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Hybrid Electric Vehicle Torque Split Algorithm For Reduction Of Engine Torque Transients

Derek George

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**HYBRID ELECTRIC VEHICLE TORQUE SPLIT ALGORITHM FOR REDUCTION OF
ENGINE TORQUE TRANSIENTS**

Derek George

**Thesis submitted to the Statler College of Engineering and Mineral Resources
at West Virginia University
in partial fulfillment of the requirements
for the degree of**

**Master of Science
in
Mechanical Engineering**

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Department of Mechanical Engineering

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2018**

**Keywords: Hybrid Electric Vehicle Torque Split Algorithm
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Abstract

HYBRID ELECTRIC VEHICLE TORQUE SPLIT ALGORITHM FOR REDUCTION OF ENGINE TORQUE TRANSIENTS

Derek George

The increased concern over energy efficiency and emissions in recent years has led to the deployment of cleaner alternatives for vehicle powertrains, including electrified vehicles. Electrified vehicles have shown promise in increasing fuel economy as well as reducing emissions. As part of the increased push for electrification, various technologies aiding in the deployment of electrified vehicles have been studied. One such important technology is the hybrid electric vehicle torque-split algorithm. Research on these algorithms has largely focused on improving the ability of a hybrid electric vehicle to reduce emissions and energy consumption. In this thesis the development of a hybrid torque split algorithm that proposes a method for reducing engine torque transients over a base power-loss minimization cost function algorithm is presented and compared for the reduction of engine torque transients. Engine torque transients can increase HC and CO emissions, as well as potentially increase fuel consumption, so the developed algorithms were also compared in terms of fuel economy and emissions. The correlation of engine torque transients to emissions and fuel consumption was also assessed. From model-in-the-loop testing an 8.25% decrease of engine torque transients was found over the base power-loss minimization cost function algorithm. From vehicle-in-the-loop testing a 14.6% reduction of engine torque transients was measured, with a 4.84% reduction approximated to be attributable to the difference in algorithms alone, over the base power-loss minimization cost function algorithm. A moderate positive correlation was shown to exist between CO emissions and the engine torque transients, and a 10.4% reduction in CO was found while testing the algorithm against the base power-loss minimization cost function algorithm.

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Dedication

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Table of Contents	
Acknowledgements	iii
Dedication	v
List of Figures	ix
List of Tables	xii
List of Acronyms	xiv
1 Introduction	1
1.1 Importance of Electrification	1
1.2 Advance Vehicle Technology Competitions and EcoCAR 3	2
1.3 West Virginia University EcoCAR3 Camaro	3
1.4 Hybrid Electric Vehicle Algorithms	4
1.5 Study Objectives	4
1.6 Contributions	5
2 Literature Review	6
2.1 Introduction	6
2.2 Electrification	6
2.3 Overview of Hybrid Algorithms	7
2.3.1 Rules-based Hybrid Algorithms	8
2.3.2 Global Optimization Hybrid Algorithms	9
2.3.3 Real-time Optimization Hybrid Algorithms	10
2.4 Engine Transient Impacts on Emissions and Fuel Economy	13
2.5 Drive Cycle Testing	14
3 Methods	16
3.1 Development Contributions	16
3.2 Model Development	16
3.3 Vehicle Model	18
3.3.1 Vehicle Model Overview	18
3.3.2 Vehicle Glider Model	22
3.3.3 High Voltage Battery Model	24
3.3.4 Electric Motor and Inverter Model	29
3.3.5 Transmission and Torque Converter Model	32
3.3.6 Engine Model	35
3.3.7 Fuel Tank Model	41

3.4 Driver Model	43
3.5 Controller	45
3.5.1 Controller Overview	45
3.5.2 Driver Torque Demand Interpreter	46
3.5.3 Component Torque Limits	46
3.5.4 Gear Selection	47
3.5.5 Torque Split Algorithm	47
3.6 Vehicle Dynamometer and Emissions Testing Setup	54
3.6.1 Testing Facility	54
3.6.2 In-Vehicle Hardware	55
3.7 Results Calculations	56
4 Results and Discussion	59
4.1 Vehicle Model Test Results	59
4.1.1 Cost Function Version 1 Model Test Results	59
4.1.2 Cost Function Version 2 Model Test Results	62
4.1.3 Cost Function Version 3 Model Test Results	65
4.1.4 Cost Function Version Comparison Model Results	71
4.2 Vehicle Dynamometer Test Results	73
4.2.1 Cost Function Version 1 Dynamometer Test Results	74
4.2.2 Cost Function Version 3 Dynamometer Test Results	77
4.2.3 Cost Function Version Comparison Dynamometer Results	82
4.2.4 Dynamometer Test Result Statistical Comparison	85
4.2.5 Model Replication of Vehicle Test Conditions and Analysis	93
4.2.6 Correlation Between Measured Engine Effects and Engine Operation	98
5 Conclusion and Recommendations	101
6 References	103
7 Appendix A	107
7.1 EcoCAR 4-cycle Drive Cycle	107
7.2 UDDS Drive Cycle	107
8 Appendix B	108
8.1 Model Top-Level Component Variables	108
8.2 Vehicle Glider Component Variables	109
8.2.1 Vehicle Glider Top-Level Component Variables	109

