6.8 s</td>
<td>≤ 7.0 s</td>
</tr>
<tr>
<td>Vehicle curb weight</td>
<td>3796 lbs</td>
<td>≤ 4400 lbs</td>
<td>≤ 4275 lbs</td>
</tr>
<tr>
<td>Combined mpgge</td>
<td>~ 24.9 mpgge</td>
<td>≥ 32.0 mpgge</td>
<td>≥ 35.0 mpgge</td>
</tr>
<tr>
<td>Highway range</td>
<td>~ 511.5 mi</td>
<td>≥ 200 mi</td>
<td>≥ 275 mi</td>
</tr>
<tr>
<td>Passenger capacity</td>
<td>5 passengers</td>
<td>5 passengers</td>
<td>5 passengers</td>
</tr>
<tr>
<td>Emissions level</td>
<td>Tier 2 bin 5</td>
<td>Tier 2 bin 5</td>
<td>Tier 2 bin 5</td>
</tr>
<tr>
<td>Trailering capability</td>
<td>3500 lbs</td>
<td>2500 lbs</td>
<td>2500 lbs</td>
</tr>
<tr>
<td>Starting time</td>
<td>≤ 2.0 s</td>
<td>≤ 5.0 s</td>
<td>&lt; 2.0 s</td>
</tr>
<tr>
<td>Start/stop</td>
<td>Engine off at vehicle idle. All accessories run electrically.</td>
<td></td>
<td></td>
</tr>
<tr>
<td>4WD mode</td>
<td>Four-wheel drive (4WD) for poor-traction situations.</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table 2.1: Vehicle Technical Specifications

to any further architecture analysis, the Ohio State team decided to make a fuel selection. Fixing this degree of freedom allowed a broader study of vehicle configurations with the particular chosen fuel and thus resulted in a more specific architecture determination. Additionally, this procedure mitigated redundant analysis that may have been encountered when considering vehicle architectures on a fuel-dependent basis.

Since 1995, an addition to the Clean Air Act has required cities with outstanding levels of air pollution to distribute reformulated gasoline (RFG) instead of conventional gasoline. Cities may additionally choose to use RFG on their own, and today approximately 30% of the gasoline sold in the United States is reformulated [52]. RFG entails a reduction of sulfur and benzene from that of conventional gasoline with an addition of oxygen. The initial goal of RFG was to reduce both smog-forming and toxic air pollutants, and a recent study indicates that RFG does indeed significantly
reduce particulate emissions [17]. The same study also shows engine-out reductions in hydrocarbons and oxides of nitrogen (NO\textsubscript{x}) emissions with negligible changes in carbon monoxide as compared to conventional gasoline.

BioDiesel is produced from plant or animal fat-based oils and is thus considered a renewable resource. The bioDiesel option for the Challenge X competition is a blend of 26% bioDiesel and 80% low-sulfur Diesel (B20). Modest bioDiesel blends are some of the only alternative fuels to require little or no engine modification relative to its conventional counterpart. These blends tend to reduce hydrocarbon, carbon monoxide, and particulates while showing negligible or slight increases in NO\textsubscript{x} emissions relative to normal Diesel fuel [50]. Performance with bioDiesel is comparable to conventional Diesel; however, the higher freezing temperatures of rich bioDiesel blends may stem problems at cold engine start situations.

The ethanol option is a blend composed of 85% pure Ethanol and 15% RFG (E85). Ethanol is also known as a renewable resource as it can be produced from a variety of agricultural products and food wastes which contain sugar, starch, or cellulose [51]. On an engine level, there are emission reductions of carbon monoxide, particulates, and nitrogen oxides. Although Ethanol has a higher octane level than conventional gasoline, its lower heating value is about 33% less [15] and thus contributes to a decrease in vehicle range for a given fuel tank volume. Additionally, ethanol has material compatibility issues related to components found in traditional fuel systems. Overall engine performance using E85 is equivalent to a conventionally fueled engine.

Gaseous hydrogen (H\textsubscript{2}) is also available for use in the Challenge X competition in combination with a fuel cell or hydrogen internal combustion engine. Fuel cells are not constrained by the thermodynamic Carnot efficiencies that govern combustion
engines and are instead limited by their Nernst potential with a much greater potential efficiency. Increased efficiency (albeit of a lesser degree) also occurs with hydrogen engines as compared to their conventional counterparts. With regards to emissions, hydrogen engines relatively decrease carbon dioxide [7] but NOx remains an issue. Hydrogen has a large lower heating value of 120 MJ/kg [15] but is also the lightest known element; therefore, even when stored as a highly compressed gas or cryogenic liquid and used with a more-efficient fuel cell, hydrogen automobiles tend to have a decreased range relative to conventionally fueled vehicles. On a well-to-tank basis, hydrogen presents many challenges with regards to production, transportation, and distribution. As hydrogen does not occur naturally on Earth, energy must be expelled to produce it, and thus hydrogen as an energy carrier is warranted only by unique benefits associated with its use [36].

With support from the U.S. Department of Energy, Argonne National Laboratory has developed the Greenhouse Gases, Regulated Emissions, and Energy Use in Transportation (GREET) tool. This program models and compares well-to-wheel energy and emissions from a variety of fuels and vehicle technologies. In particular, GREET is a unique and valuable tool to investigate prevehicle energy and emissions which are commonly overlooked when solely considering a vehicle's fuel economy and emission rating. The GREET analysis by the Ohio State team considered four passenger car configurations with long-term vehicle technologies; namely, a hybrid-electric vehicle with RFG, a HEV running on B30, a HEV on E85, and a fuel cell vehicle (FCV) using gaseous hydrogen. Figure 2.1 displays the percent change of energy relative to a conventional vehicle running on reformulated gasoline. On a well-to-tank (WTT)
analysis, the energy expenditure to produce ethanol significantly exceeds that to refine RFG as does the generation and transportation of pure hydrogen, although on a lesser extent. Oppositely, the RFG and bioDiesel hybrids use less energy well-to-tank mainly because the more-efficient hybrids use less fuel in the vehicle and thus require less to be reformed and transported. All configurations demonstrate decreases in used energy on a tank-to-wheel (TTW) basis as a result of the hybrid-electric and fuel cell technologies. The Diesel hybrid, with a greater engine efficiency due in part to a higher compression ratio and lack of engine pumping losses, outperforms the RFG and ethanol hybrids. On an overall well-to-wheel (WTW) energy analysis, the most-efficient fuel cell vehicle slightly improves upon that of the bioDiesel-electric hybrid configuration.

![Energy Percent Change](image)

**Figure 2.1: GREET Energy Analysis**
The greenhouse gas (GHG) emissions of each configuration were also analyzed relative to a conventional vehicle run on RFG (Figure 2.2). Being primarily derived from corn which removes carbon dioxide from the environment, ethanol results in a net reduction of greenhouse gases on a WTT basis. Oppositely, the GHGs emitted during the production of hydrogen (primarily from natural gas reformation) more than double that of reformulated gasoline. On the more heavily weighted TTW aspect, bioDiesel significantly reduces GHGs as a consequence from lean Diesel combustion. The FCV emits only water and thus reduces vehicle GHG emissions by a full 100%. Considering a well-to-wheel basis, the fuel cell vehicle again shows the largest decrease relative to a conventional vehicle, while the E85 HEV and B20 HEV also significantly reduce GHG emissions.

![GREET Greenhouse Gas Percent Change](image)

**Figure 2.2: GREET Greenhouse Gas Emissions Analysis**
Figure 2.3 displays WTW NO\textsubscript{x} emissions using the GREET tool. The derivation of E85 emits nitrogen oxide emissions over twice as much as any other configuration on a WTT basis. The notoriously difficult aftertreatment for Diesel engines results in large amounts of tailpipe NO\textsubscript{x} emissions, and thus the B20 HEV has the largest TTW NO\textsubscript{x} emissions of those configurations considered. On a WTW basis, the fuel cell configuration releases the least amount of NO\textsubscript{x}, while the B20 and E85 HEVs both result in a net increase relative to the conventional vehicle.

The Ohio State team also gave significant consideration to other intangibles during the fuel selection process. The team, as well as the support personnel at the Ohio State Center for Automotive Research (CAR), lacks sufficient experience in implementing fuel cell technology into practical vehicle applications. CAR is currently in the process
of expanding its fuel cell laboratories, but the level required for the Challenge X competition will not be completed in a timely manner. Further, the scarce availability of adequate fuel cell stacks greatly limits unique architecture design and flexibility. Ethanol poses similar issues with familiarity and experience. Conversely, CAR has a strong research thrust with both Diesel engine control and exhaust aftertreatment. Additionally, the team utilized a Diesel engine in the previous HEV competition and feels more comfortable and proficient with Diesel fuel compared to the other available choices. The GREET analysis shows the bioDiesel-electric hybrid as one of the least consuming fuels on a WTT, TTW, and overall WTW energy basis. The B20 HEV also ranks preferably with respect to greenhouse gas emissions. NOx emissions are an issue with any Diesel configuration, but this problem can be mitigated by employing appropriate exhaust treatment technologies such as those mentioned later in this thesis. Therefore, the Ohio State team selected B20 bioDiesel as the fuel for use in the Challenge X competition.

2.3 Initial Brainstorming

After bioDiesel was selected upon as the fuel of choice, the Ohio State team conceptualized and presented several potential hybrid vehicle architecture designs and ideas. Summarizing, these were:

1. “The Rudy Diesel Special”: A large Diesel engine with a starter/generator electric machine coupled directly to the crankshaft on the accessory side. Minor regenerative braking, no engine load shifting, and stock driveline retention. Extensive exhaust treatment required.
2. "Dyno-on-Wheels": A large Diesel engine running homogenous-charge compression-ignition (HCCI) with a single calibrated power trajectory. Large generator and electric machine and modest high-voltage battery pack. Oxidation catalysts only.

3. Through-the-Road (TTR): A medium-sized Diesel engine in the front and electric machine in the rear for strong hybridization. Extensive exhaust treatment required.

4. ISA: A strong integrated starter/alternator (ISA) packaged between the transaxle and medium-sized Diesel engine with a clutch between the ISA and engine. Extensive exhaust treatment required.

The team performed an intuitive high-level evaluation of the pros and cons of these potential architectures as a basis for literature search topics and subsequent in-depth analysis. The Rudy Diesel Special configuration presents a very low-risk solution with relatively simple packaging and integration. Its large Diesel would be comparable to the stock Equinox in terms of towing, acceleration, and drivability. However, this plan, at best, would result in modest fuel economy gains and perhaps lacks some of the ambition and innovation that the Challenge X competition encourages. The Dyno-on-Wheels architecture offers flexible packaging and smooth drivability while greatly reducing emissions through HCCI combustion. Fuel economy should improve with the series hybrid design, but acceptable towing and acceleration would require large and powerful electric machines. The main disadvantages with the Dyno-on-Wheels architecture are the unfavorable net weight addition through these large electrical components and the overall high technical risk associated with
any series configuration. The Through-the-Road configuration conveniently separates the conventional and electrical powertrains from a direct mechanical coupling in the vehicle and thus would facilitate a certain independence regarding implementation. This parallel arrangement often results in large fuel economy gains through significant engine downsizing and strong hybridization, of which the extent would be limited from continuous performance requirements. Also, control would be challenging in instances when the axles may be rotating at different speeds (as in off-road or inclement weather situations). The ISA arrangement differs from the Rudy Diesel Special in the sizing and location of its components. Packaging of a second clutch and thin ISA between the engine and transaxle in a transverse fashion would be difficult at best and perhaps not even possible. This configuration would also result in significant fuel economy gains but may lack power needed for acceptable performance.

2.4 Hybridization Analysis

Modeling and simulation are important tools in designing a vehicle architecture and selecting components so as to meet vehicle technical specifications. As such, the Ohio State team initially used the Powertrain Systems Analysis Toolkit (PSAT) supplied by Argonne National Laboratory to estimate relative changes in fuel economy and performance as a function of several parameters. Specifically, to gain insight to the benefits of hybridization, the stock Chevrolet Equinox model was transformed by varying the engine displacement and inserting an electric machine and high-voltage battery pack in a parallel arrangement. In an effort to control variables as much as possible, all other component models (transmission, differential, vehicle body, etc.)
were left unaltered. The simulation was run through several driving cycles and acceleration tests with varying combinations of engine and electric machine size. Results relative to stock Equinox data for an Federal Urban Driving Schedule (FUDS) and Federal Highway Driving Schedule (FHDS) are illustrated in Figure 2.4.

![Graph showing fuel economy analysis](image)

**Figure 2.4: PSAT Fuel Economy Analysis**

A strong HEV expectantly increases fuel economy over the urban cycle, as a large electric machine can recover vehicle kinetic energy of a greater magnitude and frequency during the stop-start behavior of the FUDS cycle. Further, larger electric machines provide more opportunity for significant engine load shifting to facilitate engine operation in a higher-efficiency region. High-powered (and large-displacement) engines use more fuel at most operating conditions and thus decrease thermal efficiency at most operating conditions. Highway operation suppresses the full potential.
of a hybrid as the quasi-steady-state driving (already at medium to high load) limits opportunity for significant load shifting or regenerative braking. However, even a simple on-off control strategy can lead to fuel economy gains at steady-state driving and the PSAT control strategy does indeed result in modest fuel economy gains on the FHDS cycle. Notably, PSAT indicates a preference towards a medium-sized electric machine for highway driving but a strong inclination for a large electric machine during urban driving.

Figure 2.5: PSAT Combined Fuel Economy Analysis

As the vehicle technical specifications for the Ohio State hybrid vehicle are based on a combined EPA fuel economy, the appropriately weighted average of the FUDS and FHDS fuel economy values are shown in Figure 2.5. A similar trend as noticed previously for the FUDS cycle is again present: in general, a smaller engine and larger
electric machine result in better fuel economy. This agrees with intuitive logic, as the downsized engine alone brings more operating points into a higher efficiency region of the engine while consuming less fuel at most operating conditions. Then, hybridization provides supplemental power at large power demands while adding benefits such as engine load shifting and regenerative braking.

Surprisingly, PSAT simulations suggest a strong independence between 0-60 mph acceleration times and electric machine size (the same pattern occurred for 50-70 mph acceleration). Intuition and simple calculations reveal that more power to the road (from the tractive motor) should result in a quicker acceleration; however, extensive investigation into PSAT did not expose the true reason for the aberration. The investigation did reveal that an electric machine of size greater than about 35 kW unsuccessfully provides its maximum power to the road. A device upstream of the electric machine (e.g., the high-voltage battery pack) is most likely the limiting factor, but then this conclusion surfaces questions about the legitimacy of the seemingly strong dependence of fuel economy on electric machine size as described previously. Nevertheless, the Ohio State team recognizes that a larger electric machine (with an appropriately sized high-voltage battery pack) will provide more power and result in quicker acceleration given a suitable control strategy.

2.5 Towing Analysis

Conventional vehicles specify a maximum trailer weight rating with the assumption of being able to successfully traverse the California "Baker grade" (55 mph at a 7% grade for 15 miles). This requirement is unambiguous for conventional vehicles which typically rely on continuous power from an engine. However, the main
Figure 2.6: Towing Road Load Analysis

fuel economy benefit of a hybrid results from the downsizing of the engine, so the vehicle’s continuous power (in the truest sense) decreases. With the assumption of a Diesel engine capable of at least 100 kW (a significant reduction from the stock Equinox gasoline engine), a heavy Equinox hybrid with an aerodynamically poor 2500 lb trailer could continuously run the baker’s grade at approximately 50 mph (as in Figure 2.6. The relatively modest addition of approximately 25 kW from an electric machine could then intermittently meet the Baker’s grade velocity requirement of 55 mph; however, with a reasonably sized high-voltage battery pack, this vehicle could not sustain the velocity and grade for a full 15 miles of travel. The Ohio State team recognizes the inherent trade-offs with a hybrid vehicle and realizes that a hybrid is typically not meant as a pure towing vehicle. However, most sport-utility vehicle drivers (hybrid or not) do expect a certain amount of minimal towing capability. The
Challenge X competition tests towing ability on a course with up and down grades of reasonable magnitude and length. Therefore, although it will not satisfy continuous Baker grade requirements, the Ohio State team decided that a minimum engine size of 100 kW and supplemental electrical power of at least that mentioned previously is sufficient to pass the Challenge X towing test and meet the towing needs of hybrid Equinox drivers.

2.6 Mass Analysis

From strictly a power-at-the-wheel standpoint, three primary and independent variables exist which can be altered to vary the road load energy for a given driving cycle: effective vehicle frontal area \((C_d \cdot A_f)\), rolling resistance coefficient, and vehicle mass. Energy sensitivities with respect to these variables about their 2005 stock Equinox values were found and are displayed in Table 2.2 for both the conventional vehicle and a hybrid capable of full regeneration at any negative power demand.

<table>
<thead>
<tr>
<th>With Respect to</th>
<th>FUDS</th>
<th>FHDS</th>
</tr>
</thead>
<tbody>
<tr>
<td>Drag Coefficient</td>
<td>0.225</td>
<td>0.361</td>
</tr>
<tr>
<td>Rolling Coefficient</td>
<td>0.333</td>
<td>0.594</td>
</tr>
<tr>
<td>Mass</td>
<td>0.775</td>
<td>0.639</td>
</tr>
</tbody>
</table>

Table 2.2: Road Load Energy Sensitivities to Vehicle Parameters

The high velocities encountered during a FHDS cycle result in the drag coefficient being the most sensitive variable on an energy basis. The city cycle, with its relatively low speeds and frequent stopping and starting, is most sensitive to changes in vehicle
mass. Notably, the sensitivity with respect to mass decreases after the vehicle is hybridized for both driving cycles. This is a consequence of the typical ability of a hybrid to have a regenerative braking capability, in which the electric machine functions as a generator to apply a negative torque to the driveline so as to recover some of the vehicle's kinetic energy for storage in the high-voltage battery. The full potential of this regenerative ability on a road-load energy basis is seen in Figure 2.7, in which a 29.7% reduction in energy usage can be realized by recovering the inertial energy instead of dissipating it through frictional braking. Thus, as the ability to recover inertial energy increases, the impact of mass on energy consumption becomes less significant for hybrids as relative to conventional vehicles [1]. Therefore, although hybridization of an automobile involves the addition of several hardware components relative to its conventional counterpart, a proven hybrid-electric system which strives for efficiency with regards to powertrain operation (including regenerative braking) is significantly more important than vehicle mass. For instance, this is evidenced by the Toyota Prius which has increased in both mass and fuel economy with each subsequent model year. However, it is noted that the full regenerative potential of a vehicle can never be attained due to power limitations and inefficiencies of the power electronics. Also, large curb weights limit towing capability up a grade as well as directly increasing inertial, grade, and rolling resistance loads. Further, Table 2.2 shows that energy consumption is significantly sensitive to mass for a variety of driving conditions. Therefore, the Ohio State team has recognized the inherent benefits associated with a weight saving campaign and will exert a valiant effort towards reducing or eliminating mass from the vehicle as possible. Several possibilities, along
with the estimated mass impact of converting a 2005 Equinox into the anticipated Ohio State hybrid-electric design, are outlined in Appendix Tables A.1 and A.2.

![Energy Potential from Hybridization](image)

**Figure 2.7: Road Load Energy Analysis**

### 2.7 Decision Matrix

The selection of a vehicle architecture involves many factors of differing weightings as derived by the team’s goals and vehicle technical specifications. To assist in the evaluation of advantages and disadvantages, the Ohio State team constructed a decision matrix which assigned numeric scores to identified metrics for several potential vehicle architectures. The architectures analyzed are the same as those from the initial brainstorming session as described previously in Section 2.3. The decision matrix considered the degree to which each architecture could meet the metrics, the
technical risk associated with doing so, and the business risk related to procurement of the necessary components. A simplified schematic representation of the decision matrix can be seen in Figure 2.8 with the fully-detailed decision matrix located in Appendix Figure A.1.

![Architecture Selection Chart]

Figure 2.8: Architecture Selection Chart

With regards to fuel economy, all architectures would improve upon the stock vehicle solely due to the conversion to a downsized Diesel engine (of varying degrees). The ISA configuration further contributes to enhanced fuel economy through significant engine load shifting, the ability for a start-stop strategy, and mild regenerative braking. The strongly hybridized TTR architecture features a rear electric transaxle that is directly coupled to the road for a more ideal regenerative braking capability. Further, the TTR option provides opportunities for engine load shifting; however, the
connection of the engine to the electric machine through the road and both drivetrains result in energy losses absent from that of the ISA architecture. The small electric machine in the Rudy Diesel Special provides the ability for engine start/stop along with minimal engine load shifting. A large Diesel engine combined with large electric machines significantly adds mass to the Dyno-on-Wheels series configuration, but a fuel economy increase still results from the engine's quasi-independent operation with respect to the vehicle.

As commonly derived from a renewable resource (soybean oil), pure bioDiesel is carbon neutral on a well-to-wheel basis. Combined with the low greenhouse gas emissions inherent to the highly efficient (relatively speaking) Diesel combustion, all four architectures will achieve improved GHG emissions with a low technology risk. Oppositely, the conventional lean combustion of Diesels cause difficulties in the aftertreatment of NOx and particulates. Although several options (as discussed later) can potentially achieve the Tier 2 bin 5 emission level as desired, the final tailpipe emission levels will most likely still be greater than that of the gasoline-powered stock Equinox. The nature of homogeneous-charge compression-ignition could bring the series architecture to NOx emissions levels of the same order of magnitude as that of the stock vehicle; however, the technical risk of a functional HCCI system is very high. Further, HCCI combustion increases engine-out hydrocarbon emissions, and thus the aftertreatment system for the Dyno-on-Wheels architecture would solely consist of several oxidation catalysts.

With proper transaxle ratios and its large low-end torque, the Rudy Diesel Special should accelerate as well as the stock vehicle and is also ideal for continuous
towing. The Dyno-on-Wheels HEV would require overly large powertrain components to maintain continuous towing and is thus anticipated to be worse than stock. This series hybrid’s large drivetrain mass will most likely prevent improved acceleration even while considering the large and smooth low-end torque from its electric machine. The remaining two configurations have smaller Diesel engines that limit continuous towing as discussed previously. However, both have electrical systems that provide sufficient supplemental power, and thus both the ISA (with torque assist) and TTR (with electric launch) architectures should exceed that of the stock vehicle for intermittent quick accelerations or large power requests.