8.2.2 Vehicle Glider Wheel Component Variables	110
8.3 High Voltage Battery Component Parameters	110
8.4 Motor and Inverter Component Parameters	111
8.5 Transmission and Torque Converter Component Parameters	111
8.5.1 Transmission Component Parameters	111
8.5.2 Torque Converter Component Parameters	113
8.6 Engine Component Variables	115
8.6.1 Engine Top-Level Component Variables	115
8.6.2 Throttle Body Component Variables	116
8.6.3 Intake Manifold Component Variables	116
8.6.4 Core Engine Component Variables	116
8.7 Fuel Tank Component Variables	116
9 Appendix C	117
9.1 Vehicle Test Results Emissions Traces Cost Function Version 1	117
9.2 Vehicle Test Results Emissions Traces Cost Function Version 3	120

List of Figures

Figure 1: West Virginia University EcoCAR3 Camaro powertrain architecture	3
Figure 2: High-level view of model top level	17
Figure 3: High-level view of model top level block diagram	18
Figure 4: Vehicle model high-level view	19
Figure 5: High-level view of vehicle model top level	20
Figure 6: High-level view of vehicle model top level left portion.....	20
Figure 7: High-level view of vehicle model top level right portion.....	21
Figure 8: High-level view of vehicle model top level block diagram	21
Figure 9: Vehicle glider model and controller	22
Figure 10: Vehicle glider model top level	23
Figure 11: Front wheels model	23
Figure 12: Rear wheels model	24
Figure 13: High voltage battery model, controller and junction box	25
Figure 14: High voltage battery equivalence circuit	26
Figure 15: High voltage battery model high-level view	27
Figure 16: High voltage battery model current and voltage calculations	27
Figure 17: High voltage battery model SOC calculation	28
Figure 18: High voltage battery model resistance and open circuit voltage lookup tables ..	28
Figure 19: High voltage battery model current limits lookup tables	29
Figure 20: Electric motor and inverter model and controller.....	30
Figure 21: Electric motor and inverter model high-level view.....	31
Figure 22: Electric motor and inverter model current calculations	32
Figure 23: Electric motor and inverter model view of dynamics.....	32
Figure 24: Transmission and torque converter model and controller	33
Figure 25: Transmission model left portion	34
Figure 26: Transmission model right portion.....	34
Figure 27: Torque converter model	35
Figure 28: Engine dynamics model high-level	36
Figure 29: Engine mechanical dynamics model and startup logic	37
Figure 30: Engine thermodynamics model high-level.....	37
Figure 31: Throttle body model.....	38
Figure 32: Intake manifold model.....	38
Figure 33: Core engine model high-level.....	39
Figure 34: Engine efficiency section core engine model	40
Figure 35: Engine fuel, torque and emissions look-up tables	41
Figure 36: Fuel tank and fuel tank controller model	42
Figure 37: Fuel tank model.....	42
Figure 38: Driver model high-level view	43
Figure 39: Driving subsystem high-level view	44
Figure 40: Driving subsystem drive cycle timer and error section.....	44
Figure 41: Driving subsystem automatic driver section.....	45
Figure 42: Control algorithm high-level block diagram.....	46
Figure 43: Golden section search minimization flowchart.....	49
Figure 44: Engine torque transient cost diagram.....	53

Figure 45: Engine torque transient feedback scheme 1	53
Figure 46: Engine torque transient feedback scheme 2	54
Figure 47: Horiba® 4WD Vulcan II emission chassis dynamometer.....	55
Figure 48: Cost function Version 1 vehicle speed and SOC plot, model test.....	60
Figure 49: Cost function Version 1 engine torque and transmission gear plot, model test..	61
Figure 50: Cost function Version 1, engine torque, motor torque and APP plot, model test	62
Figure 51: Cost function Version 2 vehicle speed and SOC plot, model test.....	63
Figure 52: Cost function Version 2 engine torque and transmission gear plot, model test..	64
Figure 53: Cost function Version 2, engine torque, motor torque and APP plot, model test	65
Figure 54: Cost function Version 3 vehicle speed and SOC plot, model test.....	67
Figure 55: Cost function Version 3 engine torque and transmission gear plot, model test..	68
Figure 56: Cost function Version 3, engine torque, motor torque and APP plot, model test	69
Figure 57: Cost function comparison, engine torque and motor torque, model test	70
Figure 58: Cost function comparison, APP, model test	70
Figure 59: Cost function Version 1 vehicle speed and SOC plot, dynamometer test.....	74
Figure 60: Cost function Version 1 engine torque and transmission gear plot part 1, dynamometer test	75
Figure 61: Cost function Version 1 engine torque and transmission gear plot part 2, dynamometer test	75
Figure 62: Cost function Version 1, engine torque, motor torque and APP plot, dynamometer test	76
Figure 63: Cost function Version 3 vehicle speed and SOC plot, dynamometer test.....	77
Figure 64: Cost function Version 3 engine torque and transmission gear plot part 1, dynamometer test	78
Figure 65: Cost function Version 3 engine torque and transmission gear plot part 2, dynamometer test	79
Figure 66: Cost function Version 3, engine torque, motor torque and APP plot, dynamometer test	80
Figure 67: Cost function comparison, engine torque and motor torque, dynamometer test	81
Figure 68: Cost function comparison, APP, dynamometer test	81
Figure 69: Drive cycle comparison dynamometer test results, second cycle.....	86
Figure 70: Vehicle speed, APP and BPP comparison 1, dynamometer test	87
Figure 71: Engine torque and motor torque comparison 1, dynamometer test	88
Figure 72: Vehicle speed, APP and BPP comparison 2, dynamometer test	89
Figure 73: Engine torque and motor torque comparison 2, dynamometer test	90
Figure 74: EcoCAR 4-cycle drive cycle.....	107
Figure 75: UDDS "505" portion.....	107
Figure 76: Vehicle test results cost function Version 1 CO rate and vehicle speed.....	117
Figure 77: Vehicle test results cost function Version 1 HC rate and vehicle speed.....	117
Figure 78: Vehicle test results cost function Version 1 HC rate and vehicle speed, zoomed	118
Figure 79: Vehicle test results cost function Version 1 NOx rate and vehicle speed	118

Figure 80: Vehicle test results cost function Version 1 NO _x rate and vehicle speed, zoomed	119
Figure 81: Vehicle test results cost function Version 1 CO ₂ rate and vehicle speed	119
Figure 82: Vehicle test results cost function Version 3 CO rate and vehicle speed.....	120
Figure 83: Vehicle test results cost function Version 3 HC rate and vehicle speed.....	120
Figure 84: Vehicle test results cost function Version 3 HC rate and vehicle speed, zoomed	121
Figure 85: Vehicle test results cost function Version 3 NO _x rate and vehicle speed.....	121
Figure 86: Vehicle test results cost function Version 3 NO _x rate and vehicle speed, zoomed	122
Figure 87: Vehicle test results cost function Version 3 CO ₂ rate and vehicle speed	122

List of Tables

Table 1: Cost function Version 1 test weights	59
Table 2: Cost function Version 1 test charge sustaining calibratable variables	59
Table 3: Cost function Version 2 test weights	63
Table 4: Cost function Version 2 test engine transient cost calibratable variables	63
Table 5: Cost function Version 3 test weights	66
Table 6: Cost function Version 3 test engine transient cost calibratable variables	66
Table 7: Engine torque transient metric model test results	71
Table 8: Energy consumption model test results	72
Table 9: Emissions model test results	72
Table 10: Coefficient of determination model test results	73
Table 11: Engine torque transient metric dynamometer test results both cycles	82
Table 12: Engine torque transient metric dynamometer test results second cycle	82
Table 13: Energy consumption dynamometer test results both cycles	83
Table 14: Energy consumption dynamometer test results second cycle	83
Table 15: Emissions dynamometer test results both cycles	84
Table 16: Emissions dynamometer test results second cycle	84
Table 17: Carbon balance fuel consumption, dynamometer results, second cycle	85
Table 18: Coefficient of determination dynamometer test results both cycles	85
Table 19: Coefficient of determination dynamometer test results, second cycle	86
Table 20: Mean and standard deviation of APP and BPP, dynamometer test second cycle	91
Table 21: APP busyness metric dynamometer test results, second cycle	91
Table 22: Correlation coefficient between engine torque parameters and APP, dynamometer tests second cycle	92
Table 23: APP metrics, model replicated vehicle tests	93
Table 24: Engine torque transient metric, model replicated vehicle tests	94
Table 25: Energy consumption results, model replicated vehicle tests	94
Table 26: Emissions results, model replicated vehicle tests	94
Table 27: APP metrics, model replicated starting SOC only from vehicle tests	95
Table 28: Engine torque transient metric, model replicated starting SOC only from vehicle tests	96
Table 29: Energy consumption results, model replicated starting SOC only from vehicle tests	96
Table 30: Emissions results, model replicated starting SOC only from vehicle tests	96
Table 31: APP metrics, original model conditions	97
Table 32: Percent change between tests	97
Table 33: Correlation coefficient between engine parameters and measured effects, second cycle	99
Table 34: Cross correlation lag between absolute value of derivative of engine torque and measured effects	100
Table 35: Correlation coefficient with delay between absolute value of derivative of engine torque and measured effects	100
Table 36: Flexible shaft (between engine and torque converter subsystems)	108
Table 37: Flexible shaft (between transmission and vehicle glider subsystems)	108
Table 38: Solver configuration	108

Table 39: Differential (8L45 differential)	109
Table 40: Inertia (right/left halfshaft inertia).....	109
Table 41: Inertia (front axle inertia)	109
Table 42: Rolling resistance	109
Table 43: Vehicle body.....	109
Table 44: Double-shoe brake	110
Table 45: Tire (simple)	110
Table 46: HV battery capacity	110
Table 47: Inertia (motor inertia)	111
Table 48: Simple gear (gearbox).....	111
Table 49: Transmission gear values	111
Table 50: Disk friction clutch (neutral)	111
Table 51: Disk friction clutch (parking pin).....	112
Table 52: Flexible shaft (impeller side).....	113
Table 53: Flexible shaft (turbine side)	113
Table 54: Torque converter	113
Table 55: Synchronizer.....	114
Table 56: Inertia (engine inertia).....	115
Table 57: Solver configuration (engine thermodynamics)	115
Table 58: Constant area pneumatic orifice (exhaust system).....	115
Table 59: Gas properties (ambient air properties and states).....	115
Table 60: Variable area pneumatic orifice (throttle throat)	116
Table 61: Transfer fcn (ETC actuator dynamics)	116
Table 62: Constant volume pneumatic chamber (intake manifold volume)	116
Table 63: Transfer fcn (LP filter)	116
Table 64: Fuel tank capacity.....	116