The consumer acceptability aspect of a vehicle architecture consists of several different factors. With the exception of the series hybrid, which lacks a mechanical connection between engine and road, all of the vehicle configurations can be packaged (albeit, not always easily) to maintain a certain amount of relation to a conventional vehicle. Therefore, packaging of the series architecture may induce less consumer confidence than more conventionally packaged architectures. Further, the Dyno-on-Wheels configuration’s HCCI operation may cause noise and vibration issues, but drivability (with respect to driveline oscillations) would improve via smooth power flow from the electric machine. Also, the series design is probably the most innovative architecture considered, while the mild Rudy Diesel Special presents a very simple solution. All of the hybridized configurations add mass to the stock Equinox which results in varying degrees of decreased vehicle response and handling ability.

In a competition such as Challenge X, it is very important to recognize limitations from a procurement and technical standpoint and then evaluate the appropriate risks.
The business risk associated with achieving the previously stated metrics is dependent upon procuring the appropriate components. All components require a certain amount of effort to obtain, and the business risk changes as more components are found and details about these components discovered. However, certain components inherently have a high business risk, such as the relatively rare electric machines and inverters for automotive applications. On the other hand, off-the-shelf items or components available directly through dealerships or donations have a low business risk. The technical risk deals with the ability and required effort to get said components to achieve the degree of effectiveness as stated in the vehicle technical specifications. Overall, the series hybrid-electric design has the largest technical risk, as its fundamental nature requires function across its entire platform for any vehicle operation. Conversely, the Rudy Diesel Special mainly relies on its Diesel engine to meet competition goals, and thus the technical risk is low. Technical risk associated with the ISA arrangement is primarily the result of very tight packaging under the hood, while the TTR architecture allows slightly more space and flexibility. Additionally, the stronger hybrids require more control and integration to effectively meet the vehicle’s technical specifications which further increases technical risk.

2.8 Vehicle Architecture Selection

After thoughtful consideration of the decision matrix and other analyses outlined in this report, the Ohio State team made a decision regarding the vehicle architecture to be pursued for the Challenge X competition. The design is a combination of the Rudy Diesel Special and TTR architectures as analyzed previously in the decision matrix. The Rudy Diesel Special alone meets many of the technical goals,
but the Ohio State team felt its simplicity contradicts the spirit of the competition. Meanwhile, the TTR is a strong hybrid-electric design that adds a level of ambition and uniqueness stressed with Challenge X. The final architecture selection strives to combine these two designs to maximize overall benefits in a power-split configuration that can potentially act as a series hybrid, parallel hybrid, or any combination in between. Further, the chosen architecture provides the potential for vehicle operation in a number of advantageous modes which will be further described later in this thesis. The architecture is illustrated in Figure 2.9 and can be summarized as including a:

- Small Diesel internal combustion engine (hereafter also referred to as the ICE) and automatic transaxle driving the front axle.
- Integrated starter/alternator (hereafter also referred to as the ISA) directly and rigidly coupled between the engine and transaxle.

- Larger tractive electric machine (hereafter also referred to as the EM) connected to the rear axle through a single-speed differential gearbox.

- High-voltage (HV) battery pack serving both electric machines.

- Separate inverter for each electric machine.

- Switchbox to route the HV electrical power flow.

- DC-DC converter to power the 12V system.

2.9 Component Selection

The Ohio State team has further decided upon specific components to compose the vehicle architecture as defined previously. A brief description of the components with appropriate reasoning subsequently follows, and additional component specifications can be found in Table 2.3.

As discussed previously, simulations show beneficial trends in fuel economy as the vehicle degree of hybridization increases; that is, as engine displacement decreases and power electronics intensify. However, analysis also evidences a downsizing limit to maintain acceptable towing capabilities. The 1.9L Fiat engine is the current state-of-the-art in Diesel technology with a multijet and common-rail fuel injection system, variable-swirl valve actuation, and a Euro 4 emissions rating. The engine produces 305 Nm at a low 2000 rpm and boasts 103 kW peak power, and thus the Ohio State team has chosen this engine to significantly increase fuel economy while providing the vehicle with an ample amount of continuous power.
<table>
<thead>
<tr>
<th>Component</th>
<th>General</th>
<th>Notes</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fuel</td>
<td>BioDiesel</td>
<td>Supplied by BP; 20% bioDiesel, 80% Low sulfur Diesel</td>
</tr>
<tr>
<td>Engine</td>
<td>1.9L Diesel</td>
<td>Fiat multijet; used in Opel Vectra; 4 cylinders; 16 valves; turbocharged; Euro 4; 103 kW @ 4000 rpm; 305 Nm @ 2000 rpm; comp. ratio = 17.5:1; bore = 82.0 mm; stroke = 90.4 mm</td>
</tr>
<tr>
<td>Transaxle</td>
<td>6-speed automatic</td>
<td>Aisin-Warner; used with Fiat 1.9L Diesel engine in Opel Vectra</td>
</tr>
<tr>
<td>Integrated starter/alternator</td>
<td>Permanent magnet</td>
<td>Electric machine from IMA system in Honda Accord Hybrid; 12 kW peak; 68 mm in width</td>
</tr>
<tr>
<td>Tractive electric machine</td>
<td>Three-phase AC induction</td>
<td>Delphi electric machine and Delco inverter from 1997 GM EV1, 102 kW @ 7000 - 13,000 RPM; 150 Nm @ 0 - 7000 RPM; gearbox ratio = 10.946:1</td>
</tr>
<tr>
<td>High-voltage battery pack</td>
<td>NiMH</td>
<td>Panasonic Prismatic modules; used in Toyota Prius; 7.2V nominal; 6.5 Ahr</td>
</tr>
</tbody>
</table>

Table 2.3: Component Selection and Specifications

Transmissions are usually one of five types: electronically-variable (EVT), automatic, continuously-variable (CVT), manual, or automated-manual (AMT). EVTs (such as in the Toyota Prius) typically couple one engine with two electric machines through a series of planetary gear sets and thus favorably result in the most independent engine operation relative to the road load. The Ohio State team initially considered developing an EVT for the Challenge X competition but decided the complexity and required resources would present an overly high technical risk. Therefore, the EVT was eliminated as a viable option and is noticeably absent from the vehicle architecture decision process as explained previously. The CVT eliminates discrete
gear ratios and also allows a certain level of speed independence from the road; however, the efficiency is still less than that of manual transmissions and its typical low torque rating that may not suit a Diesel engine. Pure manual transmissions tend to be highly efficient and simple with regards to integration but are not commonly preferred by sport-utility vehicle (SUV) drivers. Further, manual transmissions permit the driver to completely decouple the engine from the driveline (through the clutch) and/or have influence on the engine speed (through the transmission gear selection); this freedom is not ideal for HEVs which typically operate more efficiently when in a greater amount of control of the drivetrain. The high efficiency of manuals with the consumer acceptability and control of an automatic can be found in an automated-manual transmission. These transmissions are still relatively uncommon to the automotive market and, although the Ohio State team took an initial effort to pursue an AMT, it was decided that the high technical and business risks made the automated-manual transmission an unfeasible choice for the Challenge X competition. The current North American SUV market is dominated by automatic transmissions with a high level of consumer acceptability. The efficiency of automatic transaxles (due to the torque converter) lags that of manual transmissions but the controllability is appropriate considering the chosen power-split vehicle architecture. Specifically, the automatic transaxle facilitates a start-stop strategy without the complexity of automating the clutch in conjunction with a manual transmission. After considerations such as those depicted in Figure 2.10, the Ohio State team chose the 6-speed automatic transaxle that accompanies the Fiat engine in the Opel Vectra.
The appropriate Diesel gearing and engine controller calibration specific to the 6-speed transmission make the Fiat engine and the automatic transaxle an obvious and appropriate match.

Figure 2.10: Transmission Selection Chart

Integrating a start-stop strategy significantly complicates control as the transition between engine states must be as transparent to the driver as possible. However, with certain engines and architectures, fuel economy can notably increase (especially in urban driving situations) from eliminating fuel flow to the engine during vehicle idle situations while powering all accessories electrically. The ISA will quickly and smoothly ramp the engine to the correct operational speed during starting and may also act as a generator to power the rear-axle motor during continuous all-wheel drive operation. To eventually act as the ISA in Ohio State's hybridized Equinox, the team choose the electric machine from the Integrated Motor Assist package (IMA) in the
production Honda Accord Hybrid. This electric machine has a “pancake” shape that facilitates a transverse packaging and produces enough power for engine load shifting or tractive electric machine powering. Further, the Honda ISA is capable of starting a 3.0L V6 gasoline engine in the Accord Hybrid which indicates a large potential for its ability to start a smaller, 4 cylinder Diesel engine. The EM will be the more-powerful electric machine that significantly increases the degree of hybridization of the vehicle, aids in improved fuel economy by allowing significant regenerative braking, and delivers the power to the road during an electric vehicle launch. This unit’s direct connection to the road mitigates excessive driveline losses and thus is ideal for traction and regeneration situations. Due to its immediate availability and compact package, the Ohio State team has decided to utilize the electric machine and attached gearbox, along with a customized controller-inverter, from the previous production GM EV1 electric vehicle. Although oversized with respect to potential electrical power to or from the current high-voltage battery pack selection (as described subsequently), the EV1 machine is a proven unit that allows the opportunity for HV battery pack augmentation or the addition of supercapacitors in the future.

Nickel-metal hydride (NiMH) tends to be the battery chemistry of choice in the current generation of hybrids due to their high power and energy densities, long cycle life, and maturity. The Ohio State team decided to utilize the Panasonic Prismatic modules that are present in the production Toyota Prius hybrid-electric vehicle. Experience with these particular modules at CAR and current related research further promotes using these proven batteries. The high-voltage battery pack will consist of modules strung in series to nominally provide approximately 345V. Power flow to and from the HV battery pack and two inverters will be routed with a custom-designed
switchbox. When the ISA acts as a generator and the EM provides tractive power (or vice versa), the switchbox will connect the two inverters directly on the same high voltage bus and thus mitigate unnecessary energy flow through the HV battery pack. A high-to-low voltage DC-DC converter eliminates the need for a belt-driven alternator, efficiently powers the conventional 12V system, and facilitates a start-stop strategy.

2.10 Exhaust Aftertreatment Selection

Diesel engines offer considerable advantages over spark-ignition (SI) gasoline engines with regards to fuel consumption and greenhouse gas emissions; however, these gains are somewhat offset by certain other emission problems. Unfortunately, conventional in-cylinder techniques (such as exhaust gas recirculation) are ineffective in reducing both NOₓ and particulate matter emissions. This is known as the NOₓ-particulate matter trade-off and, consequently then, any conventional in-cylinder techniques that reduce NOₓ emissions inherently increase particulate matter and vice versa. Additionally, lean exhaust conditions (from Diesel combustion) make three-way catalytic converters used for gasoline engines ineffective as well as making NOₓ reduction particularly challenging. Diesels also emit small quantities of particulate matter which is a known carcinogen. As the Challenge X competition requires U.S. Tier 2 bin 5 emission standards (Table 2.4), unconventional combustion techniques and/or exhaust aftertreatment is a necessity.

One option for the reduction of nitrogen oxide emissions is the utilization of a lean NOₓ trap (LNT). LNT's function by initially storing NOₓ on a barium-coated catalyst

¹The great majority of the background in this section references [34].
<table>
<thead>
<tr>
<th>NMOG (g/mi)</th>
<th>CO (g/mi)</th>
<th>NOx (g/mi)</th>
<th>PM (g/mi)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.090</td>
<td>4.2</td>
<td>0.07</td>
<td>0.01</td>
</tr>
</tbody>
</table>

Table 2.4: Tier 2 Bin 5 Light-Duty Emission Standards [12]

during lean (excess oxygen) exhaust conditions and then reducing it with hydrocarbon fuel and excess carbon monoxide (left over from rich operating conditions after purging of the stored NOx). The effectiveness of the purge function is facilitated by barium nitrates that form during lean storage conditions, become unstable, and readily react with hydrocarbons under rich conditions. This technique requires excess fuel delivered to the exhaust by postinjection at the engine or an auxiliary fuel injector located in the exhaust. Inherently then, there is a slight fuel economy penalty for NOx reduction using LNTs, and the trade-off between fuel economy and emission reduction must be analyzed to find optimal LNT operation [38]. As a reference, studies have shown an 80% conversion of NOx while decreasing fuel economy by 1.8% at an equivalence ratio of 1.5 [6]. Durability issues also arise with LNTs, as high temperatures (~700 °C) are required to remove sulfur which tend to occupy those storage sites originally intended for NOx [38].

Diesel particulate filters (DPF) remove particulates from the exhaust through a variety of different filtration processes, of which the three most common are diffusion, interception, and impaction. In each of these methods, particulate matter accumulates on the filter which regularly requires regeneration to avoid excessive exhaust backpressure. Unfortunately, Diesel exhaust is generally not warm enough to initiate the regeneration process (oxidation of carbon particles) and therefore other methods
must be explored. Studies have shown that electrically heating the filter uses an unreasonable amount of power (\(\sim 27\text{ kW}\)) [11], while certain coatings or fuel additives may lower the regeneration temperature [16].

Homogeneous-charge compression-ignition is an alternative combustion mode that provides efficiencies nearly as high as compression-ignition direct-injection (CIDI) Diesel engines while greatly reducing NO\(_x\) and particulate matter emissions. HCCI engines operate with a dilute premixed charge that reacts and burns volumetrically rather than in a flame-front (as in SI gasoline and conventional CIDI Diesel engines). This simultaneous combustion results in cooler combustion and drastically reduces engine-out NO\(_x\) emissions. In a sense, HCCI combines the advantages of spark-ignition engines and CIDI engines: a well mixed charge as in SI engines and the high efficiency of CIDI engines [13]. Unfortunately, HCCI combustion is limited in cold-engine situations and by the inability to deliver large premixed charges which usually confines HCCI operation to part-load applications.

The final exhaust aftertreatment selection by the Ohio State team combines the technologies previously discussed. The basis of the system includes a DPF-LNT catalyst arrangement as illustrated in Figure 2.11. A novel flame reformer (developed at CAR [19]) inputs a fuel-air mixture at the inlet of the catalyst arrangement during regeneration cycles. The flame reformer operates in a rich fashion to provide the hydrocarbons, hydrogen, and carbon monoxide required for regeneration of the LNT and functions in a lean manner to simply provide the heat needed for DPF regeneration. A great advantage of this arrangement is its complete independence from the engine: the system does not rely on any conventional and inefficient engine-based regeneration.
methods (e.g., post-injection or engine throttling) and thus requires no engine control unit calibration. Experimental results indicate approximately 2.0 grams of Diesel fuel to regenerate half of a moderately sized catalyst [19]. Simulations of a mid-size sport-utility vehicle with a medium-sized Diesel engine suggest around 4 regeneration cycles per FUDS cycle that only result in a fuel economy penalty of approximately 1% [20] (compared to upwards of 5% using engine-based regeneration methods). For maximum regeneration efficiency, the system incorporates a bypass valve that directs the exhaust stream to one side of the catalyst arrangement during regeneration of the opposite side. A small (~ 1.25 L) oxidation catalyst upstream of the DPF-LNT arrangement oxidizes excess engine-out hydrocarbons, while a similar-sized oxidation catalyst at the conclusion of the system alleviates hydrocarbon, carbon monoxide, or hydrogen slip from regeneration of the lean NO\textsubscript{x} trap.

![Diagram of Exhaust Aftertreatment System](image)

**Figure 2.11: Exhaust Aftertreatment System**

In addition to the exhaust gas recirculation strategy integrated into the Fiat’s engine controller, the Ohio State team will further implement HCCI combustion as
another in-cylinder technique to reduce engine-out NO\textsubscript{x} emissions. The team will most likely use a proprietary fuel atomizer developed at the Ohio State Center for Automotive Research and limit HCCI to part-load operation at 50 ft \cdot lbs or less.

2.11 Control System Selection

A control system, as defined by the Ohio State team, is an assembly of hardware and software that is coordinated to execute a function by collecting and processing data from a set of components in order to control them. Therefore, the control hardware and software is identified as any active device that collects and processes data and responds with an action that ultimately affects the dynamics of the system. The Ohio State team defines the primary responsibility of the vehicle controller as to reliably command the powertrain components in real-time as specified by a control strategy and dependent upon the driver input and current state of the vehicle system. The Ohio State team also defines several secondary objectives of vehicle control: to monitor the state of the components to ensure safety, to control the cooling systems, to act as an equivalent onboard diagnostic unit for the hybrid-electric system, and to control the exhaust aftertreatment technologies. In general, the secondary level of control passively reacts to the states of the drivetrain components and only interferes (acts as an emergency shut-off) when a component deems to be operating in an unsafe manner.

Consistent to the control hierarchy previously described, these levels of control are executed on two physically separate embedded controllers. One processor is referred to as the primary controller and runs a primary (supervisory) control strategy, while
the other, designated as the secondary controller, manages secondary control responsibilities. The Ohio State team chose a dual-processor control system as a result of several benefits. First, the controllers can reside near those components from which they gather signals so as to reduce wiring and the possibility of electrical noise and interference. Further, the system provides flexibility by allowing parallel development of different sections of the control system. Finally, the redundancy of the proposed control architecture adds safety by letting the controllers to monitor one another and verify correct operation. The selected controllers are Motorola MPC555-based microcontrollers mounted on Phytec development boards. Each Phytec development board contains numerous inputs and outputs, such as analog-to-digital converters, digital-to-analog converters, pulse width modulation, and digital inputs and outputs. The controllers are connected by two separate network buses:

1. Controller area network (CAN) A (GM’s land area network), which connects the two electric machine controllers to the two MPC555 boards

2. CAN B, which connects the data acquisition hardware and a “gateway box” (bridge to the low-speed GM Class II J1850 network) to the two MPC555 boards

The overall control system architecture is detailed in Figure A.2 with the legend in Figure A.3.

2.12 Validation Approach

The Ohio State team devised a plan to develop and validate its vehicle architecture, vehicle control system, and control strategy through a multi-step, progressive, and reversible process as summarized in Figure 2.12. The initial stage utilizes a
software-in-the-loop (SIL) approach to develop a high-level supervisory control strategy. SIL is defined as a simulation technique in which a software-based model (plant) of an actual component is controlled by a piece of software emulating the actual hardware controller with no requirement to function in real-time. Hardware-in-the-loop (HIL), or the practice of interfacing the actual controller hardware with a software plant model operating in real-time, validates compatibility between control software and hardware. The Ohio State team then defines rapid prototyping as the exercise of replacing software plant component models from HIL with the actual component itself to validate overall compatibility between a control strategy, control hardware, and component hardware.

2.13 Summary

As an initial task of Challenge X, the first year, the Ohio State team engaged in an involved process to select a vehicle architecture for design, construction, and optimization in an attempt to eventually realize its vehicle technical specifications. The procedure included a literature search, a fuel selection, vehicle modeling and simulation, a weight analysis, and an extensive decision matrix to conceive the final hybrid-electric configuration.

The selected vehicle architecture is a charge-sustaining, through-the-road, power-split, bioDiesel-electric hybrid design. A downsized Diesel engine and integrated starter/alternator electric machine drives the front axle through an automatic transaxle. A larger electric machine for more-powerful traction purposes resides in the rear of the vehicle and mechanically couples to the road through a single-speed gearbox. A switchbox routes electrical power flow to the inverters from a high-voltage electrical
battery pack which further powers the conventional 12V system through a DC-DC converter. Emissions reductions occur via a novel system consisting of a lean NOx trap, Diesel particulate filter, and flame reformer. Vehicle systems connect and synergize through a control system consisting of a primary and secondary microprocessor controller networked by CAN buses. Validation of the vehicle architecture and control strategy occurs through software-in-the-loop, hardware-in-the-loop, and rapid prototyping techniques, of which the former and latter are demonstrated through works of this thesis as described in subsequent chapters.
CHAPTER 3

QUASI-STATIC MODEL

The initial step in software-in-the-loop testing is to create a vehicle model to act as a basis for the formation of a computer simulator. Technically, a model solely consists of a set of equations which describe the motion or behavior of a physical system; for clarity, this chapter occasionally illustrates the models through block diagrams in addition to equation formulation. The objective of the vehicle model as described in this chapter is to aid in validation of a control strategy with respect to energy management. Previous vehicle modeling experience indicates that a quasi-static model appropriately provides a basis for this energy management analysis. The model is termed "quasi-static" as dynamics exclusively occur at the vehicle level with a relatively low bandwidth (low frequency response). Appropriately then, the model lumps all drivetrain inertias into a single effective vehicle mass, neglects any driveline stiffness or damping, and assumes an infinitely fast response of the powertrain actuators (Diesel engine, Isa, and EM). At the highest level, the vehicle model is represented by Figure 3.1 and contains three primary divisions: the driver, the hybrid-electric powertrain, and the vehicle. As a general summary, the model feedbacks vehicle velocity to the driver who compares it with a desired velocity and responds with
an accelerator or brake pedal position that then actuates the powertrain and consequently moves the vehicle. This particular sequence is representative of a forward model (with respect to causality). Oppositely, a backward model initially considers a desired velocity profile and subsequently calculates the road load which is then completely satisfied through a combination of powertrain actuator torque outputs as dictated by the control strategy. Thus, a backward model exactly follows the desired driving cycle while the forward model attempts to as closely as possible. Forward modeling more precisely represents real world driving and thus is the appropriate choice in conjunction with development of a control strategy.

![Quasi-Static Model: High-Level Structure](image)

**Figure 3.1: Quasi-Static Model: High-Level Structure**

### 3.1 Driver

The developed model of the driver observes the difference in desired and current vehicle velocity and commands pedal positions according to a proportional-integral-derivative (PID) controller:

\[
\alpha = P_{dr} \cdot (v_{des} - v_{veh}) + I_{dr} \int_0^{t_e} (v_{des} - v_{veh}) \, dt + D_{dr} \frac{d(v_{des} - v_{veh})}{dt} \tag{3.1}
\]

\[
\beta = P_{dr} \cdot (v_{veh} - v_{des}) + I_{dr} \int_0^{t_e} (v_{veh} - v_{des}) \, dt + D_{dr} \frac{d(v_{veh} - v_{des})}{dt} \tag{3.2}
\]
To avoid undesired error accumulation over lengthy driving cycles, the integrator resets to zero when the vehicle rests with a zero velocity. Moreover, the accelerator and brake pedal positions saturate such that they are normalized values (i.e., $\alpha \in [0, 1]$ and $\beta \in [0, 1]$). Correct tuning of the $P_{dr}$, $I_{dr}$, and $D_{dr}$ parameters usually keeps $\alpha$ and $\beta$ between zero and one (thus avoiding saturation) while also resulting in an acceptable and smooth trace of the desired driving cycle. Due to the saturation and the nature of Equations (3.1) and (3.2), the accelerator and brake pedal positions are orthogonal (i.e., $\alpha \perp \beta$) such that depression of both pedals cannot occur simultaneously in the model.

### 3.2 Hybrid-Electric Powertrain

The hybrid-electric vehicle architecture as previously decided upon contains several powertrain components, each of which are statically modeled in this section. Most simply, the powertrain and vehicle is schematically represented in Figure 3.2 as a first-order, nonlinear model. This illustration conveys the absolute minimum inputs and states (along with several parameters) needed to express the shown system with a set of equations. However, for a level of precision eventually utilized in a vehicle simulator, this section will proceed through a progression such as that in Figure 3.3 to clearly model each powertrain component.

The pedal positions $\alpha$ and $\beta$ from the driver enter the hybrid-electric powertrain through the vehicle supervisory controller. This physical microprocessor board contains the code and logic (i.e., a control strategy) that considers the driver input and various states of the vehicle to determine five primary requests: the state of the engine (on/off), the ICE torque, the ISA torque, the EM torque, and the transaxle gear.
Figure 3.2: Quasi-Static Model: Drivetrain Schematic
Figure 3.3: Quasi-Static Model: Powertrain Component Level Structure
The high-level, supervisory control strategy itself is detailed in Chapter 5 while the remainder of this section will commence with the torque requests to the powertrain actuators. Again, it should be noted that the nature of the forward model induces the flow of torque (positive or negative) from the powertrain actuators to the wheel, while the velocity is fed back through the drivetrain after originating at the vehicle. This pattern will be noticed in those model equations as described throughout the rest of this chapter.

3.2.1 Diesel Engine

The quasi-static model approach assumes an infinitely quick and completely capable (within the torque limits) response of the powertrain actuators. Therefore,

$$T_{\text{ice}} = T_{\text{ice,req}}$$ (3.3)

If a control strategy ICE torque request exceeds that of the torque limit at a certain speed, the Diesel engine torque saturates at that limit such that:

$$T_{\text{ice}} \in \left[0, T_{\text{ice,max}}\right]$$ (3.4)

The engine output torque and engine speed (fed back from the torque converter model) determine the mass flow rate of fuel to the engine as interpolated or extrapolated from a steady-state map:

$$\dot{m}_f = \gamma_{\text{ice}} \cdot f(\omega_{\text{ice}}, T_{\text{ice}})$$ (3.5)

where $\gamma_{\text{ice}}$ symbolizes the state of the engine ($1 \equiv$ on, $0 \equiv$ off). Commonly, an engine with a zero torque request discontinues fuel injection until the engine decelerates to an idle velocity, and the Diesel engine map includes the appropriate data points to reflect this behavior.
The engine efficiency at a specific steady-state operating point is commonly useful for analysis purposes:

\[
\eta_{i.e} = \frac{\omega_{i.e} \cdot T_{i.e}}{\dot{m}_f \cdot Q_{tho, d}} \tag{3.6}
\]

### 3.2.2 Fuel Tank

The Diesel fuel mass flow rate as calculated in the engine model feedbacks to the fuel tank and is integrated with respect to time to determine the total fuel usage over an elapsed time period:

\[
m_f = \int_0^{t_e} \dot{m}_f \, dt \tag{3.7}
\]

The volume of Diesel fuel consumed is then simply:

\[
V_f = \frac{m_f}{\rho_d} \tag{3.8}
\]

For valid comparison with the fuel economy numbers associated with gasoline vehicles, the Diesel fuel usage can be equated to a gasoline-equivalent fuel volume by considering the different volumes of Diesel fuel and gasoline that contain the same quantity of potential energy release (according to their lower heating values):

\[
V_{f, ge} = V_f \cdot \left( \frac{\rho_d \cdot Q_{tho, d}}{\rho_g \cdot Q_{tho, g}} \right) \tag{3.9}
\]

### 3.2.3 Integrated Starter/Alternator and Tractive Electric Machine

Similar to that of the Diesel engine model, both electric machines respond infinitely fast to any torque request as a consequence of the static model approach:

\[
T_{isa} = T_{isa, rq} \tag{3.10}
\]

\[
T_{em} = T_{em, rq} \tag{3.11}
\]
Torque requests outside that of the torque limits at a given speed saturate at those torque limits:

\[
T_{\text{isa}} \in [T_{\text{isa,mn}}, T_{\text{isa,mx}}] \tag{3.12}
\]

\[
T_{\text{em}} \in [T_{\text{em,mn}}, T_{\text{em,mx}}] \tag{3.13}
\]

A steady-state map based upon speed and torque estimates the combined efficiency of the inverter and electric machine:

\[
\eta_{\text{isa}} = f(\omega_{\text{ice}}, T_{\text{isa}}) \tag{3.14}
\]

\[
\eta_{\text{em}} = f(\omega_{\text{em}}, T_{\text{em}}) \tag{3.15}
\]

Then, the electrical power to or from the inverters of the electric machines to produce or absorb torque, respectively, can be expressed as:

\[
P_{\text{isa,e}} = \omega_{\text{ice}} \cdot T_{\text{isa}} \cdot \left\{ \begin{array}{ll} 
1/\eta_{\text{isa}} & \text{if} \quad T_{\text{isa}} \geq 0 \\
\eta_{\text{isa}} & \text{if} \quad T_{\text{isa}} < 0 
\end{array} \right\} \tag{3.16}
\]

\[
P_{\text{em,e}} = \omega_{\text{em}} \cdot T_{\text{em}} \cdot \left\{ \begin{array}{ll} 
1/\eta_{\text{em}} & \text{if} \quad T_{\text{em}} \geq 0 \\
\eta_{\text{em}} & \text{if} \quad T_{\text{em}} < 0 
\end{array} \right\} \tag{3.17}
\]

Electric machines may contain a no-load and/or zero-speed magnetizing current in the leads whenever the device is “on” or initiated (analogous to idling in an engine). For simplicity, the model neglects this current and assumes its small magnitude to be negligible as compared to total current flux during normal operation.

### 3.2.4 High-Voltage Battery Pack and Switchbox

As mentioned previously, the switchbox design facilitates a direct connection of the ISA and EM on the same high-voltage bus. Therefore, the net power on this bus, plus the power delivered to the DC-DC converter to function the additional electrical...
accessories from hybridization, is considered to be the power delivered to or absorbed by the HV battery pack:

\[ P_{bt} = P_{tra,e} + P_{em,e} + P_{dc-dc,e} \]  \hspace{1cm} (3.18)

The high-voltage battery pack consists of numerous and identical battery modules that may be arranged in both a series and parallel fashion. The complete HV battery pack is modeled as a Thevenin equivalent circuit as depicted in Figure 3.4. It should be noted that the individual battery module parameter values tend highly to be functions of internal temperature; however, developed model neglects thermal dynamics and assumes a constant room temperature. Moreover, the battery module parameters also depend on the module state-of-charge (SOC). Conceptually (equations developed later in this subsection), the SOC of a battery indicates the percentage of charge in that battery relative to a fully-charged condition. As the state-of-charge cannot be assumed constant for a hybrid-electric vehicle model, the dependency exists as:

\[ V_{oc} = V_{oc}(s_{bt}) \]  \hspace{1cm} (3.19)

\[ R_{dis} = R_{dis}(s_{bt}) \]  \hspace{1cm} (3.20)

\[ R_{ch} = R_{ch}(s_{bt}) \]  \hspace{1cm} (3.21)

The internal resistance accounts for the HV battery pack inefficiency and alters the voltage experienced by the load:

\[ V_{cc} = n_{bt,s} \cdot V_{oc} - i_{bt} \cdot \frac{n_{bt,s}}{n_{bt,p}} \cdot \left\{ \begin{array}{ll} R_{dis} & \text{if } P_{bt} \geq 0 \\ R_{ch} & \text{if } P_{bt} < 0 \end{array} \right. \]  \hspace{1cm} (3.22)

The power into or out of the HV battery pack can simply be expressed as the product of the closed-circuit voltage and current:

\[ P_{bt} = i_{bt} \cdot V_{cc} \]  \hspace{1cm} (3.23)
By substitution of Equation (3.22) into Equation (3.23), the HV battery pack power equation further expands into a second-order polynomial with respect to electrical current:

\[
P_{bt} = i_{bt} \cdot n_{bt,s} \cdot V_{oc} - i_{bt}^2 \cdot \frac{n_{bt,s} \cdot R_{dis}}{n_{bt,p} \cdot R_{ch}} \cdot \begin{cases} R_{dis} & \text{if } P_{bt} \geq 0 \\ R_{ch} & \text{if } P_{bt} < 0 \end{cases}
\]  
(3.24)

Solving the quadratic of Equation (3.24) (and keeping the solution that equates to a zero current at zero electrical power) yields equations for current in the desired form as a function of power:

\[
i_{bt} = \frac{V_{oc} - \sqrt{V_{oc}^2 - 4R_{dis} \cdot P_{bt} / (n_{bt,s} \cdot n_{bt,p})}}{2R_{dis} / n_{bt,p}} \quad \text{if } P_{bt} \geq 0
\]  
(3.25)

\[
i_{bt} = \frac{V_{oc} - \sqrt{V_{oc}^2 - 4R_{ch} \cdot P_{bt} / (n_{bt,s} \cdot n_{bt,p})}}{2R_{ch} / n_{bt,p}} \quad \text{if } P_{bt} < 0
\]  
(3.26)

The electrical power into or out of the HV battery pack in Equation 3.24 has limitations based upon the electrical current capability of the battery module. For this model, the maximum and minimum currents are assumed known (from experimental
observation) and constant across the state-of-charge range of interest. Inspection of the quadratic expression in Equation 3.24 concludes that the maximum power may not necessarily occur at the maximum current. Instead, the maximum power correlates to a zero value of the derivative of Equation 3.24 with respect to current. Thus:

\[
i_{bt,p,mx} = \frac{V_{oc}}{2 \cdot R_{dis} \cdot n_{bt,p}} \quad (3.27)
\]

\[
i_{bt,p,nn} = \frac{V_{oc}}{2 \cdot R_{ch} \cdot n_{bt,p}} \quad (3.28)
\]

Brief and rough plugging in of typical parameter values concludes that Equations 3.27 and 3.28 equate to electrical currents larger in magnitude than that assumed for the maximum and minimum. Therefore, the peak of the quadratic power curve as a function of electrical current occurs past the maximum currents assumed, and so the maximum (in magnitude) HV battery pack power does indeed occur at the assumed maximum and minimum currents:

\[
P_{bt,mx} = i_{bt,mx} \cdot n_{bt,s} \cdot V_{oc} - i_{bt,mx}^2 \cdot \frac{n_{bt,s}}{n_{bt,p}} \cdot R_{dis} \quad (3.29)
\]

\[
P_{bt,nn} = i_{bt,nn} \cdot n_{bt,s} \cdot V_{oc} - i_{bt,nn}^2 \cdot \frac{n_{bt,s}}{n_{bt,p}} \cdot R_{ch} \quad (3.30)
\]

The efficiency of the high-voltage battery pack considers the power loss through the internal resistance:

\[
\eta_{bt} = \begin{cases} 
\frac{|P_{bt}|}{|P_{bt}| + i_{bt}^2 \cdot R_{dis} \cdot n_{bt,s}/n_{bt,p}} & \text{if } P_{bt} \geq 0 \\
\frac{|P_{bt}|}{|P_{bt}| + i_{bt}^2 \cdot R_{ch} \cdot n_{bt,s}/n_{bt,p}} & \text{if } P_{bt} < 0 
\end{cases} \quad (3.31)
\]

The electrical current over the duration of the driving cycle is integrated and divided by the nominal module capacity (multiplied by the number of module strings
in parallel) to calculate the change in state-of-charge:

$$\Delta s_{bt} = \frac{1}{n_{bt,g} \cdot E_{bt}} \int_{0}^{t_e} i_{bt} \, dt$$  \hspace{1cm} (3.33)

Then, an estimation of the SOC of the HV battery pack after a certain elapsed time requires knowing the state-of-charge value at the beginning of the driving cycle (assumed to be known for modeling purposes):

$$s_{bt} = s_{bt,0} - \Delta s_{bt}$$  \hspace{1cm} (3.34)

The original SOC is commonly assumed (again, for modeling purposes) to be a nominal value equidistant from the predetermined upper and lower state-of-charge limits, such that:

$$s_{bt,0} = s_{bt,nom}$$  \hspace{1cm} (3.35)

$$s_{bt,nom} = \frac{\delta_{bt,u} + \delta_{bt,l}}{2}$$  \hspace{1cm} (3.36)

The change in high-voltage battery pack state-of-charge over a driving cycle equates (theoretically) to a gasoline-equivalent fuel consumption or conservation. This deduction stems from the observation that all vehicle energy expenditure originates solely from the fuel tank (for a charge-sustaining hybrid): any net current drawn from the HV battery pack over a driving cycle requires replenishing in the future and vice versa. Therefore, Equation 3.9 can be expanded as:

$$V_{f,ge} = V_f \cdot \left( \frac{\rho_d \cdot Q_{thr,d}}{\rho_g \cdot Q_{thr,g}} \right) + \frac{n_{bt,s} \cdot V_{oc}(s_{bt,nom})}{\rho_g \cdot Q_{thr,g}} \int_{0}^{t_e} i_{bt} \, dt \cdot \begin{cases} 1/(\eta_{bt} \cdot \eta_{isa} \cdot \eta_{ice}) & \text{if } \Delta s_{bt} \geq 0 \\ (\eta_{bt} \cdot \eta_{isa})/\eta_{ice} & \text{if } \Delta s_{bt} < 0 \end{cases}$$  \hspace{1cm} (3.37)

It should be noted that Equation 3.37 refers to average efficiency values of the engine, ISA, and HV battery pack. Further, it arranges these efficiencies in a series
fashion that represents the total efficiency of this most direct energy route from the fuel tank to the high-voltage battery pack. As such, utilization of Equation 3.37 requires the acceptance of two assumptions: the assumed average efficiency values of the hybrid-electric system components and the assumed energy path through the integrated starter/alternator. Therefore, Equation 3.37 is not meant for (nor can provide) extreme accuracy but instead provides a sufficient and consistent method to use gasoline-equivalent fuel consumption as a basis for comparative purposes.

### 3.2.5 Torque Converter

As the static model neglects the rotational inertia of both the Diesel engine crankshaft and ISA rotor, the combined torque from both those powertrain actuators is completely experienced at the torque converter pump:

$$T_{icp} = T_{ice} + T_{isa}$$  \hspace{0.2cm} (3.38)

Developed by [18], the static model of a torque converter uses a series of characterizing coefficients determined by experimental data fitting. The physical construction of the common torque converter induces operation that can be modeled as residing in one of several distinct modes:

- **Positive Torus Flow**: $\omega_{ice} \geq \omega_{ict}$. The pump (engine side) velocity is greater than that of the turbine (transaxle side) and torque flows in a “positive” direction through the torque converter. Common during normal driving with positive power requests.
• **Negative Torus Flow.** \( \omega_{i,c} < \omega_{t,c,t} \). The turbine velocity exceeds the pump velocity and the torque flows in a “negative” direction through the torque converter. Common during a deceleration situation when the power request is negative.

• **Lock-Up.** \( \omega_{i,c} = \omega_{t,c,t} \). An internal clutch mechanically connects the pump to the turbine so that the pump and turbine angular velocities are equal. Common for higher velocities and eliminates fluid dynamic inefficiencies.

Operation in the positive torus flow mode can be further specified with two sub-modes. At low turbine-to-pump speed ratios, the torque converter operates in a **torque multiplication** mode in which the fluid flow impact on the stator results in a turbine torque greater than that of the pump torque. When the speed ratio exceeds that of the critical speed ratio threshold value, the torque converter functions in a **torque coupling** mode in which the stator has a zero net change in angular momentum and the pump and turbine torques are considered equal.

**Torque Multiplication** \( (\tau_{t,c} \leq \tau_{t,c}) \) (for Positive Torus Flow):

\[
T_{t,c,t} = C_1 \cdot \omega_{i,c}^2 + C_2 \cdot \omega_{i,c} \cdot \omega_{t,c,t} + C_3 \cdot \omega_{t,c,t}^2 \tag{3.39}
\]

\[
T_{t,c,p} = C_4 \cdot \omega_{i,c}^2 + C_5 \cdot \omega_{i,c} \cdot \omega_{t,c,t} + C_6 \cdot \omega_{t,c,t}^2 \tag{3.40}
\]

**Torque Coupling** \( (\tau_{t,c} > \tau_{t,c}) \) (for Positive Torus Flow):

\[
T_{t,c,t} = C_7 \cdot \omega_{i,c}^2 + C_8 \cdot \omega_{i,c} \cdot \omega_{t,c,t} + C_9 \cdot \omega_{t,c,t}^2 \tag{3.41}
\]

\[
T_{t,c,p} = T_{t,c,t} \tag{3.42}
\]
Recognizing the speed ratio as $\tau_{tc} = \omega_{tc,t}/\omega_{tc,e}$, Equations 3.39 - 3.42 transform to:

**Torque Multiplication ($\tau_{tc} \leq \hat{\tau}_{tc}$) (for Positive Torus Flow):**

$$T_{tc,t} = \left(C_1 + C_2 \cdot \tau_{tc} + C_3 \cdot \tau_{tc}^2\right)\omega_{tc,e}^2$$ \(\text{(3.43)}\)

$$T_{tc,p} = \left(C_4 + C_5 \cdot \tau_{tc} + C_6 \cdot \tau_{tc}^2\right)\omega_{tc,e}^2$$ \(\text{(3.44)}\)

**Torque Coupling ($\tau_{tc} > \hat{\tau}_{tc}$) (for Positive Torus Flow):**

$$T_{tc,t} = \left(C_7 + C_8 \cdot \tau_{tc} + C_9 \cdot \tau_{tc}^2\right)\omega_{tc,e}^2$$ \(\text{(3.45)}\)

$$T_{tc,p} = T_{tc,t}$$ \(\text{(3.46)}\)

The numerical value of the critical speed threshold ratio can be calculated by observing that the pump torque equals the turbine torque in the multiplication mode at the threshold speed ratio:

$$C_1 + C_2 \cdot \hat{\tau}_{tc} + C_3 \cdot \hat{\tau}_{tc}^2 = C_4 + C_5 \cdot \hat{\tau}_{tc} + C_6 \cdot \hat{\tau}_{tc}^2$$ \(\text{(3.47)}\)

Solving the quadratic in Equation 3.47 yields:

$$\hat{\tau}_{tc} = \frac{(C_2 - C_5) \pm \sqrt{(C_5 - C_2)^2 - 4(C_6 - C_3)(C_4 - C_1)}}{2(C_6 - C_3)}$$ \(\text{(3.48)}\)

Notice that, due to the particular nature of torque converter operation as discussed previously, $T_{tc,t} > T_{tc,p}$ at a stall condition ($\omega_{tc,t} = 0$) which implies that $C_1 > C_4$ (from inspection of Equations 3.39 and 3.40). If $C_4$ equals $C_1$, the turbine torque equals the pump torque in a stall condition and thus the torque converter instantly enters the torque coupling mode (and, consequently, $\hat{\tau}_{tc} = 0$). Equation 3.48 mimics this logic ($\hat{\tau}_{tc} = 0$ when $C_4 = C_1$) when the positive root is assumed and thus:

$$\hat{\tau}_{tc} = \frac{(C_2 - C_5) + \sqrt{(C_5 - C_2)^2 - 4(C_6 - C_3)(C_4 - C_1)}}{2(C_6 - C_3)}$$ \(\text{(3.49)}\)
The presentation and development of the previous equations fully model the torque converter in the positive torus mode; however, the expressions are not in a format conducive for the static model and further algebraic manipulations are required. In particular, the equation for the threshold between the torque multiplication and torque coupling modes must not be in terms of a velocity ratio as the pump (engine) speed is not known and instead must be solved for. Further, the pump torque directly results from the ICE and ISA torques, and thus the equations must treat the pump torque as the independent variable. The logic in the following paragraph is used to convert the positive torus flow torque converter equations into the appropriate form.