List of Acronyms

A-ECMS	Adaptive Equivalent Consumption Minimization Strategy
AF	Air Fuel
AFDC	Alternative Fuels Data Center
A-PMP	Adaptive Pontryagin's Minimum Principle
APP	Accelerator Pedal Position
AVTC	Advanced Vehicle Technology Competition
BPP	Brake Pedal Position
CO	Carbon Monoxide
CO ₂	Carbon Dioxide
CR	Consumer Reports
DP	Dynamic Programming
ECMS	Equivalent Consumption Minimization Strategy
EF	Equivalence Factor
EM	Electric Motor
EPA	Environmental Protection Agency
GA	Genetic Algorithm
HC	Hydrocarbon
HEV	Hybrid Electric Vehicle
HSC	Hybrid Supervisory Controller
HV	High Voltage
IC	Internal Combustion
IEEE	Institute of Electrical and Electronics Engineers
LHV	Lower Heating Value
NO ₂	Nitrogen Dioxide
NO _x	Nitrogen Oxides
OCV	Open Circuit Voltage
PFCS	Power Follower Control Strategy
PHEV	Plug-in Hybrid Electric Vehicle
PID	Proportional Integral Derivative
PMP	Pontryagin's Minimum Principle
SDP	Stochastic Dynamic Programming
SOC	State of Charge
TSA	Torque Split Algorithm
U.S.	United States
WVU	West Virginia University

1 Introduction

1.1 Importance of Electrification

The development of, and transition to, electrified vehicles is of vital importance to both the citizens of the United States (U.S.) and the world. According to the Alternative Fuels Data Center (AFDC) electrification can benefit the citizens of the U.S. in several ways, including: energy security, fuel economy, cheaper fuel and reduced emissions [1]. According to the AFDC the U.S. imported about 24% of the petroleum it consumed in 2015, of which nearly three-fourths was consumed by transportation. Vehicle electrification can help to reduce this dependence through increased vehicle efficiency resulting in less required petroleum imports. According to Consumer Reports (CR) 53% of American vehicle owners expect better fuel economy in their next car purchase [2]. In addition, CR points out that for 65% of these consumers the reason for American vehicle owners expecting better fuel economy is decreasing spending on fuel. Vehicle electrification results in better fuel economy, which will save consumers money when they refuel their vehicles. According to the Environmental Protection Agency (EPA) the transportation sector contributed nearly 28.5% of greenhouse gas emissions in the U.S. in 2016 [3]. It is estimated that 1.540 billion metric tons of carbon dioxide (CO₂) were produced because of fuel consumption for transportation in 2016 [4]. According to NASA, the effects of the buildup of CO₂ in the atmosphere are expected to result in increased droughts, heat waves, more intense hurricanes and a rise in sea level [5]. Additional emissions of concern aside from CO₂ include criteria air pollutants. The EPA is required to set National Ambient Air Quality Standards for the criteria emissions, which includes carbon monoxide (CO) and nitrogen dioxide (NO₂) [6]. The pollutant NO₂ is harmful to human health and is part of the group of gases called nitrogen oxides (NO_x) [7]. Electrification will result in fewer tailpipe emissions due to increased use of electric drive

systems and less reliance on internal combustion (IC) engines, which produce a significant amount of greenhouse gases as well as criteria emissions. While electrification decreases tailpipe emissions, an increased reliance on the power grid increases emissions from the power grid, however the power grid is generally more energy efficient. Therefore, from a well-to-wheels perspective the electrified vehicle shows promise for reduced emissions production overall. According to an Argonne National Labs report a plug-in hybrid electric vehicle (PHEV) recharging from a potential renewable or nonfossil generation mix reduces greenhouse gas emissions by more than 60% for the power-split PHEV configuration and by more than 90% for the series configuration compared with a baseline gasoline IC engine vehicle [8]. The benefits of electrification are numerous, and the continued improvement of the algorithms that control electrified vehicles is of vital importance to continue the increase in vehicle efficiency and reduction of emissions.

1.2 Advance Vehicle Technology Competitions and EcoCAR 3

The U.S. Department of Energy Advanced Vehicle Technology Competitions (AVTCs) are a program sponsored by industry, government and universities to provide students with an opportunity to build a vehicle that implements future technologies to be applied in the automotive industry [9]. A goal of the AVTC program is to develop a vehicle that is both attractive to the consumer, but also to reduce emissions and improve the fuel economy of vehicles [9]. West Virginia University (WVU) was one of 16 universities granted the opportunity to participate in the EcoCAR 3 competition, in which students converted a 2016 Chevy Camaro into a hybrid electric vehicle over 4 years, following the General Motors vehicle development process.

1.3 West Virginia University EcoCAR3 Camaro

The WVU Camaro is a P3-parallel plug-in hybrid electric vehicle, the parallel designation meaning it had two sources of torque to the wheels, an electric motor (EM) and an internal combustion engine. The P3 designation indicates the EM was positioned between the transmission and the wheels, with power flow to the same axle as the engine. A diagram of the WVU EcoCAR Camaro architecture is shown in Figure 1 below.

Figure 1: West Virginia University EcoCAR3 Camaro powertrain architecture

The engine shown was a 2014 2.4L LEA engine with 136kW of peak power running on E85 fuel, coupled to the engine was the stock 8-speed 8L45 transmission. Supplying the engine was a 7-gallon tank holding E85 fuel. The electrical energy storage system consisted of a high voltage A123 systems 340V Lithium Iron-Phosphate battery pack with 39.2Ah of capacity, and 12.6Kwh of energy storing capability. The EM was a Parker GVM 210-200S, with 118kW of peak output power, and was controlled by a Reinhart Motion Systems inverter controller.

1.4 Hybrid Electric Vehicle Algorithms

Hybrid electric vehicle (HEV) algorithms have been developed to split torque between multiple components in complicated powertrains that often have several degrees of operating freedom. Work has been performed to achieve full optimization given the complete knowledge of operating conditions a priori, such as the drive cycle, however this is unfeasible given the current state of technology. A priori knowledge of the expected driver inputs and operating conditions cannot be predicted with 100% certainty. Improvements have been made to develop real-time optimization that produces a close to optimal global solution in terms of fuel consumption [10]. In addition, work has been done to further improve fuel economy and emissions, as well as the robustness of maintaining high voltage (HV) battery state of charge (SOC) the details of which are discussed in the Literature Review section. Further improvements could be made to stabilize the torque commands that result from such algorithms.

1.5 Study Objectives

It was a high-level goal of this study to produce a hybrid electric vehicle torque-split algorithm via a power-loss minimization cost function and improve the algorithm by reducing the engine torque transients present, in the hopes of potentially reducing greenhouse gas or pollutant emissions and/or reducing fuel consumption. Specifically, the individual objectives included:

- Develop and implement a power-loss minimization control strategy as a base version
- Develop and implement an improved power-loss minimization control strategy that reduces the amount of engine torque transients

- Perform model-in-the-loop and vehicle-in-the-loop testing of the base version of the control strategy and the improved versions
- Assess the success of the improved strategies in reducing the engine torque transients
- Assess the resulting energy consumption and emissions, and their possible correlation with the torque transient

1.6 Contributions

The primary contributions of this study included a methodology for a cost function term, as part of a power-loss minimization cost function, that reduced the engine torque transients present when applied in the model-in-the-loop and vehicle-in-the-loop testing environments. Additionally, results were discussed from model-in-the-loop testing of a method found to be both difficult to calibrate and that resulted in worsening of the engine torque transients.

2 Literature Review

2.1 Introduction

Hybrid electric vehicles have been conceptualized for some time, in 1899 Ferdinand Porsche developed his first HEV with an engine generator providing electricity for four wheel-mounted EMs [11]. Kurz [12] noted that there was a surge in HEV development around 1993 with the formation of the Partnership for a New Generation of Vehicles, and the foundation set out to produce a car with increased fuel efficiency without reduction in performance, safety, or comfort that does not cost more and reduces pollutants. Sciarretta et al. [11] pointed out the rise in interest in hybrid vehicles in the past 15-20 years, from 2007, which is when the article was written. The authors also pointed out the uptick in research activities surrounding HEVs, with less than five papers with the key words “hybrid vehicle” in the Institute of Electrical and Electronics Engineers (IEEE) database in 1991 and more than thirty-five in 2004. When one types the key words “hybrid vehicle” in the IEEE database for 2017, 1,259 papers are listed. Research interest has grown significantly surrounding HEVs and for good reason. Shen et al. [13] pointed out approximations of expected global vehicles to increase from 700 million in 2000 to 2.5 billion in 2050. With the number of cars on the road ever increasing, the importance of vehicle electrification and consumer adoption has subsequently increased. The importance of research in HEV algorithms, has also increased. The body of research in the areas of HEV algorithms has been reviewed and has subsequently been used to guide the research catalogued in this thesis.

2.2 Electrification

Vehicle electrification can be and has been adopted to different degrees. Electrified vehicles range from an HEV with a small electric motor to a fully electric car, referred to as a battery electric vehicle [14]. HEVs, which are a combination of conventional and fully electric

vehicles on the scale of electrification, can be comprised of different configurations, including: series, parallel, and a combination of series and parallel [13]. In addition, amongst these configurations, placement of components relative to one another in the powertrain is important, as this will affect vehicle operation. Shen et al. [13] discussed the importance of the controller design, which controls the energy conversion process by the powertrain. They pointed out three goals that HEV's attempt to satisfy through the control strategies employed including: minimizing fuel consumption, minimizing emissions and good drivability.