In the torque coupling mode for positive torus flow, the pump speed and turbine speed are approximately equal (but not exactly as the torque converter cannot operate 100% efficiently). If turbine speed is held constant, an increasing pump torque (which equals the turbine torque) would necessitate an increased pump speed. Therefore, for the torque coupling mode, it stands to reason that there exists a maximum pump torque limit before the pump speed deviates too far from that of the turbine speed to continually remain in that torque coupling mode. Logically then, this maximum threshold pump torque ($\tilde{T}_{tc,p}$) occurs at the critical speed ratio threshold $\tilde{\tau}_{tc}$ and is expressed as a derivation of Equations 3.45 and 3.46:

$$\tilde{T}_{tc,p} = \left( \frac{C_7}{\tilde{\tau}_{tc}^2} + \frac{C_8}{\tilde{\tau}_{tc}} + C_9 \right) \omega_{tc,t}^2 \quad (3.50)$$

For the torque multiplication mode, Equation 3.40 solved for pump (engine) velocity and Equation 3.39 satisfy a complete model while presenting the equations in terms of the correct independent variables for the static modeling approach. Similarly, Equation 3.41 solved for pump velocity and Equation 3.42 completely model the torque converter in the torque coupling mode. The choice of a positive or negative
root for the developed equations correlates to the solution which brings the pump speed closer to the turbine speed as the pump torque decreases. Summarizing:

**Torque Multiplication** \( (T_{tc,p} \geq \dot{T}_{tc,p}) \) **(for Positive Torus Flow):**

\[
\omega_{ice} = \frac{-C_5 \cdot \omega_{tc,t} + \sqrt{(C_5 \cdot \omega_{tc,t})^2 - 4C_4(C_6 \cdot \omega_{tc,t}^2 - T_{tc,p})}}{2C_4} \quad (3.51)
\]

\[
T_{tc,t} = C_1 \cdot \omega_{ice}^2 + C_2 \cdot \omega_{ice} \cdot \omega_{tc,t} + C_3 \cdot \omega_{tc,t}^2 \quad (3.52)
\]

**Torque Coupling** \( (T_{tc,p} < \dot{T}_{tc,p}) \) **(for Positive Torus Flow):**

\[
\omega_{ice} = \frac{-C_5 \cdot \omega_{tc,t} + \sqrt{(C_5 \cdot \omega_{tc,t})^2 - 4C_4(C_6 \cdot \omega_{tc,t}^2 - T_{tc,p})}}{2C_4} \quad (3.53)
\]

\[
T_{tc,t} = T_{tc,p} \quad (3.54)
\]

As mentioned previously, a negative torus flow occurs when the velocity of the turbine exceeds that of the pump (engine side) and torque flows in a negative direction through the torque converter. In this case, the stator freely rotates in the direction of the pump and prevents any torque multiplication; therefore, the torque converter constantly resides in a torque coupling mode:

**Negative Torus Flow** \( (\omega_{ice} < \omega_{tc,t}) \):

\[
T_{tc,p} = -\left( C_6 \cdot \omega_{ice}^2 + C_8 \cdot \omega_{ice} \cdot \omega_{tc,t} + C_7 \cdot \omega_{tc,t}^2 \right) \quad (3.55)
\]

\[
T_{tc,t} = T_{tc,p} \quad (3.56)
\]

Again, however, Equations 3.55 and 3.56 exist in a form better suited for a dynamic model. For static modeling purposes, it is desired for the turbine velocity and pump torque to act as inputs and the pump velocity and turbine torque to be outputs. It is worth noting that a negative torus flow solely occurs when the pump torque is negative or zero. For the quasi-static model, this exactly correlates to a negative
or zero sum of torques from the ICE and ISA since the inertias of these powertrain actuators are lumped at the vehicle level. Algebraic manipulation similar to that for the positive torus flow case results in the desired form of expression; in other words, solving the quadratic of Equation 3.55 and keeping the solution for which pump speed decreases with an increasing magnitude of negative pump torque (and a constant turbine speed) yields:

**Negative Torus Flow** ($T_{tc,p} \leq 0$):

$$\omega_{tc} = \frac{-C_8 \cdot \omega_{tc,t} - \sqrt{(C_8 \cdot \omega_{tc,t})^2 - 4C_9(C_7 \cdot \omega_{tc,t}^2 + T_{tc,p})}}{2C_9}$$  \hspace{1cm} (3.57)

$$T_{tc,t} = T_{tc,p}$$  \hspace{1cm} (3.58)

A torque converter lock-up commanded (designated as $\gamma_{tc}$) originates from a control strategy and is typically initiated from the torque coupling mode when pump and turbine speeds are approximately equal. As previously described, the physical mechanism consists of an internal clutch that mechanically connects the pump to the turbine and consequently eliminates fluid coupling inefficiencies:

**Lock-Up** (if $\gamma_{tc} = 1$):

$$\omega_{tc} = \omega_{tc,t}$$  \hspace{1cm} (3.59)

$$T_{tc,t} = T_{tc,p}$$  \hspace{1cm} (3.60)

In any mode, the efficiency of the torque converter can simply be calculated:

$$\eta_{tc} = \begin{cases} 
(\omega_{tc,t} \cdot T_{tc,t})/(\omega_{tc} \cdot T_{tc,p}) & \text{if } T_{tc,p} \geq 0 \\
(\omega_{tc} \cdot T_{tc,p})/(\omega_{tc,t} \cdot T_{tc,t}) & \text{if } T_{tc,p} < 0 \end{cases}$$  \hspace{1cm} (3.61)

### 3.2.6 Automatic Transaxle and Single-Speed Gearbox

Both the automatic transaxle in the front of the vehicle and the single-speed gearbox connected to the EM in the rear are modeled as completely rigid and as
responding infinitely quick (e.g., no gear tooth backlash, no gear changing dynamics, etc.). Power loss occurs through an assumed constant efficiency such that:

\[
T_{tr} = \eta_{tr} \cdot T_{tc,t} \cdot \begin{cases} \eta_{tr} & \text{if } T_{tc,t} \geq 0 \\ 1/\eta_{tr} & \text{if } T_{tc,t} < 0 \end{cases} \tag{3.62}
\]

\[
T_{gb} = \tau_{gb} \cdot T_{em} \cdot \begin{cases} \eta_{gb} & \text{if } T_{em} \geq 0 \\ 1/\eta_{gb} & \text{if } T_{em} < 0 \end{cases} \tag{3.63}
\]

where the automatic transaxle ratio is dependent on its present gear as requested by a control strategy \( \tau_{tr} = \tau_{tr}(\gamma_{tr}) \).

The gearing ratios represent a ratio of input (actuator side) speed to output (road side) speed. The neutral gear of the automatic transaxle assumes dependency on the state of the engine; in other words, when and only when the engine is off (zero velocity), the automatic transaxle assumes a neutral gearing in which the torque converter turbine speed is then consequently zero:

\[
\omega_{tc,t} = \gamma_{tc} \cdot \tau_{tr} \cdot \omega_{tr} \tag{3.64}
\]

\[
\omega_{em} = \tau_{gb} \cdot \omega_{gb} \tag{3.65}
\]

### 3.2.7 Brakes

During a braking (negative power request) situation, the maximum negative torque that can theoretically be realized at the wheels depends on the coefficient of friction between the tire and the ground:

\[
T_{brk,mz} = m_{veh} \cdot g \cdot \cos(\phi) \cdot r_{wh} \cdot \mu \tag{3.66}
\]

The front and rear brakes then apply a negative torque to the driveline as directly proportional to that maximum braking torque and the brake pedal position:

\[
T_{brk,f} = \beta \cdot T_{brk,mz} \cdot \zeta_{brk,f} \tag{3.67}
\]

\[
T_{brk,r} = \beta \cdot T_{brk,mz} \cdot \zeta_{brk,r} \tag{3.68}
\]
This braking torque subtracts from the transaxle and gearbox output torques to result in the torque at the front or rear wheels, respectively:

\[ T_{wh,f} = T_{tr} - T_{brk,f} \]  
\[ T_{wh,r} = T_{gb} - T_{brk,r} \]  

(3.69)  
(3.70)

The assumption of completely rigid brake components implies that the angular velocity does not vary through the brake model:

\[ \omega_{tr} = \omega_{wh,f} \]  
\[ \omega_{gb} = \omega_{wh,r} \]  

(3.71)  
(3.72)

### 3.2.8 Wheels

The tires and wheels are assumed to be completely rigid so that the total radius \( r_{wh} \) is the sum of the wheel radius and the tire sidewall height. Other tire dynamics (e.g., slip and any other inefficiencies) are neglected in the static model; therefore, the powertrain tractive force and linear velocity with respect to the road are related to the wheel by simple relations:

\[ F_{wh,f} = T_{wh,f}/r_{wh} \]  
\[ F_{wh,r} = T_{wh,r}/r_{wh} \]  
\[ \omega_{wh,f} = \upsilon_{veh}/r_{wh} \]  
\[ \omega_{wh,r} = \upsilon_{veh}/r_{wh} \]  

(3.73)  
(3.74)  
(3.75)  
(3.76)

The sum of the tractive wheel forces equates to the total tractive force experienced by the vehicle:

\[ F_{trac} = F_{wh,f} + F_{wh,r} \]  

(3.77)
3.3 Vehicle

Along with the tractive force delivered by the powertrain, the vehicle experiences a number of road loads that influences its dynamics. The most influential dissipative effects (aerodynamic drag and rolling resistance) and road properties (e.g., grade) model the total road load:

\[ F_{rd} = \frac{1}{2} \cdot \rho_{air} \cdot C_d \cdot A_f \cdot v_{veh}^2 + m_{veh} \cdot g \cdot C_r \cdot \cos(\phi) + m_{veh} \cdot g \cdot \sin(\phi) \] (3.78)

It should be noted that the rolling resistance is a parasitic loss that only works to decelerate the vehicle; therefore, that particular contribution to the road load equates to zero at a vehicle rest condition.

The static model as presented thus far lumps all mass into a single vehicle inertia and consequently neglects the effects of the rotating drivetrain components. A common solution to this issue employs a mass factor that, when multiplied with the vehicle curb mass, defines an effective vehicle mass:

\[ m_{veh,e} = \gamma_{veh} \cdot m_{veh} \] (3.79)

Recognizing the conservation of kinetic energy through gear reductions, rigidly attached rotational inertias can be reflected to the vehicle level as proportional to the square of the speed ratio. For instance, the reflected inertia of the EM at the vehicle equates to:

\[ m_{em,r} = \frac{J_{em} \cdot \tau_g^2}{\tau_{wh}^2} \] (3.80)

The sum of these reflected inertias for all drivetrain components \( \phi \) then allows calculation of the mass factor:

\[ \gamma_{veh} = \frac{m_{veh} + \sum m_{\phi,r}}{m_{veh}} \] (3.81)
It should be noted that Equation 3.81 should further take into account situations at which drivetrain components are inactive (e.g., a zero reflected inertia of the ICE and ISA when the engine is off). The reflected inertia of all drivetrain rotational inertias upstream of the transaxle are a function of the varying transmission gear. Moreover, with a torque converter, the angular acceleration of the pump side may be nonlinearly related to the angular acceleration of the turbine side, and thus application of the appropriate equations becomes somewhat ambiguous for the engine-side inertias. Therefore, the model utilizes the simplicity and consistency of a common mass factor equation [14]:

\[
\gamma_{veh} = 1 + 0.04 \cdot \tau_{tr} + 0.0025 \cdot \tau_{tr}^2
\]  

(3.82)

As mentioned previously, the overall model is termed quasi-static as the sole dynamics occur at the vehicle level. Summing forces about the effective vehicle mass results in this single differential equation:

\[
m_{veh,e} \cdot \ddot{v}_{veh} = F_{trac} - F_{rd}
\]

(3.83)

It is observed that Equation 3.83 can easily be solved for the derivative of vehicle velocity with respect to time. Integration of this with respect to time calculates velocity which is the primary vehicle state fed back throughout the model as previously outlined. Integration again with respect to time then estimates traversed vehicle translation:

\[
v_{veh} = \int_0^{t_e} \dot{v}_{veh} \, dt
\]

(3.84)

\[
x_{veh} = \int_0^{t_e} v_{veh} \, dt
\]

(3.85)

Although not a state of the model and therefore not necessary for model completeness, the vehicle displacement facilitates the calculation of the commonly-analyzed metric
\[ \eta_{fe} = \frac{x_{veh}}{V_{f,ge}} \] (3.86)

### 3.4 Summary

Validation of a control strategy with respect to energy management allows utilization of a computer simulator based upon a quasi-static vehicle model. The model is termed quasi-static as all powertrain inertias are lumped at the vehicle level, all driveline components are assumed rigid, and all powertrain elements are assumed as responding infinitely fast to torque requests, transaxle gear changes, etc. The overall drivetrain system model is highly nonlinear but also first-order with respect to vehicle velocity. As such, the vehicle model itself responds with a relatively slow bandwidth and thus is computationally cheap when implemented into a computer vehicle simulator. The model is also discontinuous as a result of certain events (e.g., at a transmission gear shift or torque converter lock-up) and can therefore be referred to as a hybrid model that requires hybrid control such as that explained in Chapter 5.

The quasi-static model is forward (with respect to causality) and divides into three sections: (i) the driver, (ii) the hybrid-electric powertrain, and (iii) the vehicle. The driver compares the vehicle velocity with a desired velocity profile and responds with an accelerator or brake pedal position. These commands input into the vehicle controller and, along with several vehicle states, are considered by a control strategy such as that described in a subsequent chapter. Powertrain actuator torque requests are received at the component level and translate into torques which move through the drivelines. Tractive forces result at the wheels and accelerate the effective vehicle inertia against the road load to cause a change in velocity. The dynamic equation

68
formulations of Chapter 4 generally follows this same progression as does a computer vehicle simulator as described in Section 6.2.
CHAPTER 4

DYNAMIC MODEL

The static model and simulator of Chapter 3 unsuccessfully captures those dynamic effects of interest (and rightfully so) when considering vehicle drivability. Therefore, software-in-the-loop techniques to validate a control strategy with respect to vehicle drivability necessitates a dynamic model [42], [43]. The developed dynamic model and simulator in this chapter strives to capture that frequency range (\(\sim 1 - 10\) Hz) that the driver relates to vehicle drivability. At the highest level, the vehicle model is represented by Figure 4.1 and again contains three primary divisions: the driver, the hybrid-electric powertrain, and the vehicle. As with the static model, the dynamic model feedbacks vehicle velocity to the driver who responds with an accelerator or brake pedal position that then actuates the powertrain and consequently moves the vehicle. It is worth noting again that the chain of events and direction of causality are representative of a forward model.

4.1 Driver

The model of the driver replicates that previously developed in Section 3.1 for the static model: the driver compares the current vehicle velocity with the desired velocity at that moment and acts as a PID controller to command an accelerator
or brake pedal position. However, as explained later in this chapter, the dynamic model defines a front and rear vehicle velocity; both of these values will typically be extremely close to one another and the model chooses to use the frontal vehicle velocity such that:

\[
\alpha = P_{dr} \cdot (v_{des} - v_{veh,f}) + I_{dr} \int_0^{t_e} (v_{des} - v_{veh,f}) \, dt + D_{dr} \frac{d(v_{des} - v_{veh,f})}{dt} \tag{4.1}
\]

\[
\beta = P_{dr} \cdot (v_{veh,f} - v_{des}) + I_{dr} \int_0^{t_e} (v_{veh,f} - v_{des}) \, dt + D_{dr} \frac{d(v_{veh,f} - v_{des})}{dt} \tag{4.2}
\]

### 4.2 Hybrid-Electric Drivetrain

The schematic representation of Figure 4.2 (based upon that in [2]) shows the absolute minimum states and inputs required for a fully dynamic drivetrain model. However, for clarity and to follow a progression as in a dynamic simulator, this section proceeds to model each drivetrain component with the variables as shown in Figure 4.3.

The accelerator and brake pedal positions from the driver model exclusively compose the inputs to the vehicle supervisory controller. The controller then, through the logic of the control strategy as described in Chapter 5, commands the powertrain
Figure 4.2: Dynamic Model: Drivetrain Schematic
Figure 4.3: Dynamic Model: Powertrain Component Level Structure
actuators and selects an appropriate transaxle gear. The remainder of this section derives the dynamic equations of those components that compose the drivetrain itself.

4.2.1 Diesel Engine and Integrated Starter/Alternator

A dynamic model of a Diesel engine that appropriately considers the effects of exhaust gas recirculation, turbocharging, etc. has not yet been determined. A current idea is to experimentally map a series of trajectories with common engine transients and then create a “black box” model that estimates the dynamic response of the engine to a torque request:

\[ T_{\text{ice}} = f_{\text{blackbox}}(\omega_{\text{ice}}, T_{\text{ice},r_q}, \text{etc.}) \]  

(4.3)

For the ISA, a first-order response satisfactorily models its quick and smooth reaction to a torque request:

\[ \tau_{\text{isa}} \frac{dT_{\text{isa}}}{dt} + T_{\text{isa}} = T_{\text{isa},r_q} \]  

(4.4)

The ICE and ISA brake torques combine and work to accelerate the lumped inertia of the integrated starter/alternator and Diesel engine:

\[ (J_{\text{isa}} + J_{\text{ice}}) \cdot \dot{\omega}_{\text{ice}} = T_{\text{isa}} + T_{\text{ice}} - T_{\text{r}},p \]  

(4.5)

Fuel usage from the Diesel engine is approximated by the same steady-state map interpolation approach as in Equation 3.5, and, similarly, required electrical power to or from the ISA inverter mimics that approach conveyed in Equation 3.16.

4.2.2 Fuel Tank

Please refer to Subsection 3.2.2.
4.2.3 Torque Converter

Please refer to Subsection 3.2.5. It should be noted that these equations model the torque converter statically. A fully dynamic model would likely consist of a fluid dynamics analysis using physical parameters of the torque converter (instead of experimentally found characterizing parameters). Retaining the simplicity of the static equations while appropriately modeling the torque converter for a dynamic system may necessitate a few additions to the model (e.g., lumping of a pump inertia with the ICE and ISA inertias, damping of torque oscillations from the driven side, etc.).

4.2.4 Tractive Electric Machine

Similar to that for the ISA, the tractive electric machine is modeled as responding in a first-order fashion to a control strategy torque request:

\[ \tau_{em} \frac{dT_{em}}{dt} + T_{em} = T_{em,rq} \] (4.6)

The resultant torque works to accelerate the lumped inertia of the EM and its accompanying single-speed gearbox:

\[ J_{em} \cdot \dot{\omega}_{ice} = T_{em} - T_{em,net} \] (4.7)

The required electrical power to provide or absorb the EM torque is the same as that in Equation 3.17.

4.2.5 High-Voltage Battery Pack and Switchbox

Please refer to Subsection 3.2.4.
4.2.6 Automatic Transaxle and Single-Speed Gearbox

The transaxle dynamic model includes a (post gear reduction) inertia that accelerates according to the torques experienced on either side:

\[
J_{tr} \cdot \dot{\omega}_{tr} = \tau_{tr} \cdot T_{tc,t} \cdot \begin{cases} 
\eta_{tr} & \text{if } T_{tc,t} \geq 0 \\
1/\eta_{tr} & \text{if } T_{tc,t} < 0 
\end{cases} - T_{tr} \tag{4.8}
\]

The single-speed gearbox rotational inertia is lumped with the EM inertia; therefore, the net torque at the tractive electric machine rotor is directly related to the output torque of the gearbox:

\[
T_{em,net} = \frac{T_{gb}}{\tau_{gb}} \cdot \begin{cases} 
1/\eta_{gb} & \text{if } T_{gb} \geq 0 \\
\eta_{gb} & \text{if } T_{gb} < 0 
\end{cases} \tag{4.9}
\]

Expressions of rotational velocity relationships for the automatic transaxle and EM gearbox are identical to that presented in Equations 3.64 and 3.65.

4.2.7 Half Shafts

The modeled stiffness and damping of the half shafts transmit torque between the wheels and the remainder of the driveline:

\[
T_{hs,f} = k_{hs,f} \int_0^{t_e} (\omega_{tr} - \omega_{hs,f}) \, dt + b_{hs,f} \cdot (\omega_{tr} - \omega_{hs,f}) \tag{4.10}
\]

\[
T_{hs,r} = k_{hs,r} \int_0^{t_e} (\omega_{gb} - \omega_{hs,r}) \, dt + b_{hs,r} \cdot (\omega_{gb} - \omega_{hs,r}) \tag{4.11}
\]

These torques respectively equate to those at the output of the automatic transaxle and single-speed gearbox, such that:

\[
T_{tr} = T_{hs,f} \tag{4.12}
\]

\[
T_{gb} = T_{hs,r} \tag{4.13}
\]
4.2.8 Brakes

Please refer to Equations 3.67 and 3.68 for the frictional braking torque given a certain brake pedal position. The braking torque subtracts from the half shaft torque directly upstream to produce the torque at the wheel:

\[ T_{wh,f} = T_{hs,f} - T_{brk,f} \]  \hspace{1cm} (4.14)
\[ T_{wh,r} = T_{hs,r} - T_{brk,r} \]  \hspace{1cm} (4.15)

The assumption of a completely rigid brake system leads to:

\[ \omega_{hs,f} = \omega_{wh,f} \]  \hspace{1cm} (4.16)
\[ \omega_{hs,r} = \omega_{wh,r} \]  \hspace{1cm} (4.17)

4.2.9 Wheels

The dynamic model assumes a stiffness and damping of the tires that can be lumped into overall vehicle chassis qualities as described in the subsequent section. Therefore, for the purpose of modeling, the tires themselves behave as completely rigid:

\[ F_{wh,f} = T_{wh,f}/r_{wh} \]  \hspace{1cm} (4.18)
\[ F_{wh,r} = T_{wh,r}/r_{wh} \]  \hspace{1cm} (4.19)
\[ \omega_{wh,f} = v_{veh,f}/r_{wh} \]  \hspace{1cm} (4.20)
\[ \omega_{wh,r} = v_{veh,r}/r_{wh} \]  \hspace{1cm} (4.21)

4.3 Vehicle

In order to capture those dynamics that may occur from powertrain component actuation and consequently may concern the driver, the dynamic model assumes a
lumped (combination of tire, suspension, etc.) stiffness and damping at the vehicle level. For simplicity, the model also assumes the vehicle mass to be split as proportional to its physical mass distribution from the front to rear axle (where \( \zeta_{m,f} + \zeta_{m,r} = 1 \)). Summing forces about each inertia leads to:

\[
\zeta_{m,f} \cdot m_{veh} \cdot \ddot{v}_{veh,f} = F_{veh,f} - F_{rd} - F_{veh} \quad (4.22)
\]

\[
\zeta_{m,r} \cdot m_{veh} \cdot \ddot{v}_{veh,r} = F_{veh,r} + F_{veh} \quad (4.23)
\]

Please refer to Equation 3.78 for the vehicle road load algorithm (in this case, \( v_{veh} \equiv v_{veh,f} \)). The force transmitted through the lumped vehicle stiffness and damping equates to:

\[
F_{veh} = k_{veh} \int_0^t (v_{veh,f} - v_{veh,r}) \, dt + b_{veh} \cdot (v_{veh,f} - v_{veh,r}) \quad (4.24)
\]

### 4.4 Summary

Although not included in the work of this thesis, eventual software-in-the-loop validation of a control strategy with respect to drivability necessitates a dynamic computer vehicle simulator. The dynamic equations of this chapter form an initial basis for which a dynamic simulator can be based upon. It should be noted that, to ensure all relevant dynamics have been captured (e.g., transmission gear shifting dynamics [10], [46]), a more comprehensive study of the vehicle drivetrain should be undertaken as an appropriate future task.

The forward dynamic model separates into driver, hybrid-electric powertrain, and vehicle sections. The progression mimics that used in the quasi-static model formulation: the driver compares the current velocity with a desired velocity and commands accelerator and brake pedal positions which, through the controller, causes a response
of the powertrain and resultant drivetrain and vehicle dynamics. The overall vehicle system model is again highly nonlinear but also of a much larger order to appropriately capture the dynamics of interest. Including the first-order response of the electric machines to a torque request, the drivetrain model is 11th order with engine angular speed, transaxle angular speed and displacement, front and rear wheel angular speed and displacement, and EM angular speed and displacement as the states. The high order model promotes high frequency responses that are significantly more computationally expensive than that of the quasi-static model. Therefore, validation of a control strategy with respect to drivability through software-in-the-loop techniques should appropriately be a final (but very important) adaptation to the previously validated control strategy energy management logic.
CHAPTER 5

CONTROL STRATEGY

As mentioned previously, control strategy responsibilities are divided among two groups: those that are designated as primary and those that are allocated as secondary. This chapter heavily focuses on primary (supervisory) control strategy development as its eventual validation is the foremost reason for the vehicle models as formulated in Chapters 3 and 4. However, a few secondary considerations are also summarized at the conclusion of this chapter. It should further be noted that the supervisory control strategy as described in this chapter is very high level and, as such, generally commands the powertrain on what to do but not necessarily the specific manner in which it should be done. This high-level control strategy is appropriate for initial validation with respect to overall system energy management on a quasi-static based simulator. The final control strategy as actually implemented in the vehicle (or even used with a dynamic model-based simulator) must consider a more precise level of control on a lower, component level.
5.1 Primary Control Strategy Objectives

Analogous to the role of vehicle technical specifications in deciding a vehicle architecture, a clear vision of objectives is vital for development of the control strategy. The following summarizes the objectives of the supervisory control strategy:

- **Vehicle Mode Management.** The specified HEV architecture permits numerous vehicle modes. The primary control strategy is required to select the appropriate mode and transmission gear as well as managing the transition between vehicle modes.