2.3 Overview of Hybrid Algorithms

To achieve the optimization of HEVs, which contain two independent energy sources and potentially several degrees of freedom, one must apply a control strategy. Control strategies can be classified into rule-based and optimization-based strategies [15]. Salmasi [15] described a rules-based control strategy to be effective in real-time supervisory control, and generally developed based on heuristics or human intuition without a priori knowledge of a drive cycle. He further classified rules-based control into two subtypes: deterministic and fuzzy, and described the deterministic rules-based strategy as being based on heuristics and generally implemented via lookup tables. It is noted that the use of lookup tables over an online processing algorithm lends itself to real-time controller implementation. Fuzzy rule-based methods were described by Salmasi as using fuzzy logic controls to set the rules. The main advantages are the robustness of fuzzy logic, since it is tolerant of slight imprecision and variations in component states, as well as adaptation, due to tunability.

He classified optimization-based strategies into two subtypes including: global and real-time optimization, and described global optimization as optimization of a cost function representing efficiency and emissions over a drive cycle. He stated that global optimization methods are not applicable in real-time due to their preview nature and computational

complexity. He discussed real-time optimization as applying a cost function in instantaneous optimization and using information regarding electrical energy variations to guarantee electrical self-sustainability. Examples of each of the four categories of hybrid algorithm are discussed in more detail in the following sections.

2.3.1 Rules-based Hybrid Algorithms

An example of a deterministic rules-based strategy is the Power Follower Control Strategy (PFCS), described by Gao et al. for a series HEV [16]. The authors described the PFCS as using the engine generator set as the main power source, while adjusting the output power to meet the driver power demand and using the HV battery as a power equalizer. They stated that the engine generator set is active during almost all driving states except for when low power is required and the HV battery state of charge is above a set upper limit. They concluded that the PFCS control strategy provides stable bus voltage, improving the durability of the pack and that PFCS was better than a thermostatic control in terms of system level efficiency, however the Equivalent Consumption Minimization Strategy (ECMS) application performed better in terms of fuel economy by 9.54% over PFCS. An additional example of a deterministic rules-based strategy was applied by Zhu [17]. Zhu applied the strategy to a 2-mode diesel hybrid-electric vehicle and achieved an improved fuel economy over the base vehicle by 24%. Another example of a deterministic rules-based strategy was employed by Ward on a compound-split diesel hybrid-electric Saturn Vue, in which different rules adjustments were made and the effects reviewed [18]. The author adjusted such things as: up/down shift speed, engine on/off speed and the target SOC, through adjustments the authors achieved a 4.2% combined fuel economy over a pre-production gasoline-powered 2-mode Saturn Vue.

An example of a fuzzy logic-based controller was developed by Lee et al. and applied to a parallel HEV configuration [19]. The authors used a fuzzy logic controller to adapt a normalized ratio of torque command to rated torque at a given speed as a function of both accelerator pedal position (APP) and induction machine speed. They concluded the fuzzy controller had robust properties resulting in insensitivity to various disturbances. They also found about a 20% reduction in NO_x emissions over a diesel engine powertrain alone when the powertrain was fully assisted by the induction machine.

In summary, rules-based algorithms have been applied with some level of success in some scenarios, but improvements over these methods have been made with optimization algorithms. The strength of rules-based algorithms is their simplicity as this lends itself to real-time operation.

2.3.2 Global Optimization Hybrid Algorithms

A few examples of global optimization of HEV algorithms include: Dynamic Programming (DP) [20], Genetic Algorithm (GA) [21], Stochastic Dynamic Programming (SDP) [22], and others. Wang et al. [20] discussed the DP algorithm application procedure on three HEV configurations and applied it to a series-parallel HEV model. The authors stated that the DP technique is based on Bellman's Principle of Optimality, which resorts to solving a single-stage sub-problem starting with the last stage, then the last two stages and so forth. They stated that the DP algorithm is based on a given drive cycle, which is unknown in real cases, and the calculation burden is too heavy to implement in real-time. They also stated that with optimal operation points obtained via DP, an implementable online control strategy could be developed with quasi-optimal performance, and the DP results can serve as a benchmark for other real-time algorithms. Panday et al. [21] applied a GA to a series-parallel configuration HEV. The authors described a GA as a heuristic search algorithm to generate the solution to

optimization and search problems. A GA uses a set of chromosomes to represent individual solutions in a population and the chromosomes of the individual solutions with better fitness are more likely to continue into the next population. The process of producing the next population set is usually continued until some convergence criterion is met. The authors used a GA to adjust parameters in the hybrid controls to maximize fuel economy, such as the engine on criteria. They concluded that the GA is robust in giving the global optima. It is noted that the optimization of the GA was performed with a priori knowledge of the drive cycle. The limitation of the GA is the requirement of a priori knowledge as well as the requisite computations to determine the solution. Lin et al. [22] applied the SDP algorithm to a parallel configuration HEV. The authors stated that the SDP algorithm was used to generate an optimal control policy look-up table that produced the power ratio as a function of power demand and HV battery SOC. They described the SDP process as being based on the Bellman's optimality equation. They used representative driving cycles to construct a Markov model to estimate power demand and solve for the optimal control policy, the power ratio, at all points in the look-up table. The authors stated that the resulting control policy was optimized over all the possible random cycles in an average sense. They concluded that the resulting control policy improved performance in most testing scenarios over the rule-based control strategy trained based on deterministic DP-results.