- **Power Request Satisfaction.** A principal objective of the primary control strategy is to continuously meet the driver's local power request that is based upon accelerator and brake pedal positions and available power in the current vehicle state and mode.

- **Fuel Consumption Minimization.** The primary control strategy must aim to minimize fuel usage on a global basis.

- **Battery State-of-Charge Management.** For reasons mainly dealing with consumer acceptability, the proposed vehicle architecture is charge sustaining; as such, the primary control strategy assumes responsibility of maintaining the HV battery pack SOC between two predetermined boundaries.

- **Drivability Assurance.** The primary control algorithm is required to assure acceptable drivability with respect to oscillations and impulses of the drivetrain and vehicle.
5.2 Vehicle Mode Management

The choice of an automatic transaxle facilitates improved drivetrain control as the driver is removed from making (sometimes nonideal) selections regarding transmission gear. However, the driver still selects the high-level vehicle mode, and the four options for this mode selection are shown at the top of Figure 5.1.

![Diagram of vehicle modes and gear selection logic]

Figure 5.1: Transaxle State and Gear Selection Logic

Upon initial entrance into the vehicle, the driver encounters the vehicle in a Park mode. As in a conventional vehicle, the Park mode implies that the vehicle is static (with respect to translational motion) and exactly corresponds to the transmission residing in a park condition with engagement of the parking brake mechanism teeth. Driver initiation of the vehicle into a Reverse mode (by shifter, switch, or otherwise) transfers the transaxle into a reverse gearing. The four-quadrant capability of the EM facilitates a sole electric powering of the vehicle during modest reverse situations, and
the presence of the torque converter allows the transaxle to remain in a reverse gearing during these situations. As will be seen subsequently, a significant depression of the accelerator pedal then utilizes power from the Diesel engine, at which point the reverse gearing condition of the transaxle is fully appropriate. The vehicle in a Neutral mode (as commanded by the driver) places the automatic transaxle in its neutral state as it would exist in any conventional vehicle. Driver initiation of the vehicle into the Drive mode indicates a desire to move the vehicle in a positive direction. Unlike correlations as noticed with the other previously discussed modes, the Drive mode does not imply a drive gearing of the transaxle; instead, the transaxle initially exists more appropriately in a neutral state. The particular reason for this is unique to the next-highest level of vehicle modes as described subsequently.

The flexibility of vehicle modes (in the Drive state) is certainly an advantage of the chosen vehicle architecture and indirectly factored into the decision process. The initial task of the primary control strategy is to select the proper vehicle mode and transmission gear. Table 5.1 summarizes the six identified and possible vehicle modes once the vehicle is in a Drive state as commanded by the driver.

Figure 5.2 illustrates the high-level hybrid logic to determine the appropriate instances for a transition between vehicle modes. The HEV begins in the Vehicle Idle mode at a Drive state as initiated by the driver. In this mode, the three powertrain actuators rest inactively and all vehicle accessories (air conditioning compressor, etc.) operate electrically from the 12V grid as facilitated by the DC-DC converter. It should be noted that all further accessories conventionally coupled to and powered by an engine (e.g., power steering pump) are also driven electrically throughout all other vehicle modes as well. The Vehicle Idle mode remains pragmatic so long as
<table>
<thead>
<tr>
<th>Mode</th>
<th>ICE</th>
<th>ISA</th>
<th>EM</th>
<th>Transaxle</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vehicle Idle</td>
<td>Off</td>
<td>Off</td>
<td>Off</td>
<td>Neutral</td>
</tr>
<tr>
<td>All accessories run electrically at vehicle rest.</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Electric Only</td>
<td>Off</td>
<td>Off</td>
<td>Mot. or Gen.</td>
<td>Neutral</td>
</tr>
<tr>
<td>Vehicle launched from rest with EM.</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Engine Start</td>
<td>Off</td>
<td>Mot.</td>
<td>Mot. or Gen.</td>
<td>Neutral</td>
</tr>
<tr>
<td>Engine quickly started by ISA.</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Normal</td>
<td>On</td>
<td>Mot. or Gen.</td>
<td>Mot. or Gen.</td>
<td>Drive</td>
</tr>
<tr>
<td>Torque requests determined by control strategy.</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Deceleration</td>
<td>On</td>
<td>Off</td>
<td>Gen.</td>
<td>Drive</td>
</tr>
<tr>
<td>Regenerative braking by EM as HV battery pack allows.</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>4WD</td>
<td>On</td>
<td>Gen.</td>
<td>Mot. or Gen.</td>
<td>Drive</td>
</tr>
<tr>
<td>EM receives continuous power through HV bus from ISA.</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table 5.1: Vehicle Modes Description

the HV battery pack can amply power the electrical accessories; if the SOC depletes too low, the vehicle transitions directly into the *Engine Start* mode. Otherwise, the vehicle rests in the *Vehicle Idle* mode until a positive accelerator pedal position input from the driver. At this point, the vehicle launches from rest via the tractive electric machine and operates in an *Electric Only* mode. The EM both accelerates and decelerates (through regenerative braking) the vehicle in this mode. Three conditions exist at which point the vehicle leaves the *Electric Only* mode: (i) if the vehicle velocity exceeds that of a predetermined threshold, (ii) if the accelerator pedal is fully depressed and thus indicating the driver’s desire for additional power (from the engine), or (iii) if the HV battery SOC depletes to a level not conducive to supporting operation of the EM. The *Engine Start* mode is very brief; the vehicle resides in this mode so long as
for the ISA to accelerate the Diesel engine to a proper operational speed. The vehicle itself continues to be propelled by the tractive electric machine in a manner similar to that of the Electric Only mode. The commencement of fuel injection into the Diesel engine signifies the complete transition into the most-complex Normal mode. In the Normal mode, all three powertrain actuators operate in a manner according to further control theory as explained later in this chapter. The selection of the automatic transaxle gear is summarized in Figure 5.1: up and down shifts commence as the engine velocity reaches certain speed thresholds. These thresholds are functions of accelerator pedal position \((\dot{\omega}_{\text{ice,up}} = \dot{\omega}_{\text{ice,up}}(\alpha) \text{ and } \dot{\omega}_{\text{ice,dn}} = \dot{\omega}_{\text{ice,dn}}(\alpha))\) so that the Diesel engine operates at a higher speed with a greater potential power output as the accelerator pedal position is further depressed. At smaller accelerator pedal position depressions, the Diesel engine up shifts at a lesser velocity to maintain operation at a higher efficiency for a given power demand. The Normal mode is primarily
defined for zero or positive power requests; when the power request turns negative (a nonzero depression of the brake pedal), the vehicle transitions into the Deceleration mode. The vehicle decelerates by a combination of regenerative braking and the conventional friction brake system. To reduce the frequency of engine start/stops (especially from light or short brake pedal depressions), the Diesel engine remains in an idling condition. Transaxle downshifts occur as the engine speed decreases below a predetermined threshold value. The HEV either reverts to the Normal mode as the power request turns positive or transitions into the Electric Only mode once the vehicle velocity drops below a certain threshold (given the HV battery pack at a sufficient SOC). It should be noted that an intermediate Engine Stop mode will likely exist between the Deceleration and Electric Only modes to provide a quick and seamless stop of the Diesel engine. The progression returns to the Vehicle Idle mode as the vehicle decelerates to a complete rest.

From any vehicle mode, the driver is able to command the 4WD mode, although the transition takes place through the Engine Start mode. In the 4WD mode, the Diesel engine powers the front axle while also generating electrical power via the ISA. This power travels to the EM on a directly connected high-voltage electrical bus; thus, the HV battery pack acts as an electrical buffer to greatly mitigate power losses. The 4WD mode is unique to the chosen architecture and provides the opportunity for continuous (limited by the power rating of the ISA) four-wheel drive (ideal for off-road, inclement weather, or low-friction surface conditions). Intensive details of the 4WD mode will not be discussed in this thesis as the control necessitates a completely different approach. Specifically, 4WD control must strive to closely maintain similar
front and rear axle velocities, while equal axle speeds are assumed for the quasi-static model of Chapter 3 that aids in validation of the high-level control strategy as developed in this thesis.

5.3 Power Request Satisfaction

Instantaneous fuel economy and power deliverance of a vehicle typically exhibit an inverse relationship to each other: the increase of one commonly results in the decrease of the other and vice versa. The goal of the great majority of HEVs is to increase fuel economy; however, the control strategy must also command the powertrain actuators to continuously meet the driver’s local power request. This requirement not only satisfies the driver (and thus creates a dependence of fuel economy on driving behavior) but also aids in the overall reliability, predictability, and safety of the vehicle.

To meet the power request, the control strategy must know the power request; thus, it is necessary to continuously calculate the power request (which varies between vehicle modes and depends upon several states and inputs). The particular through-the-road HEV architecture as chosen requires analysis of the power request as perceived at the road. The presence of the torque converter effectively (but not completely in a mathematical sense) uncouples the engine speed from the other states of the vehicle. Consequently, the torque converter efficiency is highly variant (unlike the assumed constant efficiencies of the transaxle and gearbox), and thus the effect of the engine at the road varies greatly as well. The importance of recognizing this concept is apparent when considering the fuel economy strategy of Section 5.4: commanding powertrain actuators to minimize equivalent fuel consumption while meeting a power request defined on the shaft output power capabilities does not make sense.
when using fuel economy (overall vehicle system result) as a metric. Therefore, the
general form of the power request is $P_{rd,rq} = \alpha \cdot P_{rd,ma} + \beta \cdot P_{rd,mn}$. The maximum
power potential at the road is defined to be solely the contribution from the three
powertrain actuators, while the minimum power further considers the conventional
friction brake system. Specifically, every relevant mode assumes a parallel braking
system in which a brake pedal depression simultaneously causes both friction brak-
ing and regenerative braking. It should be noted that the maximum and minimum
road power potential varies between vehicle modes and/or as a result of vehicle state
(because of the state of the engine, HV battery pack limitations, vehicle stability
considerations, etc.). However, the control strategy assumes that the driver uses the
vehicle response as the basis for subsequent pedal commands (instead of absolute
pedal displacements) and thus makes the proper pedal position adjustments in an
attempt to achieve the desired vehicle response.

By definition, the power request in the Vehicle Idle mode is zero since the vehicle
exists in a resting condition while the accelerator pedal position remains undisturbed.
The instant that the driver depresses the accelerator pedal (and thus enters the Electric Only mode) a power request ensues; however, this power request is initially
somewhat ambiguous since the vehicle velocity, and thus power availability, is zero.
Therefore, for modes in which the power request consideration at the road proves
redundant because of lone operation from a single powertrain actuator, the power
request at the road converts into a torque request at the component output shaft.
Moreover, for all modes, the defined power request must be satisfied by a series of
powertrain actuator torque requests, since the powertrain actuators commonly accept
a torque request (as opposed to a power request) as an input. For the Electric Only

88
mode, the torque request is singularly met by the EM which implies:

\[ T_{\text{ice},r} = 0 \]  
(5.1)

\[ T_{\text{iso},r} = 0 \]  
(5.2)

\[ T_{\text{em},r} = \alpha \cdot T_{\text{em, bt, max}} + \zeta_{\text{em}} \cdot \beta \cdot T_{\text{em, bt, mn}} \]  
(5.3)

The maximum and minimum torque curves (as a function of rotor speed) incorporate the HV battery pack power limitations of Equations 3.29 and 3.30. A gain factor \( \zeta_{\text{em}} \) multiplies the brake pedal position to resultantly increase the proportionality of regenerative braking from the EM. As before, this product saturates such that \( \zeta_{\text{em}} \cdot \beta \in [0, 1] \). The gain can also be adjusted to decrease regenerative braking at a large HV battery pack state-of-charge and further regulates regenerative braking during situations of suspect vehicle stability (due to overly excessive rear-axle braking).

The Engine Start mode requires the EM to continue operation in a fashion as described by Equation 5.3. Torque output from the Diesel engine itself remains zero as it accelerates to an operational velocity via the ISA. The torque request to perform the engine start operation is modeled as in a proportional-integral fashion:

\[ T_{\text{ice},r} = 0 \]  
(5.4)

\[ T_{\text{iso},r} = P_{\text{iso}} \cdot (\dot{\omega}_{\text{ice}} - \omega_{\text{ice}}) + I_{\text{iso}} \int_{0}^{t_e} (\dot{\omega}_{\text{ice}} - \omega_{\text{ice}}) \, dt \]  
(5.5)

\[ T_{\text{em},r} = \alpha \cdot T_{\text{em, bt, max}} + \zeta_{\text{em}} \cdot \beta \cdot T_{\text{em, bt, mn}} \]  
(5.6)

The concern of a zero velocity does not exist in the Normal mode because a release of the brake pedal causes vehicle movement from Diesel engine power transfer through the torque converter (even at an engine idling condition). Any combination of the three powertrain actuators can contribute power in the Normal mode; therefore, the
power request must consider the power that can be delivered to the road. This power request (solely defined as positive for the Normal mode) can be expressed as:

\[ P_{rd,rq} = \alpha \cdot \left( \dot{\omega}_{ice} \cdot T_{ice,m} \cdot \eta_{tc} \cdot \eta_{tr} + P_{rd,ma,ce} \right) \]  

(5.7)

\( P_{rd,ma,ce} \) represents the maximum power that can be delivered to the road through a combination of electric machine torque outputs while respecting the discharging power limitation of the HV battery pack:

\[ P_{rd,ma,ce} = \max \left[ \frac{\omega_{ice} \cdot T_{isa} \cdot \eta_{tc} \cdot \eta_{tr} + \omega_{em} \cdot T_{em} \cdot \eta_{gb}}{\eta_{isa}} \right] \]  

(5.8)

\[ \frac{\omega_{ice} \cdot T_{isa}}{\eta_{isa}} + \frac{\omega_{em} \cdot T_{em}}{\eta_{em}} \leq P_{bt,ma} \]  

(5.9)

The actual determination of the powertrain actuator torque outputs to satisfy the power request further depends on the fuel minimization approach as described in Section 5.4.

The Deceleration mode solely satisfies negative power requests through a combination of conventional friction braking and regenerative braking. The EM singularly supplies a negative torque to the driveline such that:

\[ T_{ice,rq} = 0 \]  

(5.10)

\[ T_{isa,rq} = 0 \]  

(5.11)

\[ T_{em,rq} = \zeta_{em} \cdot \beta \cdot T_{em,\beta,\text{mn}} \]  

(5.12)

As in the Electric Only mode, the \( \zeta_{em} \) factor can work to increase regenerative braking for improved energy efficiency or can decrease regenerative braking to ensure vehicle stability and respect high-voltage battery pack SOC limits.
5.4 Fuel Consumption Minimization

A primary reason (and benefit from a consumer standpoint) for hybrid technology in automotive applications is to minimize the vehicle fuel consumption. This goal exists in a global sense: it is desired to command the powertrain actuators so as to minimize fuel consumption over the complete life of the HEV:

\[
\left\{ T_{i(t)} , T_{ia}(t) , T_{em}(t) \right\} = \text{arg min} \int_{0}^{t_{f}} \dot{m}_f dt \quad \text{for } t = 0 \ldots t_{f} \quad (5.13)
\]

This responsibility of the supervisory control strategy to command the powertrain actuators so as to minimize global fuel consumption is only relevant for the mode(s) that contain a sufficient number of degrees of freedom. In other words, fuel consumption minimization logic only occurs in the Normal mode in which the positive power request can be satisfied by numerous combinations of powertrain actuator power deliveries. The complete control for all other modes (Electric Only, Deceleration, etc.) has been explained in Section 5.3 and simply realizes the particular advantages (e.g., electric launch, regenerative braking) for which those modes are identified for.

It is relevant to note that the driving cycle over the life of the vehicle, or even the near future, is not known beforehand. Therefore, the global goal of Equation 5.13 must be addressed in real-time with a particular and local control solution. The chosen method for solvation of this issue involves considering the instantaneous (local) rate of equivalent fuel consumption. This general approach is labeled the Equivalent Consumption Minimization Strategy [27], [24], [25] which, while satisfying the power request of Equations 5.7 - 5.9, strives to reduce the global challenge of Equation 5.13 into the local problem:

\[
\left\{ T_{i(t)} , T_{ia} , T_{em} \right\} = \text{arg min} \dot{m}_{f,eq} \quad \text{at each } t \quad (5.14)
\]
The basis of the Equivalent Consumption Minimization Strategy (ECMS) recognizes that, for a charge-sustaining hybrid-electric architecture, all vehicle energy originates solely from the fuel tank (i.e., the electrical HV battery pack acts as an energy buffer). Further application of this reasoning leads to mathematical definitions of the local equivalent fuel consumption. For instance, an extraction of power from the HV battery pack (for use through the electric machines) may instantaneously save a quantity of Diesel fuel, but this immediate advantage requires an electrical replenishing at some point in the future (nonlocally). Consequently, the power drawn from the high-voltage battery correlates to an instantaneous equivalent fuel usage. The formulation of an equation that expresses this equivalent fuel usage considers both the HV battery pack efficiency and an average charging efficiency that represents a theoretical energy path from the Diesel engine to the high-voltage battery pack; Figure 5.3 illustrates the concept with the equation as:

\[
\dot{m}_{f, bt, ea} = \frac{P_{bt}}{Q_{bvo, d} \cdot \eta_{bt} \cdot \eta_{eh}}
\]

(5.15)

In Normal mode operation, driving electrical power into the HV battery pack requires further effort (and thus fuel) from the ICE, but this instant burden facilitates a contribution from the electrical powertrain at some moment in the future. Therefore, the electrical power input into the HV battery pack associates with an instantaneous equivalent fuel savings. The equation formulation again considers the high-voltage battery pack efficiency and the average efficiency of a discharging path, such that:

\[
\dot{m}_{f, bt, eq} = \frac{P_{bt} \cdot \eta_{bs}}{Q_{bvo, d} \cdot \eta_{dis}}
\]

(5.16)

Appropriately then, the total equivalent fuel consumption term in Equation 5.14 simply adds the rate of Diesel fuel consumption to the electrical equivalent rate of
Equivalent Fuel Consumption Usage

The fuel consumption into or out of the high-voltage battery pack:

$$\dot{m}_{f,eq} = \dot{m}_f + \dot{m}_{f, bt, eq}$$  \hspace{1cm} (5.17)

A source of ambiguity with the ECMS approach involves a definition of the $\bar{\eta}_{ch}$ and $\bar{\eta}_{dis}$ values. This thesis refers to these parameters as the average charging and discharging combined efficiency values. In this sense, $\bar{\eta}_{ch}$ represents the series of inefficiencies that a unit of energy may experience during an average route from the fuel tank to the HV battery pack (to increase the SOC of that pack):

$$\bar{\eta}_{ch} \equiv \bar{\eta}_{bt} \cdot \ldots \cdot \bar{\eta}_{ree}$$  \hspace{1cm} (5.18)

Note that, as depicted in Equation 5.18, the complete theoretical equation is not even known as the power-split HEV architecture presents two electric machines and a corresponding multitude of energy paths. In reality, a combination of efficiencies from
both electric machines and drivetrains in a lumped fashion may be more appropriate, but an accurate determination of this would necessitate a functioning control strategy that depends on those same values as trying to be estimated. The same imprecision can be seen in a representative equation for the $\bar{\eta}_{dis}$ value:

$$\bar{\eta}_{dis} \equiv \frac{\bar{\eta}_{hce}}{\bar{\eta}_{iso}} \cdot \ldots$$ \hspace{1cm} (5.19)

Note that, although Equations 5.18 and 5.19 may increase theoretical understanding of the general meaning of the average combined efficiency values, the calculation of these values is still ambiguous as the average efficiencies of the components are not known \textit{a priori}. Therefore, utilization of the ECMS requires either an assumption of the average efficiency values and average energy path or a determination of the most optimal values through modeling, simulation, and/or experimentation. The latter approach typically works better in achieving minimal fuel consumption as the average
combined efficiency values are treated as calibration parameters (which surfaces a
great advantage of the Equivalent Consumption Minimization Strategy: it can be
applied to a wide variety of HEV configurations). Further details of the specific
determination of the $\tilde{\eta}_{ch}$ and $\tilde{\eta}_{ch}$ calibration parameters are discussed in Section 5.5
and Subsection 6.2.3.

5.5 High-Voltage Battery Pack State-of-Charge Management

Charge-sustaining architectures dominate the current hybrid-electric vehicle mar-
et as a consequence of their transparency upon the typical consumer; the driver may
notice the benefits (improved fuel economy, smooth vehicle launch, etc.) but really
appreciates the normalcy as compared to conventional vehicles. In other words, the
consumer does not need to electrically charge a charge-sustaining hybrid: as with
conventional vehicles, trips to the gas station (hopefully less frequent) are all that are
required. From an energy management standpoint, plug-in (charge-depleting) hybrids
particularly offer benefits by allowing an electric only mode of vehicle operation until
the HV battery pack reaches a minimum SOC threshold, at which point the vehicle
operates in a fashion similar to that of a charge-sustaining HEV [37]. Nevertheless,
the primary control strategy was chosen to maintain a charge-sustaining vehicle as to
retain consistency with the reliable standard (as it relates to consumer acceptability)
of the current class of production hybrid-electric vehicles.

The premise of a charge-sustaining HEV involves maintaining the HV battery
pack SOC between two predefined boundaries:

$$s_{bt} \in [s_{bt,l}, s_{bt,u}]$$

(5.20)
These limits for a specific battery chemistry are chosen so as to ensure a sufficient life of the HV battery pack, be consistent with the voltage range of the inverters of both electric machines, and avoid excessively large internal resistances. As mentioned in Section 5.2, the logic for mode selection inherently (to an extent and for some of the modes) maintains high-voltage battery pack SOC within its limits by switching modes as deemed appropriate. For instance, the vehicle transitions out of the Vehicle Idle and Electric Only modes if the state-of-charge depletes too low. Further, the $\zeta_{em}$ factor fades out regenerative braking at large state-of-charge values so as to not exceed the upper boundary. However, the Normal mode, as explained thus far to satisfy the power request and minimize fuel consumption, does not inherently contain any HV battery pack SOC control. The state-of-charge control for the Normal mode is implemented in the following two ways:

1. Strict limits for the high-voltage battery pack state-of-charge

2. Adaptive tuning of the combined average efficiency values

The strict limits control simply acts as a hard boundary: it will not allow the electric machines to operate in such a way that influences the HV battery pack SOC to drift outside of the predetermined boundaries. The method taken to implement the strict limit involves formulating a correction factor on the maximum power into or out of the high-voltage battery pack. Therefore, the power request satisfaction and fuel consumption minimization control techniques actually respect the corrected HV battery pack power limitations:

$$P_{bt,max,cor} = (1 - \zeta_{bt,dis}) \cdot P_{bt,max} \quad (5.21)$$

$$P_{bt,min,cor} = (1 - \zeta_{bt,chg}) \cdot P_{bt,min} \quad (5.22)$$
As seen in Figure 5.5, the power correction factors are defined as a function of HV battery pack SOC so that they impart a negligible influence when the state-of-charge does not threaten either its upper or lower limit. Numerous equations can produce the desired effect as influenced by the curve such as that depicted in Figure 5.5; the curves as shown result from use of a Lyapunov potential function which takes the general form:

\[
\tau = A \cdot \left( \frac{1}{ \left( (\hat{s}_{bl,u} - \hat{s}_{bl,nom})^p - (\hat{s}_{bl} - \hat{s}_{bl,nom})^p \right)^{1/p}} - \frac{1}{ (\hat{s}_{bl,u} - \hat{s}_{bl,nom}) } \right) \quad (5.23)
\]

where \( p \) equates to an even integer that affects the gradient (steepness) of the curve near the upper and lower limits.

![High-Voltage Battery Power Limit Correction Factors](image)

**Figure 5.5: HV Battery Pack Power Correction Factors**

As described in Section 5.4, there exists a certain level of ambiguity with the combined average efficiency values \( \bar{\eta}_{ch} \) and \( \bar{\eta}_{dis} \). Modeling and simulation reveal better
combinations of these calibration values (typically between zero and one) than others in terms of achieving high fuel economy for a certain driving cycle. Most often, the preferred values influence a HV battery pack SOC profile that fluctuates freely between the state-of-charge limits without hovering near or frequently reaching the boundaries. Therefore, it is reasonable to assume that maximum fuel economy is achieved if the HV battery pack SOC profile has a continuous freedom to fluctuate in either direction; in other words, when the state-of-charge avoids reaching either strict limit. This logic is the reason behind the implementation of the adaptive tuning control with respect to the HV battery pack SOC as described subsequently.

Expectantly, variation of the combined average efficiency values influences the high-voltage battery pack to have a tendency to charge or discharge. For instance, an alteration of the values towards zero increases the magnitude of the equivalent electrical fuel consumption and thus results in an overall tendency to charge of the HV battery pack. Oppositely, an adjustment of $\tilde{\eta}_{ch}$ and $\tilde{\eta}_{kh}$ towards one decreases the magnitude of the equivalent fuel consumption and thus causes increased depletion of the HV battery pack. Therefore, on-line tuning of the combined average efficiency values presents a method for state-of-charge control:

\begin{align}
\tilde{\eta}_{ch} &= \zeta_{ch} \cdot \tilde{\eta}_{ch,0} \\
\tilde{\eta}_{dis} &= \zeta_{dis} \cdot \tilde{\eta}_{dis,0}
\end{align}

(5.24)

(5.25)

The overall goal with this approach involves keeping the average combined efficiencies close to "optimal," nominal values; in this way, HV battery pack SOC can be maintained and fuel economy kept high without knowing any driving cycles a priori. A further application of the adaptive tuning method considers the recent driving history of the vehicle in an attempt to predict the near driving future. A decent
prediction can lead to less tuning or, at the very least, provide a base value for which to tune from. The actual tuning approach as determined through simulation results are discussed in Subsection 6.2.3.

5.6 Drivability Assurance

Along with those previous control strategy objectives as discussed, a HEV must exhibit a certain level of drivability. The term drivability can generally be defined as the degree to which the vehicle operation responds favorably to driver requests and commands as evaluated with respect to the driver's qualitative opinion. This definition implies, among others, dealing with both the ability of the vehicle to satisfy the driver's power request and the ability to coordinate the powertrain actuators in an acceptably smooth and synergistic manner. As Section 5.3 considers the former concern, this thesis solely associates drivability with the latter concern.

To maintain sufficient drivability (as defined previously) in a HEV, the control strategy must aim to transparentize the complexity of that hybrid-electric vehicle architecture and its operation; with the exception of the technologically savvy consumer, the average driver does not care about particulars of the hybrid-electric powertrain operation but is more concerned with the vehicle's familiarity as compared to conventional vehicle operation. Familiar operation of a vehicle is a vague and subjective qualitative property, but certain quantitative metrics have been identified to serve as guidelines for satisfying those primary drivability concerns of drivers. Several of these metrics are outlined in Table 5.2.
<table>
<thead>
<tr>
<th>Metric</th>
<th>Values and Reaction</th>
</tr>
</thead>
<tbody>
<tr>
<td>RMS acceleration</td>
<td>$&lt; 0.315 , m/s^2$ : Not uncomfortable</td>
</tr>
<tr>
<td>Transient vibration</td>
<td>$0.8g \sim 0.9g$ : Normal</td>
</tr>
<tr>
<td>Jerk</td>
<td>$\pm 1.0 , m/s^3$ : Comfortable</td>
</tr>
<tr>
<td>Vibration dose value</td>
<td>$8.5 , m/s^{1.75}$ : Caution zone</td>
</tr>
</tbody>
</table>

Table 5.2: Drivability Metrics [40]

Although the specific approach to achieve acceptable drivability has not yet been completely determined, the control strategy will generally attempt to control the powertrain actuators and vehicle mode transitions in such a way as to respect the metric limits as in Table 5.2. Further details of drivability control methods are beyond the scope of this thesis as this lower-level control requires an adaptation to the behavior of the specific powertrain components. Consequently, the development of drivability control is more productive when experimental measurements can be taken from a complete hybrid-electric powertrain testing apparatus as to indicate the situations for said drivability control. Nevertheless, to complement eventual hybrid-electric powertrain testing, the dynamic model of Chapter 4 forms a basis for future development of a dynamic model-based computer vehicle simulator.

5.7 Secondary Control Strategy

Intensive details about the secondary control strategy are beyond the scope and goals of this thesis; however, for completeness, a few of the objectives are recognized
and mentioned in this section. The complexity of the secondary control strategy is significantly less than that of the primary (supervisory) control strategy. Still, this level of control is required to meet the vehicle technical specification with respect to emissions as well as ensuring safety and functionality on both a component and vehicle level.

5.7.1 Exhaust Aftertreatment

As described in Section 2.10, the proposed exhaust aftertreatment system consists of a Diesel particulate filter and lean NO\textsubscript{x} trap catalytic arrangement with a bypass valve and a novel flame reformer. This assembly is flanked on both sides by small oxidation catalysts. A sizable advantage of this particular emissions reduction system is the control simplicity: it effectively operates in a passive manner and only requires a determination of the appropriate time for regeneration of the DPF and LNT. In the most basic approach, a differential pressure sensor across the DPF indicates the necessity for regeneration once the pressure exceeds that of a predetermined threshold value. For the lean LNT trap, an exhaust gas oxygen sensor combined with a dynamic NO\textsubscript{x} estimator model predicts the NO\textsubscript{x} stored in the trap and, if successful, eliminates the need for expensive NO\textsubscript{x} sensors. The operation of the flame reformer also requires a minimal level of control to output the correct but different fuel-to-air ratios for regeneration of the Diesel particulate filter and lean NO\textsubscript{x} trap. From a control standpoint, it is again worthy to note that this system's independence from the ICE eliminates the need for expensive and time-consuming Diesel engine calibration that is required in conjunction with conventional, engine-based regeneration techniques.
The implementation of homogeneous-charge compression-ignition on the Diesel engine necessitates further control. Unlike the emission-reduction system as described previously, this control directly affects the operation of the Diesel engine. Details have not yet been determined, but the integration of homogeneous-charge compression-ignition at low loads requires control of both the atomizer and transition between conventional direct injection and HCCI operation.

Emission-reduction challenges are usually highly and singularly related to the type of engine in the vehicle; in other words, any vehicle (no matter the configuration) with a Diesel engine typically presents the same emission challenges on the same order of magnitude. However, the chosen HEV architecture and developed control strategy also provides a few additional and unique opportunities for emissions reduction. For instance, when the control strategy in the Normal mode has a tendency to load shift the already downsized Diesel engine, the resulting higher exhaust gas temperatures increases catalytic efficiency and thus relieves some responsibility from the flame reformer to provide that heat. The start-stop strategy also keeps the catalysts warmer at a vehicle rest by eliminating the relatively cool exhaust gas flow produced by an engine during idling. Another method of reducing emissions relates to the Equivalent Consumption Minimization Strategy as formulated in Section 5.4. The dynamic estimator model (as required to determine lean NOx trap storage capacity) estimates engine-out NOx at a particular engine operating point. Equating the engine-out NOx to an equivalent fuel consumption expands Equation 5.17 to:

\[ \dot{m}_{f,eq} = \dot{m}_f + \dot{m}_{f,\text{loss}} + \dot{m}_{f,NOx,eq} \]  \hspace{1cm} (5.26)

In this way, the control strategy can simultaneously attempt to minimize fuel consumption as well as engine-out NOx emissions. The exact form of \( \dot{m}_{f,NOx,eq} \) has not
yet been determined, but the expression may imitate:

$$\dot{m}_{f,NO_x,eq} = \gamma_{NO_x} \cdot \dot{m}_{NO_x}$$

(5.27)

where the $\gamma_{NO_x}$ factor can be tuned to appropriately represent the respective priority of minimizing engine-out NOx as compared to fuel consumption. This determination will most likely depend on the efficiency and capability of the previously described exhaust aftertreatment system to meet the vehicle technical specifications with respect to emissions.

5.7.2 Component Cooling and Safety Assurance

The majority of the control for cooling and safety monitoring will be very basic and will react to states of and inputs to the powertrain components. As mentioned previously, the secondary control will operate independently of the primary control except in emergency shut-off situations. Some secondary control responsibilities include:

- The actuation of the power electronics cooling loop pump when the vehicle is moving.
- The actuation of the EM gearbox oil pump when the vehicle is moving.
- The actuation of the cooling fans when necessary (engine coolant overly hot, power electronics coolant overly hot, air conditioning active, etc.).
- The actuation of the HV battery pack fans such that the battery modules do not exceed a certain temperature.
- The monitoring of the engine oil pressure such that the engine can be immobilized if pressure is lost.

103
• The monitoring of the rotational speeds of the engine and electric machines such that the appropriate action (disable component or vehicle) can take place if the components reach their maximum mechanical speeds.

By taking responsibility of the above actions, the secondary controller acts as an equivalent on-board diagnostics unit for the hybrid-electric system. The secondary controller’s responsibilities further extend to resetting the primary controller if a reset is deemed necessary for any of a number of reasons. Finally, the secondary controller supports the vehicle’s limp-home mode: if the primary controller and/or hybrid-electric system fail, the secondary controller facilitates vehicle operation in a Diesel-only mode until the vehicle can be driven to an area of safety.

5.8 Summary

During typical operation, the high-level supervisory control strategy initially selects the appropriate vehicle mode from those as summarized subsequently using logic based upon several driver inputs and system states. At a vehicle rest in the Vehicle Idle mode, the DC-DC converter powers all accessories while the engine is disabled. A nonzero accelerator pedal position transitions the vehicle into the Electric Only mode if sufficiently supported by the HV battery pack. At a certain translational vehicle velocity, the Engine Start mode quickly and smoothly accelerates the Diesel engine to a proper operational speed. The vehicle then immediately enters the Normal mode in which all three powertrain actuators may contribute in powering the vehicle. A brake pedal depression by the driver initiates the Deceleration mode which prompts regenerative braking in parallel with the conventional friction braking system. Deceleration below a certain threshold velocity transitions the vehicle back into the Electric
Only mode and eventually the Vehicle Idle mode if the vehicle brakes to a complete rest. While operating in a mode, the control strategy calculates the power request as a function of the normalized pedal positions and maximum and minimum power that could potentially be realized at the road. The control strategy then determines the torque requests from each of the powertrain actuators to satisfy the power request. In the multiple degree-of-freedom Normal mode, the appropriate torque split is found through local theory aimed at minimizing fuel consumption on a global basis. High-voltage battery pack state-of-charge control is indirectly achieved through manipulation of several variables in the fuel consumption minimization logic. Details notwithstanding, drivability assurance involves, at the minimum, appropriate adjustments of the previously determined torque requests so as to satisfy certain levels of several identified drivability metrics.

Although not a primary focus of this thesis, a secondary level of control presents several additional objectives. The proposed exhaust aftertreatment system primarily relies on models and measurements to estimate NOₓ storage and thus specify appropriate situations for regeneration. Engine-out emissions can also be reduced by equating engine-out NOₓ to an equivalent fuel consumption in an attempt to further bring tailpipe emission levels to that as stated in the vehicle technical specifications. Another objective of the secondary level of control involves ensuring component and vehicle safety, in which the secondary controller acts as a diagnostics unit for the hybrid-electric powertrain system.
CHAPTER 6

EXPERIMENTATION AND SIMULATION

The work associated with this thesis yielded investigation and validation of the proposed vehicle architecture and high-level supervisory control strategy through two means. First, rapid prototyping with a rolling chassis apparatus allowed functionality and concept feasibility testing of using the Delphi EV1 electric machine and intended HV battery pack for the Electric Only mode. Secondly, software-in-the-loop analysis with a quasi-static vehicle simulator provided the necessary preciseness for both vehicle architecture and control strategy investigation and validation with respect to energy management. This simulator may further act as a template for future construction of a dynamic vehicle simulator (based upon those equations as discussed in Chapter 4) for supervisory control strategy validation with respect to drivability.

6.1 Rapid Prototyping Experimentation

Rapid prototyping strives to assess and ensure compatibility between the component hardware, control hardware, and/or control strategy. As an extension from hardware-in-the-loop techniques, rapid prototyping is typically the first step in implementing the actual physical hardware component in route to realizing complete
system functionality. The rapid prototyping work of this thesis included construction of a testing apparatus, termed the rolling chassis, to validate: (i) component functionality, (ii) feasibility and practicality of desired component operation, and (iii) compatibility between elements of the hybrid-electric system in a controlled, convenient, and out-of-vehicle testing environment.

Delays in procurement of several key powertrain components limited the opportunity for complete hybrid-electric powertrain integration onto the rolling chassis; however, the apparatus facilitated progress towards validating component functionality and concept feasibility with respect to the Delphi EV1 electric machine and NiMH high-voltage battery pack. As mentioned previously, the Delphi EV1 electric machine is the proposed unit to reside in the rear of the vehicle and provide significant traction as necessitated in the Electric Only and Deceleration modes. The intended HV battery pack consists of 44 Panasonic Prismatic NiMH modules connected together in a series configuration. Figure 6.1 illustrates the rolling chassis apparatus resting on a chassis dynamometer and coupled to the EM.

The rolling chassis itself was a 2000 Chevrolet Suburban significantly stripped-down to allow freedom and flexibility in the mounting of components; in other words, the choice of the chassis frame was somewhat arbitrary and only acted as an aid in transmitting power to the dynamometer rollers. The chassis dynamometer was a single-axle unit and thus better suited exclusively front-wheel or rear-wheel vehicles. However, the chosen through-the-road HEV architecture transmitted power to the road through both axles as determined by the control strategy. Therefore, to mimic through-the-road operation on a single-axle chassis dynamometer, the EM resided outside the chassis frame and was directly coupled to the rolling chassis wheel. The
Figure 6.1: Rolling Chassis Apparatus

Delphi EM and attached Delco inverter sat, through the original mounts and in its intended orientation, on an aluminum frame anchored to the surrounding concrete floor. The spline shaft from the production GM EV1 vehicle, adapted appropriately, connects to a universal-joint slip driveshaft. The opposite end of the shaft then concentrically coupled to the wheel through an aluminum adaptive piece. Although the slip shaft allowed some axial movement between the electric machine and rolling chassis wheel, the rolling chassis itself (after centered on the chassis dynamometer rollers and secured in the rear) was strapped taught to minimize any relative translational
movement. Further, the tightened straps induced a downward force of the rolling chassis onto the dynamometer rollers that partially offset the lost weight as necessary for sufficient traction. Prevention of overly high temperatures in the EM and inverter during experimentation was accomplished by a routing of cold tap water through the appropriate channels. The HV battery pack and switchbox supplied and routed the electrical power to the Delco inverter. Both the battery cart and switchbox also embodied that which will be eventually implemented into the hybridized Equinox but with increased freedom for modification and troubleshooting. The EM and inverter were controlled by one of two methods: (i) analog control, or (ii) CAN control. Analog control was achieved by a custom control box which inputted 12V electrical power and outputted a throttle and brake analog voltage as varied through a potentiometer. The analog signals traveled through a 24-pole connector to the inverter, and several digital input signals transmitted through this fashion as well. Among these included signals for directionality of operation, controller card reset, and controller card emergency stop. All of these commands and readings could also be transmitted to and recorded from a connection to a CAN card mounted in the interior of the inverter housing. In CAN control, the accelerator and brake commands originated as analog signals from instrumented pedals on the rolling chassis itself, were converted into controller area network messages through the primary controller, and traveled to the inverter controller card over the CAN bus. The experimental results as illustrated and described subsequently used both the analog and CAN methods to command the electric machine but exclusively utilized the CAN as the convenient method to record data. To mimic the road load (Equation 3.78) as estimated to be experienced by a
hybridized Chevrolet Equinox, the dynamometer was programmed with appropriate parameters as described in Table B.1.

Both experimental tests as shown in Figures 6.2 and 6.3 generally consist of moderately gentle accelerations, a period of constant speed retention, and some coasting before deceleration to a rest. A relative lack of weight on the front of the rolling chassis prohibited complete traction between the tires and rollers during an immediate and large torque command. Therefore, in both tests, the dynamometer rollers were gently started to an initial speed before the onset of a typically larger torque as may be experienced in an actual electric launch situation. At this point, the EM accelerated the dynamometer rollers to an effective velocity upwards of 20 mph. Figure 6.3 depicts an acceleration from about 4 mph to almost 20 mph in just over
one second. Electrical current peaked at just over 60 amps during the acceleration phase; therefore, considering the battery module capability of an estimated 120 amps of intermittent current draw, it is reasonable to expect electric launch capabilities exceeding that as experimented. The notion of being power limited with the HV battery pack is confirmed in Figure 6.2, as a torque request of only 35% from the EM draws 70 amps (over half the estimated peak current). The rolling chassis testing further validates the functionality of regenerative braking using the Delphi EV1 electric machine. However, it is clear that much more aggressive regeneration is possible (up to an estimated negative 70 amps), and this is especially the case in Figure 6.2 in which the frictional braking system dominated for a given brake pedal depression.
As previously described in Section 5.3, an appropriate method of increasing the relative proportion of regenerative braking for a given deceleration request exists through modification of the $\zeta_{em}$ factor.

6.2 Software-in-the-Loop Simulation

The computer simulator that emulates the previously described quasi-static model of Chapter 3 is entitled cX-SIM (Challenge X Simulator). cX-SIM formed as an adaptation and iteration of several previous quasi-static vehicle simulators at the Ohio State University Center for Automotive Research. VP-SIM (Vehicle Performance Simulator) first provided a tool that allowed users to compose a vehicle architecture of interest from a library of unified and scalable drivetrain components [31]. FT-SIM (FutureTruck Simulator) facilitated vehicle simulation of the sole hybrid-electric architecture as chosen by the Ohio State team in the FutureTruck competition [24]. cX-SIM is the next iteration for use by the Ohio State team throughout the Challenge X competition.

cX-SIM itself singularly replicates the particular HEV architecture as decided upon; thus, cX-SIM is not intended to act as a tool to compare numerous vehicle architectures. Instead, cX-SIM provides a desired level of precision for comparative analysis of differing components and/or control strategies in conjunction with the chosen vehicle architecture. More pointedly, cX-SIM presents a tool for software-in-the-loop validation of the control strategy with respect to vehicle mode and transmission gear selection, HV battery pack SOC management, and fuel consumption minimization (i.e., energy management).
cX-SIM operates from the MATLAB, Simulink, and Stateflow tools as developed by The MathWorks. Simulink provides the user to choose the type of solver appropriate for a particular simulation. The absence of high-frequency dynamics in the quasi-static cX-SIM simulator abides the operation of the solver in a fixed time-step mode with a step size of around half a second. Therefore, a reasonable processor is able to solve the cX-SIM vehicle model itself relatively quickly (the control strategy calculations actually take the most processing power) and thus facilitates repetitive usage as the tool is designed for.

6.2.1 cX-SIM Graphical User Interface

As a tool to be used throughout the Challenge X competition, cX-SIM requires a user-friendly graphical user interface (as in Figure 6.4). The graphical user interface (GUI) illustrates the top-layer of the simulator (driver, hybrid-electric powertrain, and vehicle) as consistent with the model development of Chapter 3. The GUI also displays fuel economy information as the simulator functions and provides the opportunity to view revealing variables (speed, torque, HV battery pack SOC, etc.) through a scope. Before the simulation, the graphical user interface provides the user the opportunity to conveniently alter model parameters such as desired driving cycle, driver proportional-integral-derivative control coefficients, and vehicle curb mass. All other vehicle and component parameter data that does not encounter frequent alteration are stored in and initiated from the MATLAB m-file Initialization_and_Component_Data.m. In summary, this file processes the raw steady-state experimental or physical component data into the variables that cX-SIM will seek during operation. The file saves the processed data into a *.mat data file.
format; in this way, `Initialization_and_Component_Data.m` only requires to be run when data within the file has been changed, updated, or altered. The file also serves to provide graphical illustrations to the user as a presimulation visual check for correctness of component data. Several of these graphs, as well as a table of additional parameter values utilized in the simulator, are presented in Appendix B.

![cX-SIM Graphical User Interface](image)

Figure 6.4: cX-SIM Graphical User Interface

As a general summary and consistent with the model formulation of Chapter 3, the forward (with respect to causality) simulator feedbacks vehicle velocity to the driver who responds with an accelerator or brake pedal position that then actuates the hybrid-electric powertrain and consequently moves the vehicle. Specifically, the driver model compares a desired velocity profile (e.g., a FUDS cycle) to the current vehicle velocity and appropriately specifies an accelerator or brake position through PID control. These commands enter the vehicle controller in the hybrid-electric powertrain system where the control strategy functions as described previously in
route to specifying vehicle mode, transmission gear, and powertrain actuator torque commands. As depicted in Figure 6.5, torques flow through a series of subsystems that statically model each drivetrain component. The result is a total tractive force that enters the vehicle system and, after comparison to the present road load, initiates vehicle acceleration.