In summary, determination of the optimal control is achievable if the full driving cycle is known ahead of time by applying a global optimization, however this is not applicable in real time due to computational burden.

2.3.3 Real-time Optimization Hybrid Algorithms

A few examples of the real-time optimization approach are the Equivalent Consumption Minimization Strategy (ECMS) [23], Adaptive ECMS (A-ECMS) [24], and Pontryagin's

Minimum Principle (PMP) [25]. The ECMS algorithm was first introduced in 1999 and is a heuristic method developed to solve the HEV energy management problem [26]. Paganelli et al. [23] applied the ECMS algorithm to a 2000 Chevrolet Suburban as part of the FutureTruck competition. The authors used a cost function that converted the electrical power flow into an equivalent fuel consumption, and the cost function was designed to ultimately minimize the total equivalent fuel consumption. In addition, as part of the ECMS algorithm they applied a HV battery SOC penalty function to the cost function to maintain state of charge around a target value. Also, they used mapping of the expected emissions at various engine operating points as an additional cost to reduce tailpipe emissions during vehicle operation. The authors showed the applicability of the ECMS algorithm in real time by using precomputed 3-D maps. They also showed the ability of the ECMS algorithm to improve the fuel economy and emissions over a conventional vehicle given a proper control strategy, while maintaining the state of charge of a HV battery. Serrao et al. [27] showed that the ECMS algorithm and the application of PMP are both equivalent in that they both reduce the global optimization problem to an instantaneous optimization problem. They also noted the applicability of the ECMS algorithm in real time, because the ECMS does not rely on drive cycle knowledge, and that the ECMS can be used to find a global optimal solution.

Improvements of the ECMS or PMP algorithms include A-ECMS or Adaptive PMP (A-PMP) algorithms [26]. In [26] three subtypes of A-ECMS or A-PMP were described including adaptation based on the following: driving cycle prediction, driving pattern recognition, and adaptation based on SOC feedback.

An example of A-ECMS with driving cycle prediction was performed by Sun et al. in [28] for a series-parallel configuration HEV. The authors used neural networks for velocity

prediction and adapted the equivalence factor (EF) of the ECMS cost function based on this prediction. They also developed a method to reduce the EF oscillations present from the EF estimation algorithm. They showed an improvement of the A-ECMS algorithm of over 3% in terms of fuel economy.

The A-ECMS with driving pattern prediction is described by Onori et al. [26]. The authors discussed that this algorithm has a predefined set of EF based on several cycle topologies, which were updated by a drive cycle pattern recognition algorithm. They discussed this pattern defined EF value being updated periodically and the ECMS algorithm used. The A-ECMS with driving pattern prediction was applied by Wang et al. on an extended range configuration HEV [29]. The authors used a feature vector to classify the driving pattern and determine the appropriate EF to apply the ECMS algorithm. They showed the applicability of this method and an improvement in fuel consumption.

The A-ECMS with adaptation of EF from SOC feedback was described by Onori et al. as having advantages in terms of tunability, as it relies on adaptation based on a single parameter [26]. A version of the A-ECMS algorithm was applied with SOC adaption by Chasse et al to a parallel configuration HEV [30]. The authors developed an A-ECMS with robust SOC control based on SOC variation, that adapted the EF by two calibratable variables in an equation. They showed that a tradeoff exists in the A-ECMS algorithm with SOC control robustness and fuel consumption.

Additionally, A-ECMS algorithms have been improved through application of Vehicle-to-Vehicle (V2V) and Vehicle-to-Infrastructure (V2I) information. Kazemi et al. [31] implemented both a standard A-ECMS algorithm as well as a modified A-ECMS algorithm, which updated the EF given information available via V2V and V2I networks in a parallel

HEV. The authors showed a 2.4% fuel economy improvement over the conventional A-ECMS algorithm. Kazemi et al. [32] further developed adaptations to the A-ECMS for use in a parallel HEV and assessed the prediction window size and the effects it had on the EF and resulting improvements over the A-ECMS. The study concluded that future investigation would be required but found that a prediction window larger than 15s started to degrade performance. Research by Al-Samari [33] was applied to a parallel hybrid electric vehicle using a looking-ahead strategy. The author showed that look-ahead strategy can give a chance for reducing the hybridization equipment sizes with improved fuel economy in comparison to the original equipment sizes without a look-ahead strategy.

In summary, the ECMS algorithm is an algorithm shown to be applicable in real-time, capable of achieving near-optimal fuel consumption results, and maintaining HV battery SOC around a target value. In addition, renditions to the ECMS have attempted to both reduce fuel consumption and tailpipe emissions simultaneously to some degree of success. Further improvements have been made by attempting to use both a priori knowledge, and state feedback. State feedback of only SOC allows for a simpler algorithm that requires tuning but allows for slight fuel economy improvements as well as over just the ECMS alone.