Figure 6.5: cX-SIM Hybrid-Electric Powertrain Structure
6.2.2 cX-SIM Control Strategy Implementation

Implementation of the high-level, supervisory control strategy (as developed thus far and described in Chapter 5) into cX-SIM requires, at the minimum, those inputs and states as shown in Figure 6.6. The accelerator pedal position facilitates calculation of positive power requests and also initiates transition between transaxle gears or vehicle modes. The brake pedal position is necessary to calculate negative power requests. The component velocities reveal the possible range of powertrain actuator torque outputs and act in conjunction with those torques to estimate efficiency or fueling rate. Component and vehicle velocities, along with the HV battery pack SOC, also commence gear shifting or vehicle mode transitions. The state-of-change is further instrumental in estimating battery module parameter values as utilized in several control strategy calculations. Outputs include torque requests to each of the three powertrain actuators and discrete commands with regards to engine state, transaxle gear, and torque converter lock-up.

It is worth recognizing the task order in which the control strategy functions in route to specifying the previously identified outputs. The initial responsibility, as depicted in Figure 6.7, is to select the vehicle mode and transmission gear. From this point, the powertrain actuator torque requests are calculated as appropriate for that particular vehicle mode.

Implementation of the control strategy into cX-SIM consists of several MATLAB tools. Stateflow performs the initial and event-based determination of vehicle mode and transmission gear. The relatively simple Vehicle Idle, Electric Only, Engine Start, and Deceleration modes singularly utilize Simulink block diagrams for control. The
more-complex Normal mode calls an S-function (ECMS_SFunction.m) that calculates the powertrain actuator torque split as detailed in the following paragraphs.

In general, the Normal mode S-function performs fuel minimization through an iterative approach of electric machine torques that satisfy all requests and respect all limitations. As inferred from Figure 6.7, the S-function inputs five necessary variables (ICE speed, EM speed, accelerator pedal position, and HV battery pack SOC) and outputs four commands (torque converter lock-up request\(^2\) and torque requests for the engine, ISA, and EM). The first task includes utilization of several of the states to interpolate necessary information from the *.mat files as created from the

\(^2\)It should be noted that, for the work of this thesis, the torque converter was assumed to never exist in a lock-up state. If control over this is indeed available, the strategy will most likely include some sort of event-based control similar to that used in the vehicle mode determination logic.
Initialization and Component Data.m file before running the simulation. Specifically, the HV battery pack SOC determines the average efficiency values \( \eta_{\text{ich}} \) and \( \eta_{\text{das}} \) and several battery module parameters (open-circuit voltage and internal resistances). An extreme state-of-charge value also induces a restriction on the use of the HV battery pack through corrected maximum and minimum battery power capabilities as previously depicted in Equations 5.21 and 5.22. The speeds allow an estimation of the maximum and minimum possible torques from the three powertrain actuators. At
this point, torques from the electric machines are discretized between their maximum and minimum values from which a series of efficiencies are then interpolated. These discretized torques form a basis for several subsequent calculations. Ideally, some sort of continuous minimization function in MATLAB could be utilized as an alternative to the iterative approach, but experimentation with this led to faulty solutions through local minimums.

The next calculation involves a determination of the maximum possible power to the road through evaluation of Equations 5.7 through 5.9. Specifically, each combination of discretized torques from the ISA and EM are input into Equation 5.8 and are only considered if the maximum battery power limitation (after correction due to HV battery pack SOC) of Equation 5.9 is respected. Inclusion of the accelerator pedal position then reveals the power request at the road. Each combination of discretized torques is also checked and noted for compliance with the minimum HV battery pack power limitation.

A final sequence of calculations considers only the combination of electric machine torques that respect the maximum and minimum HV battery pack power limitations as determined previously. For each combination of electric machine torques, the required engine torque to meet the power request at the road is back calculated. Knowledge of this torque, as well as the engine speed, facilitates an interpolation of the mass flow rate of fuel needed to support that operating point (given the calculated engine torque less than its maximum possible torque and greater than or equal to zero). The discretized electric machine torques and known angular velocities satisfy that required to interpolate operational efficiency, from which the net electrical power into or out of the HV battery pack is calculated. From this net electrical power value
and the battery module parameters as previously found, the electrical current, HV battery pack efficiency, and equivalent mass flow rate of fuel associated with the HV battery pack can then be found. The torque combination of powertrain actuator torque outputs that results in the minimum total equivalent mass flow rate of fuel is then considered to be the “best” solution given the driver inputs and system states.

6.2.3 cX-SIM Simulation Results

Unless otherwise noted, cX-SIM simulations used vehicle and component parameter values as specified in Appendix B.2.

Vehicle Mode Effects

The initial set of cX-SIM simulations aimed to compare Equinox experimental data with simulated data, reveal the benefits of converting to a downsized Diesel engine, and expose the inherent benefits and impacts of several of the previously described vehicle modes without full vehicle hybridization. Table 5.1 outlines the primary results from those simulations.

On their Chevrolet website, General Motors declares fuel economy ratings of 19 and 25 miles per gasoline gallon for the FUDS and FHDS cycle, respectively. Experimental test results provided by General Motors (for analysis in the Challenge X competition) differs slightly over the urban cycle and significantly exceeds the published value for highway driving. The same collection of experimental data reveals a quick 0 - 60 mph acceleration time of 9.23 seconds. For initial comparison purposes, cX-SIM was slightly modified so as to simulate the stock Chevrolet Equinox with the correct stock gasoline engine, transaxle ratios, curb mass, etc. The simulated stock

120
<table>
<thead>
<tr>
<th>Test #</th>
<th>FUDS (mpgge)</th>
<th>FHDS (mpgge)</th>
<th>0-60 mph (s)</th>
<th>Notes</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>19</td>
<td>25</td>
<td>-</td>
<td>Declared by GM [9]</td>
</tr>
<tr>
<td>2</td>
<td>20.7</td>
<td>30.1</td>
<td>9.23</td>
<td>Experimental test results [3]</td>
</tr>
<tr>
<td>3</td>
<td>18.8</td>
<td>26.7</td>
<td>11.00</td>
<td>cX-SIM, 3.4L gasoline, 1722 kg</td>
</tr>
<tr>
<td>4</td>
<td>27.9</td>
<td>31.8</td>
<td>12.39</td>
<td>cX-SIM, 1.9L Diesel, 1722 kg</td>
</tr>
<tr>
<td>5</td>
<td>26.1</td>
<td>30.4</td>
<td>13.75</td>
<td>cX-SIM, 1.9L Diesel, 1900 kg</td>
</tr>
<tr>
<td>6</td>
<td>26.5</td>
<td>30.4</td>
<td>II</td>
<td>+ Diesel engine “on/off”</td>
</tr>
<tr>
<td>7</td>
<td>29.3</td>
<td>31.0</td>
<td>!</td>
<td>+ Deceleration mode</td>
</tr>
<tr>
<td>8</td>
<td>29.0</td>
<td>31.0</td>
<td>II</td>
<td>+ Electric Only mode</td>
</tr>
</tbody>
</table>

Table 6.1: cX-SIM Simulation Results, Tests # 1 - 8

Equinox posts fuel economy and acceleration metrics slightly different than that experimentally found (and actually closer to those values originally declared by GM); however, it should be again noted that cX-SIM acts as a comparative analysis tool that strives for precision but not necessarily great accuracy. Inspection of Figures 6.8 and 6.9 reveals a possible reason for the fuel economy digression: the cX-SIM gasoline engine tends to consistently operate at a higher velocity than that recorded during experimentation.

Figure B.3 confirms the generality that, for a given power demand, efficiency decreases with increasing engine velocity. The apparent shift in engine velocity may be due to incorrect torque converter characterizing parameter values, as those utilized in the simulator are not specific for the actual torque converter in the Equinox. The model parameter values for the torque converter accompanying the Fiat Diesel engine and transaxle are also unknown, so a noncommon torque converter model was chosen to maintain a level of consistency in the comparative analysis. Another possibility
Figure 6.8: Test # 2 and 3: FUDS Cycle

Figure 6.9: Test # 2 and 3: FHDS Cycle
for the engine speed differential between experiment and simulation lies in the quasi-static nature of the model: any engine torque immediately induces a certain pump velocity as opposed to working to accelerate a rotational inertia to that speed. The simulated vehicle significantly lags the experimented Equinox in the acceleration trial; one potential reason for the difference may be an incorrect assumption of the magnitude of the effective vehicle inertia as used in the simulation. Inspection of Figure 6.10 reveals that the gradient of both the engine speed and vehicle velocity are most noticeably larger for the experimental vehicle while in first gear. Therefore, the mass factor (of Equation 3.81) may be unrepresentatively large for a transaxle in first gear. However, the overall similarities in the engine shifting strategy are evident which again conveys the preciseness that cX-SIM strives for. Therefore, the fuel economy and performance results of Test # 3 in Table 6.1, albeit varying from the declared and experimental values, will act as the reference (when using cX-SIM) in assessing progress towards the desired percent improvement over the stock vehicle as derived from the vehicle technical specifications.

The transformation to the 1.9L Fiat Diesel engine from the stock 3.4L gasoline engine necessitated a few accompanying changes. Namely, the stock 5-speed automatic transaxle ratios were converted into the anticipated 6-speed automatic transaxle gear ratios as appropriate for a Diesel engine, and the shifting strategy was modified to adapt to the lower maximum operational speed of the Diesel engine. Keeping all else constant, substitution of the downsized Diesel engine for the stock gasoline engine results in a drastic fuel economy improvement of 48.4% and 19.1% on the FUDS and FHDS cycles, respectively (Test # 4). However, the degradation of acceleration ability emphasizes the importance of hybridization to supplement the downsized engine.
Figure 6.10: Test #2 and 3: 0-60 mph Acceleration

(eespecially with the performance expectations of a sport-utility vehicle). Even with a larger low-end torque, the Diesel engine alone cannot account for the lost power and the acceleration time drops almost 13%. In anticipation of the significant addition of power electronics hardware in the hybridization process, the simulated vehicle mass was raised to 1900 kg (from the stock curb mass of 1722 kg) for Test #5 and all subsequent simulations.

Expectantly, the acceleration performance (Figure 6.11) suffers additionally with the increase in mass, and it is estimated that the eventual hybrid system will have cause a reduction of a simulated 2.75 seconds (20%) to salvage the stock acceleration ability as targeted in the vehicle technical specifications. Fuel economy also degrades noticeably over both the FUDS and FHDS cycles with a slightly larger change over
Figure 6.11: Test # 5: 0-60 mph Acceleration

Figure 6.12: Test # 5: Engine Operating Points
the urban route due to its more frequent and substantial accelerations of the vehicle inertia. However, the improvement over the simulated stock Equinox still measures at an impressive 38.8% (FUDS) and 13.9% (FHDS). The reason for the dramatic fuel economy increase is apparent through comparison of Figure 6.12 with Figures 6.8 and 6.9: the downsized Diesel engine operates in regions of much larger efficiency than that of the gasoline engine. It is worth noting that this efficiency increase is realized by the inherent benefits of two distinct changes. First, a smaller displacement engine requires less fuel for idling and minimizes other parasitic losses and, resultantly, generally shifts the higher efficiency regions (typically correlating to a larger relative load) towards the engine operating points. Secondly and most influential, Diesel engine operation with a high compression ratio and low pumping losses intrinsically offers a significant efficiency increase over a similar-sized gasoline engine. A further reason for the simulated fuel economy increase over the stock vehicle results from the transaxle gear ratio differences. The transaxle accompanying the Fiat 1.9L Diesel engine appropriately considers the Diesel engine's large low-end torque and smaller red-line velocity (due to its larger stroke) and thus presents a smaller ratio across all gears. Consequently, for a given output power, the Diesel engine has a tendency to operate at a (typically more-efficient) lower-speed, higher-torque operating point.

It is intuitive that immobilizing the engine at a vehicle idle condition saves fuel that would otherwise be wasted and thus should lead to a higher fuel economy. Over a simulated FUDS cycle, engine on-off operation (without consideration of the energy needed for each start and stop) saves fuel at a rate of approximately 1.5% (Test # 6). Expectantly, the advantage should be more apparent over driving conditions with more frequent vehicle start/stops such as that experienced in congested city driving.
or rush-hour traffic. A more significant increase in fuel economy results from the typical regenerative braking capability of the HEV. Modest regeneration through the EM (as that described for the Deceleration mode) increases fuel economy by 10.6% for the FUDS cycle and 2% for the FHwDS cycle in Test # 7. It is worth noting that, although the benefits through regenerative braking are more apparent for the city driving schedule, the vehicle start-stop process always results in a net energy loss and thus quasi-steady-state driving is typically more advantageous (as evidenced by inspection of this pattern in Table 6.1). Test # 8 adds the true Electric Only mode to the Diesel-electric hybrid vehicle. Again, any effects are most evident in the FUDS cycle with the frequent vehicle acceleration and deceleration situations. Notably, as in Figure 6.13, the Electric Only mode eliminates some engine operation at low speed and medium load but then requires some higher-load operation immediately after engine start. The low-speed Electric Only operation through the EM, along with its regenerative braking contribution, is depicted in Figure 6.14. As further noticed in Table 6.1, the particular addition of the Electric Only mode causes a slight decrease in urban fuel economy by about 0.3 mpgge. Initial intuition leads to the hypothesis that an Electric Only mode is capable of increasing fuel economy by eliminating Diesel engine operation at inefficient, low-load operation. This will be further investigated subsequently.

**Electric Only Mode Tuning**

Several parameters were varied to investigate the benefits, if any, of an electric launch and Electric Only mode that exclusively utilizes the Delphi EV1 electric machine. The vehicle speed threshold reference in Figures 6.15 and 6.16 refers to the $\dot{v}_{veh,i}$ parameter as previously defined as the vehicle speed at which the vehicle transitions
Figure 6.13: Test # 8: Engine Operating Points

Figure 6.14: Test # 8: Tractive Electric Machine Operating Points
into the Engine Start and Normal (or, in this case, Diesel only) mode. Interestingly enough as depicted in Figures 6.15 and 6.16, fuel economy tends to decrease as $\hat{v}_{veh,l}$ is increased to keep the vehicle in the Electric Only mode for a relatively larger proportion of time. The apparent beginnings of an increase in fuel economy as $\hat{v}_{veh,l}$ reaches approximately 10 m/s is misleading: at some moment in these simulations, the HV battery SOC reaches its lower limit and, from this point forward, the control strategy largely avoids the Electric Only mode. Alterations in the high-voltage battery pack, for the most part, proves noninfluential with respect to fuel economy. Increasing the number of battery modules in series to increase the nominal voltage, and thus power potential, of the HV battery pack has a negligible effect. The reason for this may initiate from the observation that, even though the maximum power of the proposed HV battery pack equates to less than a third of that of the Delphi EV1 unit, the difference at low electric machine speeds is less drastic. As in Figure 6.14, the EM is able to realize $\frac{2}{3}$ of its maximum torque curve or better at typical electric launch speeds even while limited by the proposed HV battery pack. In other words, the simulator suggests that, even without a battery power limitation, the EV1 unit may be incapable of providing any fuel economy benefit or successfully powering the hybridized Equinox in an Electric Only mode. The HV battery pack was further modified by adding a second or third string of battery modules in parallel to linearly increase both the maximum power and capacity. With the exception of avoiding the lower SOC limit for a slightly longer period of time, the simulations reach the same conclusions (or lack thereof) as just mentioned previously.
Figure 6.15: Electric Only Mode Investigation: HV Battery Pack Voltage

Figure 6.16: Electric Only Mode Investigation: HV Battery Pack Capacity
Normal Mode Tuning

Activation of the Normal mode fully advances the control strategy to that as described in Chapter 5. As mentioned previously, the $\bar{\eta}_{\text{dis}}$ and $\bar{\eta}_{\text{ch}}$ values act as relative weightings of the energy cost of drawing electrical power from or inducing electrical power into, respectively, the HV battery pack. In this sense, one method of approaching the problem of determining acceptable values (with respect to minimization of fuel consumption) is to simulate numerous combinations for a particular driving schedule. Figure 6.17 (FUDS) and Figure 6.18 (FHDS) illustrate fuel economy results across a grid of the average efficiency calibration parameters.

![FUDS Fuel Economy](image)

**Figure 6.17:** $\bar{\eta}_{\text{ch}}$ and $\bar{\eta}_{\text{dis}}$ Effect on FUDS Fuel Economy
Figure 6.18: $\bar{\eta}_{ch}$ and $\bar{\eta}_{dis}$ Effect on FHDS Fuel Economy

Previous simulation analyzes of other hybrid vehicle systems with an Equivalent Consumption Minimization Strategy approach have resulted in a similar pattern to that noticed in Figures 6.17 and 6.18: namely, across a grid of the calibration parameters, there exists a plateau of seemingly the largest possible fuel economies. Fortunately, these "optimal" plateaus overlap for the FUDS and FHDS cycles and consequently indicate a possible robustness with respect to driving condition or behavior. However, one area of concern arises with the magnitude of the largest fuel economy of those simulated. The control strategy which, among other objectives, is supposed to further increase fuel economy actually results in a small decrease in fuel economy from Test # 8 with sole Diesel operation in its Normal mode.
Figures 6.19 and 6.20 show the final HV battery pack SOC conditions for that same grid of calibration parameters as discussed previously. Expectantly, the larger fuel economies correlate to a final state-of-charge that did not threaten either limit; this confirms the intuition that the overall control strategy works more advantageously with flexibility in movement on the state-of-charge domain. In other words, fuel economy tends to suffer when the control strategy drives the HV battery pack SOC to either one of its limits.

It may actually be realistic that a large, or even noticeable, increase in fuel economy does not result from the ECMS in *Normal* mode operation, but instead the major benefits arise from the inherent results of hybridization (e.g., downsizing of the
engine and regenerative braking). Instead of acting very aggressively, the primary responsibility of the *Normal* mode may alternatively be to simply maintain a sufficient HV battery pack SOC for electric launch, engine starting, and supplemental power deliverance. Still, this is within the capability realm of ECMS by recognizing the opportunity to adaptively tune the average efficiency calibration values as suggested in Chapter 5. Specifically, from inspection of Figures 6.17 - 6.20, it seems like a careful tuning of the $\bar{\eta}_{ch}$ towards zero (starting from some nominal “good” value) may result in a tendency for the HV battery to charge while maintaining a presence on the high fuel economy plateau. If the HV battery state-of-charge hovers too closely to the upper limit, an adjustment of the $\bar{\eta}_{dis}$ value towards one should cause the system to act in such a way as to discharge the battery while, again, remaining on the plateau.
Therefore, it is proposed that the average efficiency parameters vary about a nominal value as predicted in Equations 5.24 and 5.25:

\[ \bar{\eta}_{ch} = \zeta_{ch} \cdot \bar{\eta}_{ch,0} \]  
\[ \bar{\eta}_{dis} = \zeta_{dis} \cdot \bar{\eta}_{dis,0} \]  

where the adaptive factors are functions of the HV battery pack SOC:

\[ \zeta_{ch} = \zeta_{ch} (s_{ht}) \]  
\[ \zeta_{dis} = \zeta_{dis} (s_{st}) \]  

One approach is to define the adaptive factors change according to the HV battery pack SOC in such a fashion as to cause minimal modification from their nominal values as the high-voltage battery pack state-of-charge lingers at intermediate values. Although a variety of functions may achieve this objective, the adaptive factors in the simulations vary with HV battery pack SOC according to a Lyapunov potential function (similar the previous example associated with Equation 5.23) as depicted in Figure 6.21. As the HV battery pack state-of-charge threatens either of its limits (for whatever reason), the adaptive factors work to influence the SOC away from that boundary.

<table>
<thead>
<tr>
<th>Test #</th>
<th>FUDS (mpgge)</th>
<th>FHDS (mpgge)</th>
<th>0-60 mph (s)</th>
<th>Notes</th>
</tr>
</thead>
<tbody>
<tr>
<td>9</td>
<td>28.2</td>
<td>30.4</td>
<td>10.54</td>
<td>Full control strategy</td>
</tr>
</tbody>
</table>

Table 6.2: cX-SIM Simulation Results, Test # 9

Table 6.2 displays simulation results for a full hybridized Equinox with the complete control strategy as developed in Chapter 5 and the adaptive tuning of the average
efficiency values based on HV battery pack SOC as just mentioned previously. As observed from comparison of Test # 9 with Test # 8, the addition of the Engine Start and Normal modes slightly decrease fuel economy on both an urban and highway driving cycle. One reason surely lies in the implementation of the Engine Start mode in which the ISA draws power from the HV battery pack to quickly start the Diesel engine, as opposed to previous simulations in which the engine automatically commenced at a proper operational speed. Although not immediately apparent purely from inspection of the final fuel economy numbers, the Engine Start mode expectantly affects energy usage more on the urban cycle as there is a greater frequency of these situations. A second reason for the lack of improved fuel economy with the implementation of the Normal mode surfaces through inspection of Figures 6.22 and 6.23.
State-of-charge fluctuation over both cycles is very mild and varies relatively minimally (under 25% of the allowable range) considering the time length of both driving schedules. In fact, for the great majority of the time, the nonzero gradients of the HV battery pack SOC correlate to electric launch or regenerative braking situations. This is confirmed through Figures 6.24 and 6.25: the control strategy clearly has a sole preference for Diesel engine power deliverance during Normal mode operation. Contributions from the ISA and EM (mostly) only occur during electric launch, engine start, or regenerative braking situations. One exception transpires immediately after every Diesel engine start when operation is at a relatively high speed, at which point the ISA briefly load shifts the engine. More aggressive driving cycles may result in power requests that necessitate the supplemental power deliverance from the electric machines, but for the FUDS cycle as seen in Figure 6.26, the accelerator pedal position only approaches or experiences a full depression during the Electric Only mode at which point the simulated power electronics can not sufficiently power the vehicle. It should again be noted that the combined efficiency of the EM and inverter is drastically reduced during the electric launch situations at low speeds. Therefore, even with a maximum HV battery pack current draw of an estimated 120 amps, the actual power at the EM rotor shaft is significantly reduced from the approximately 28 kW maximum of electrical power it may have been receiving. However, this is not the case during the acceleration test: the control strategy uses the maximum electrical power in its most efficient path to the road in route to achieving a 0-60 mph acceleration time exceeding that of the stock Equinox and thus satisfying that vehicle technical specification.
Figure 6.22: Test # 9: FUDS: Vehicle Velocity and HV Battery Pack SOC

Figure 6.23: Test # 9: FHDS: Vehicle Velocity and HV Battery Pack SOC
Figure 6.24: Test # 9: FUDS: Powertrain Actuators Torque Output

Figure 6.25: Test # 9: FHDS: Powertrain Actuators Torque Output
Figure 6.26: Test # 9: FUDS: Accelerator Pedal Position, EM Efficiency, and HV Battery Pack Current

6.3 Summary

Validation of and investigation into the vehicle architecture and control strategy occurred primarily through two methods: experimentation and validation. Rapid prototyping successfully validated functionality of the Delphi EV1 electric machine, Delco inverter, proposed high-voltage battery pack, and switchbox on a rolling chassis apparatus. In an attempt to be prepared for future implementation of the components into the actual Equinox, this insurance of functionality is important and necessary. Validation of the system with respect to concept feasibility and practicality was seemingly successful from the rolling chassis experiments. After a modest initial
start to prevent slip on the dynamometer rollers, the system accelerated the simulated Equinox vehicle from about 4 mph to just under 20 mph in about one second. Comparison to the simulated full throttle acceleration of the stock vehicle from Figure 6.10 infers a more-than-sufficient ability to electrically launch the vehicle during normal, more gentle driving. Further, the experiments left a potential for even quicker acceleration by only reaching a 35% torque request: a complete and exhaustive usage of available torque (as limited by the high-voltage battery pack) should result in even brisker accelerations.

However, it should be noted that the chassis dynamometer made no consideration of an effective mass factor to represent the rotational inertia of the drivetrain (significant at high transaxle gear ratios). This negligence may be one of the primary reasons for the digressing conclusion from cX-SIM about the ability of the proposed system to sufficiently power the vehicle in the Electric Only mode. In fact, the simulation consistently yields a full accelerator pedal depression (and indication of lack of power) in the Electric Only mode even during the relatively gentle FUDS cycle. cX-SIM further reveals the low operating efficiency of the tractive electric machine at low speed operation and resulting decrease in fuel economy with Electric Only mode operation. Even at higher electric machine operating velocities (and efficiencies), the developed control strategy indicates a clear preference for using the Diesel engine alone during most driving conditions. However, the supplemental power deliverance from the power electronics is greatly utilized in achieving a 0-60 mph acceleration better than that of the simulated stock Equinox. Further, the electrical powertrain facilitates fuel economy benefits resulting from an engine on-off strategy and regenerative braking. Therefore, for this particular vehicle architecture, ECMS may function
more appropriately by simply keeping the high-voltage battery pack state-of-charge at a high level for electric launch, engine start, or supplemental power deliverance situations (instead of using the electric machines aggressively during Normal mode operation). Effective use of ECMS in this way with average efficiency values dependent on state-of-charge yields a combined fuel economy of 29.2 mpgge: an impressive increase over the simulated stock Equinox of 30.4% but less than the 40% goal as denoted in the vehicle technical specifications.
CHAPTER 7

CONCLUSION

A final conclusion of this thesis proposes a modification of the selected architecture as another path towards meeting the vehicle technical specifications. It should be noted that the decision matrix (as described in Section 2.7) initially concluded with a relatively simple architecture design which, over time, was added upon and modified to result in the current planned configuration. These changes occurred with careful thought and reasoning and as a result of their apparent benefits; however, the original simplicity, which was attractive as pertinent to technical and business risks, faded with each variation. The proposed modified architecture reverts to the original level of simplicity while featuring several benefits (supported by simulation results) over that of its more architecturally complex counterpart. It is worthy to mention that the subsequent suggestions are aimed towards the benefits of the Challenge X student project team and are not necessarily relevant to the automotive industry (although a few may be applicable to both).

The suggested architecture maintains a downsized Diesel engine and accompanying automatic transaxle on the front drivetrain. As before, a relatively large integrated starter/alternator, such as the chosen Honda Accord Hybrid unit, tightly packages between the engine and transaxle. The primary difference in architecture results from
removal of the second electric machine and associated inverter from the rear of the vehicle.

Elimination of the tractive electric machine presents a greater number and magnitude of advantages than disadvantages. For instance, curb weight of the vehicle could potentially reduce by approximately 200 lbs when considering removal of the electric machine, gearbox, half shafts, inverter, associated section of the power electronics cooling loop, electrical wiring and accessories, etc. Control strategy complexity, especially as it relates to ensuring drivability, would be mitigated significantly. Overall system integration would be simplified, and certain small but numerous issues (such as adapting each electric machine inverter to function on the same high-voltage bus and creating compatibility within the CAN network) would be eliminated. In short, the system integration challenges that exist for implementation of the current powersplit hybrid-electric configuration would be reduced in half by removing the tractive electric machine. The significant and nontrivial packaging issues involving the tractive electric machine and its inverter would vanish and thus guarantee the valuable space in the rear to be solely proportioned between the high-voltage battery pack, switchbox, and DC-DC converter. The issue of excessive regenerative braking through the rear drivetrain as it affects vehicle stability would disappear and, although negative power flow through the torque converter and transaxle is less efficient than through a gearbox alone, the integrated starter/alternator could instead be used aggressively to its full regeneration potential without concern. And finally, fuel economy would improve from eliminating the Electric Only mode. The reasons for this have been considerably stressed in this thesis: an electric machine typically has its lowest range of efficiencies at low speed operation and, when further factoring in the efficiency of the

144
high-voltage battery pack and theoretical energy loop to the fuel tank, does not offer an advantage over solely utilizing the Diesel engine. In summary, utilization of the tractive electric machine solely in the Electric Only mode (which decreases fuel economy) and Deceleration mode (in which the ISA could instead be used for regenerative braking) does not seem to reasonably justify the vast complexity of implementation of the unit.

Several noteworthy, though manageable, disadvantages would stem from the elimination of the tractive electric machine from the architecture. The automobile would lose an ability for four-wheel drive; however, this seems not to be a principal concern to the compact sport-utility vehicle consumer as the majority of the current market of this class are not all-wheel drive standard. Secondly, the smooth and quiet vehicle launch through the rear electric transaxle would be lost and consequently reduce consumer acceptability (in that respect) from that of the original proposal. Finally, performance may suffer slightly as the limitation in the power electronics moves from the energy storage device to the power transformation unit. However, it is theorized that the Honda integrated starter/alternator, which claims a mechanical power output of around 15 kW with a 144V electrical supply, can exceed its rated power capacity when supplied with a larger voltage (given compatibility with the inverter). Further, the planned integrated starter/alternator is a permanent magnet machine with a higher efficiency than the AC induction machines under consideration for the rear and would therefore increase the percentage of electrical power delivered at the rotor in mechanical form. Of course, this ISA power would travel through a relatively
inefficient torque converter in route to the road, but the overall effect may be desirable from the torque multiplication that commonly increases the launch performance of automobiles sporting automatic transmissions and torque converters.

The proposed modification further includes an elimination of the Diesel engine start-stop strategy. Experimental data on the planned Fiat 1.9L Diesel engine indicates an extremely low idle mass flow rate of fuel of 0.11 grams per second [5]. Although not proven through experiments or simulations, the energy expended through the integrated starter/alternator to start the high compression ratio Diesel engine, along with transient effects after each engine start, may actually exceed the equivalent energy that would be saved from elimination of engine idle fueling. Of course, this effect depends on the length of time at engine idle and the frequency of engine start situations. An engine start-stop strategy would indeed be more appropriate if the vehicle would operate in a pure electric mode for a significant period of time. However, again, simulation analysis reveals that the Electric Only mode is not pragmatic (due to excessive electric machine inefficiencies at low load) and not feasible (due to inadequate power). If a start-stop strategy is truly desired, an approach can be taken that mimics that as found to be practiced for certain hybrids in the automotive industry: a release of the brake pedal by the driver initiates an engine start (with minimal delay) in anticipation of a positive power request. A further advantage of the proposed elimination of the start-stop strategy is that it would relieve dependency upon functioning electric accessories and instead would facilitate a gradual transition if time and energy permits. The overall supervisory control strategy would continue utilizing the equivalent consumption minimization approach in the typically unconventional fashion as described previously. In general, the Diesel engine would
be responsible for modest power requests while the control strategy would simply maintain the high-voltage battery pack state-of-charge at a high level in preparation for intermittent large power requests.

Arguably, the principal benefit of hybrid-electric vehicles is their ability to spur technological progress and innovation in the automotive industry through the complex engineering challenges that they present. Similarly, the original power-split architecture design offers quite a challenge for a student project team, and, indeed, this ambitiousness was perceived as favorable during the decision making process. However, as an ultimate goal of the Challenge X competition is to satisfy the vehicle technical specifications through a functioning (and somewhat finetuned) automobile, rationalized simplicity should not be perceived as either an easy or bland solution. In fact, no matter what the approach, prior experience in these programs has never demonstrated a lack of tasks to be accomplished or a lack of potential knowledge to be learned. Independent of what path the team ultimately decides upon, the architecture suggestions of this thesis should be of benefit: adaptation of these ideas could, at best, result in an improved architecture, while reasoned and supported rejection of the proposition would, at worst, stimulate thought and analysis.
## APPENDIX A

### Vehicle Architecture Selection

<table>
<thead>
<tr>
<th>Component</th>
<th>Specifications</th>
<th>$\Delta$ lbs</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine</td>
<td>Stock 3.4L V6</td>
<td>-360</td>
</tr>
<tr>
<td>Fuel tank</td>
<td>Stock 17 gallon</td>
<td>-15</td>
</tr>
<tr>
<td>Exhaust components</td>
<td>Stock catalysts &amp; muffler</td>
<td>-60</td>
</tr>
<tr>
<td>Transmission &amp; Transfer case</td>
<td>Stock 5-speed automatic</td>
<td>-190</td>
</tr>
<tr>
<td>Driveshaft</td>
<td>Stock</td>
<td>-25</td>
</tr>
<tr>
<td>Engine</td>
<td>Diesel, ~2.0L</td>
<td>+400</td>
</tr>
<tr>
<td>Fuel cell</td>
<td>8 Gallons</td>
<td>+8</td>
</tr>
<tr>
<td>Fuel</td>
<td>17 to 8 Gallons</td>
<td>-68</td>
</tr>
<tr>
<td>Exhaust components</td>
<td>Catalysts, etc.</td>
<td>+110</td>
</tr>
<tr>
<td>Transmission</td>
<td>6-speed automatic</td>
<td>+200</td>
</tr>
<tr>
<td>Electric machines &amp; coupling</td>
<td>Tractive machine &amp; gearbox</td>
<td>+140</td>
</tr>
<tr>
<td>Inverters (2)</td>
<td>ISA</td>
<td>+60</td>
</tr>
<tr>
<td>HV Battery pack &amp; Switchbox</td>
<td>~340V</td>
<td>+160</td>
</tr>
<tr>
<td>DC-DC converter</td>
<td>HV to 12V</td>
<td>+15</td>
</tr>
<tr>
<td>Systems integration</td>
<td>Controller, CAN network, etc.</td>
<td>+10</td>
</tr>
<tr>
<td>Electronics cooling loop</td>
<td>Radiator, hoses, water, etc.</td>
<td>+25</td>
</tr>
<tr>
<td>Telematics components</td>
<td>Computer, touch screen</td>
<td>+15</td>
</tr>
<tr>
<td><strong>Estimated Hybridization Weight Change</strong></td>
<td></td>
<td>+500</td>
</tr>
</tbody>
</table>

Table A.1: Estimated Weight Addition
<table>
<thead>
<tr>
<th>Component</th>
<th>Specifications</th>
<th>$\sim \Delta$ lbs</th>
</tr>
</thead>
<tbody>
<tr>
<td>Windows</td>
<td>Glass to Lexan</td>
<td>- 50</td>
</tr>
<tr>
<td>interior</td>
<td>Lightweight seats &amp; carpet</td>
<td>- 30</td>
</tr>
<tr>
<td>Brake components</td>
<td>Lightweight calipers and rotors</td>
<td>- 5 to -35</td>
</tr>
<tr>
<td>Hood, fenders, and side panels</td>
<td>Carbon fiber</td>
<td>- 15 to -30</td>
</tr>
<tr>
<td>Sound dampening</td>
<td>Lightweight material</td>
<td>- 15</td>
</tr>
</tbody>
</table>

Table A.2: Opportunities for Weight Reduction
Figure A.1: Decision Matrix
LEGEND

<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>CAN</td>
<td>Controller Area Network</td>
</tr>
<tr>
<td>DC-DC</td>
<td>High-to-Low Voltage DC to DC Converter</td>
</tr>
<tr>
<td>DOC</td>
<td>Diesel Oxidation Catalyst</td>
</tr>
<tr>
<td>DPF</td>
<td>Diesel Particulate Filter</td>
</tr>
<tr>
<td>EGO</td>
<td>Exhaust Gas Oxygen</td>
</tr>
<tr>
<td>ECU</td>
<td>Engine Control Unit</td>
</tr>
<tr>
<td>EM</td>
<td>Electric Machine</td>
</tr>
<tr>
<td>E-STOP</td>
<td>Emergency Disconnect Stop</td>
</tr>
<tr>
<td>HV</td>
<td>High-Voltage</td>
</tr>
<tr>
<td>ISA</td>
<td>Integrated Starter/Alternator</td>
</tr>
<tr>
<td>LNT</td>
<td>Lean NOx Trap</td>
</tr>
<tr>
<td>MAF</td>
<td>Mass Air Flow</td>
</tr>
</tbody>
</table>

- **CAN Components and Buses**
- **Hybrid Powertrain Components**
- **Conventional Powertrain Components**
- **12V Electrical System Devices**
- **Analog or Digital Control Signals**
- **High-Voltage Cables**

Figure A.3: Control System Diagram Legend (for Figure A.2)
APPENDIX B

Experimentation and Simulation

The chassis dynamometer programmed road load coefficients of \( A_{dy} \), \( B_{dy} \), and \( C_{dy} \) are evaluated at a velocity of 50 mph and correspond to resistive forces that are proportional to velocity to the zeroth, first, and second powers, respectively. The calculation method for these coefficients follows that of Equation 3.78 such that:

\[
A_{dy} = m_{veh} \cdot g \cdot C_r \cdot (50 \text{ mph}) \quad \text{(B.1)}
\]

\[
B_{dy} = 0 \quad \text{(B.2)}
\]

\[
C_{dy} = \frac{1}{2} \cdot \rho_{air} \cdot C_d \cdot A_f \cdot (50 \text{ mph})^3 \quad \text{(B.3)}
\]

The programmed vehicle inertia of 4190 lbs equates to the estimated hybridized Equinox mass of 1900 kg.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>( A_{dy} )</td>
<td>8.38</td>
<td>hp</td>
</tr>
<tr>
<td>( B_{dy} )</td>
<td>0</td>
<td>hp</td>
</tr>
<tr>
<td>( C_{dy} )</td>
<td>11.52</td>
<td>hp</td>
</tr>
<tr>
<td>Inertia</td>
<td>4190</td>
<td>lbs</td>
</tr>
</tbody>
</table>

Table B.1: Road Load Parameters for Chassis Dynamometer
<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>$A_f$</td>
<td>2.86</td>
<td>$m^2$</td>
</tr>
<tr>
<td>$C_d$</td>
<td>0.417</td>
<td>-</td>
</tr>
<tr>
<td>$C_r$</td>
<td>0.015</td>
<td>-</td>
</tr>
<tr>
<td>$D_{dr}$</td>
<td>0.00</td>
<td>$s^2/m$</td>
</tr>
<tr>
<td>$F_{bd}$</td>
<td>23.400</td>
<td>$A\cdot s$</td>
</tr>
<tr>
<td>$g$</td>
<td>9.81</td>
<td>$m/s^2$</td>
</tr>
<tr>
<td>$I_{dr}$</td>
<td>0.03</td>
<td>1/m</td>
</tr>
<tr>
<td>$I_{iso}$</td>
<td>100</td>
<td>$N\cdot m/rad$</td>
</tr>
<tr>
<td>$i_{bd,nn}$</td>
<td>-72</td>
<td>$A$</td>
</tr>
<tr>
<td>$i_{bd,nz}$</td>
<td>120</td>
<td>$A$</td>
</tr>
<tr>
<td>$m_{veh}$ Hybridized</td>
<td>1900</td>
<td>$kg$</td>
</tr>
<tr>
<td>$m_{veh}$ Stock</td>
<td>1722</td>
<td>$kg$</td>
</tr>
<tr>
<td>$n_{bd,p}$</td>
<td>1</td>
<td>-</td>
</tr>
<tr>
<td>$n_{bd,s}$</td>
<td>44</td>
<td>-</td>
</tr>
<tr>
<td>$P_{acc,e}$</td>
<td>0</td>
<td>$W$</td>
</tr>
<tr>
<td>$P_{dr}$</td>
<td>0.25</td>
<td>$s/m$</td>
</tr>
<tr>
<td>$P_{xax}$</td>
<td>100</td>
<td>$N\cdot m\cdot s/rad$</td>
</tr>
<tr>
<td>$Q_{thr,d}$</td>
<td>49x10^6</td>
<td>$J/kg$</td>
</tr>
<tr>
<td>$Q_{thr,g}$</td>
<td>44x10^6</td>
<td>$J/kg$</td>
</tr>
<tr>
<td>$r_{veh}$</td>
<td>0.3305</td>
<td>$m$</td>
</tr>
<tr>
<td>$\dot{s}_{bt}$</td>
<td>0.52</td>
<td>-</td>
</tr>
<tr>
<td>$\dot{s}_{bt,1}$</td>
<td>0.50</td>
<td>-</td>
</tr>
<tr>
<td>$\dot{s}_{bt,2}$</td>
<td>0.80</td>
<td>-</td>
</tr>
<tr>
<td>$\dot{v}_{veh,l}$</td>
<td>5</td>
<td>$m/s$</td>
</tr>
<tr>
<td>$\dot{v}_{veh,u}$</td>
<td>5</td>
<td>$m/s$</td>
</tr>
</tbody>
</table>

Table B.2: cX-SIM Simulation Parameter Values
(Continued)
<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\dot{\alpha}$</td>
<td>1</td>
<td>-</td>
</tr>
<tr>
<td>$\eta_{\text{in}}$</td>
<td>0.95</td>
<td>-</td>
</tr>
<tr>
<td>$\eta_{\text{gb}}$</td>
<td>0.90</td>
<td>-</td>
</tr>
<tr>
<td>$\eta_{\text{mec}}$</td>
<td>0.30</td>
<td>-</td>
</tr>
<tr>
<td>$\eta_{\text{hso}}$</td>
<td>0.80</td>
<td>-</td>
</tr>
<tr>
<td>$\eta_{\text{hr}}$</td>
<td>0.90</td>
<td>-</td>
</tr>
<tr>
<td>$\rho_{\text{air}}$</td>
<td>1.29</td>
<td>kg/m³</td>
</tr>
<tr>
<td>$\rho_d$</td>
<td>800</td>
<td>kg/m³</td>
</tr>
<tr>
<td>$\rho_g$</td>
<td>750</td>
<td>kg/m³</td>
</tr>
<tr>
<td>$\tau_{\text{gb}}$</td>
<td>10.946</td>
<td>-</td>
</tr>
<tr>
<td>$\tau_{\text{tr}, \text{w/ Diesel engine}}$</td>
<td>11.49, 6.56, 4.31, 3.20, 2.38, 1.90</td>
<td>-</td>
</tr>
<tr>
<td>$\tau_{\text{tr}, \text{Stock}}$</td>
<td>12.65, 7.49, 5.19, 3.51, 2.70</td>
<td>-</td>
</tr>
<tr>
<td>$\mu$</td>
<td>0.20</td>
<td>-</td>
</tr>
<tr>
<td>$\zeta_{\text{brk.f}}$</td>
<td>0.60</td>
<td>-</td>
</tr>
<tr>
<td>$\zeta_{\text{brk,r}}$</td>
<td>0.40</td>
<td>-</td>
</tr>
<tr>
<td>$\zeta_{\text{cm}}$</td>
<td>5</td>
<td>-</td>
</tr>
<tr>
<td>$\omega_{\text{inc}}$</td>
<td>850</td>
<td>rad/s</td>
</tr>
</tbody>
</table>

Table B.2: Continued
Figure B.1: Chevy Equinox Stock 3.4L Gasoline Engine Fuel Rate Map
(Data Courtesy of GM [30])

Figure B.2: Chevy Equinox Stock 3.4L Gasoline Engine Efficiency Map
Figure B.3: Chevy Equinox Stock 3.4L Gasoline Engine Brake Power Map

Figure B.4: Fiat 1.9L Diesel Engine Fuel Consumption Rate Map
(Data Courtesy of Fiat [5])
Figure B.5: Fiat 1.9L Diesel Engine Efficiency Map

Figure B.6: Fiat 1.9L Diesel Engine Brake Power Map
Figure B.7: Integrated Starter/Alternator Efficiency Map

*The ISA efficiency map of Figure B.7 was created as an appropriately scaled version of the EM efficiency map of Figure B.11, as experimental data on the Honda unit as proposed for use in the selected HEV architecture was not available.*
Figure B.8: Torque Converter Characteristic Curve

\[ \frac{T_{c1}}{T_{c2}} \]

\[ \frac{\omega_{c1}}{\omega_{c2}} \]

Model Coefficients
\( (\times 10^3) \):

- \( C_1 = 5.7656 \)
- \( C_2 = 0.3107 \)
- \( C_3 = -5.4323 \)
- \( C_4 = 3.4325 \)
- \( C_5 = 2.2210 \)
- \( C_6 = -4.6041 \)
- \( C_7 = -6.7644 \)
- \( C_8 = 32.0084 \)
- \( C_9 = -25.2441 \)

*The torque converter as characterized in Figure B.8 does not correlate to either that of the stock or Fiat unit as the parameter values for both were not available.*
Figure B.9: Torque Converter Efficiency Map

Figure B.10: Transaxle Shifting Schedule
Figure B.11: Delphi EV1 Tractive Electric Machine Efficiency Map  
(Data Courtesy of Delphi [4])

Figure B.12: Panasonic Prismatic Battery Module Characteristics  
(Data Courtesy of [8])
Figure B.13: Test # 9: FUDS: Diesel Engine Operating Points

Figure B.14: Test # 9: FUDS: Integrated Starter/Alternator Operating Points
Figure B.15: Test #9: FUDS: Tractive Electric Machine Operating Points

Figure B.16: Test #9: FHDS: Diesel Engine Operating Points
Figure B.17: Test # 9: FHDS: Integrated Starter/Alternator Operating Points

Figure B.18: Test # 9: FHDS: Tractive Electric Machine Operating Points

165
BIBLIOGRAPHY


[38] M. van Nieuwstadt, and O. Yanakiev. A Diesel lean NO\textsubscript{x} trap model for control strategy validation. Society of Automotive Engineers, paper no. 2004-01-0526.