2.4 Engine Transient Impacts on Emissions and Fuel Economy

An elevated level of importance has been placed on reducing both criteria emissions and greenhouse gas emissions by the EPA and others. Understanding the source of tailpipe emissions from the internal combustion (I.C.) engine in a hybrid vehicle is important for determining how best to reduce emissions. Giakoumis [34] made note of engine transient operation being a large source of emitted pollutants from I.C. engines. The author categorized the transient operation as follows: increasing speed, increasing load, and/or (cold or hot) starting. He noted the importance of acknowledging this aspect, as the government has

implemented transient cycle testing as part of new vehicle certification. Pulkrabek [35] discussed that in spark ignition engines the exhaust gases leaving the combustion chamber contain up to 6000 ppm of hydrocarbon components, the equivalent of 1-1.5% of fuel. The author also pointed out that typically the exhaust from an SI engine will be about 0.2% to 5% CO and not only does this an undesirable emission, but it also represents unutilized energy. He noted that CO and HC emissions have a strong tie to the air-fuel (AF) ratio of an engine, and a rich AF ratio results in elevated levels of hydrocarbon (HC) and CO. He stated that this is particularly true in engine startup, when AF mixture is purposefully made very rich, and true to a lesser extent during rapid acceleration under load.

Gray et al. [36] made note of increased transients of an engine in hybrid electric vehicles. Some work has been done to minimize engine transient characteristics in hybrids. Yan et al. [37] developed a model-predictive control strategy for a diesel hybrid vehicle that considered engine transient characteristics and showed potential for improving fuel economy of the vehicle. Kazemi et al. [32] applied the A-ECMS algorithm with additional adaptation based on a prediction horizon to a parallel HEV, with a spark ignition engine. The authors also worked to reduce engine transients using this algorithm, defined as engine on/off events.

In summary, engine transient operation has been shown to increase tailpipe emissions in spark ignition engines, and hybrid algorithms have increased engine transients associated with them. In [37] work was done to reduce engine transients in a diesel hybrid algorithm and in [32] work was done to reduce engine transients in a parallel HEV with a spark ignition engine.

2.5 Drive Cycle Testing

The U.S. EPA has employed emissions standards that must be met for motor vehicles, with which compliance must be shown through mandated test procedures as listed in the Code of

Federal Regulations [38]. The test procedure for a PHEV uses either the derived 5-cycle (city/highway) method or the vehicle specific 5-cycle (city/highway/US06/SC03/Cold temperature test) method [39]. It is common in the literature to see the application of a number of these individual test cycles used either in model-in-the-loop or vehicle-in-the-loop testing. An example of one of the individual cycles from the 5-cycle procedure is in [30], in which the FTP cycle was used. For the EcoCAR3 program a custom test cycle was formulated for vehicle testing, which combines the “505” portion of the UDDS cycle, HWFET cycle, US06 City portion, and US06 Highway portion [40]. The “505” portion of UDDS cycle is the approximately first 500 seconds of the UDDS cycle, as shown in Figure 75 in 7.2 UDDS Drive Cycle. According to Crain [40], this cycle addressed real-world driving conditions on-road without directly having to address A/C use and cold ambient temperatures. The weighting of the 4-cycles for the EcoCAR3 4-cycle test are as follows: 45% US06 Highway portion, 29% 505, 14% US06 City portion and 12% HWFET. The 4-cycle test has been used in [41] for comparison of HEV battery cooling strategy comparisons. The EcoCAR3 4-cycle cycle is detailed in 7.1 EcoCAR 4-cycle Drive Cycle.

In summary, the EPA typically requires that vehicles be tested using the 5-cycle test method. The EcoCAR3 4-cycle has been deemed to represent real-world driving conditions on-road without addressing A/C use and cold ambient temperature and is comprised of some of the same cycles present in the EPA 5-cycle method.

3 Methods

3.1 Development Contributions

Many individuals were involved in developing various contributing components to the WVU EcoCAR vehicle. The model, which consisted of a vehicle model, driver model and controller, was also a shared endeavor. The vehicle model and driver model specifically were a collaborative development effort from various individuals who participated in the EcoCAR3 project over the past several years. While the controller was developed collaboratively as well, aside from the search algorithm and cost function detailed in later sections, which were developed individually for this thesis.

3.2 Model Development

To validate the developed hybrid torque split algorithm a vehicle and driver model was necessary. The vehicle model was developed in the Simulink[®] environment, which is a MathWorks[®] product used to model algorithms and physical systems with block diagrams [42]. The vehicle model used a combination of Simscape[™] components along with component data in the form of look-up tables taken from manufacturer specific data, and system representative equations. Simscape[™], allows for rapid model development with physical connections that directly integrate with block diagrams and other modeling paradigms [43]. The manufacturer data was used to match as closely as possible the vehicle developed by the WVU EcoCAR3 team. While manufacturer data was used to develop the model, the model was not validated to be an exact representation of the vehicle. Additionally, the engine look-up table data that was used did not include the effects of transient operation and were from steady state testing. While the study does review the effects of engine transient operation, these effects were not evaluated from the model testing. The model testing focused on the reduction of the engine torque transients.

In addition, the vehicle model used a forward modeling approach, wherein “vehicle speed is computed as the result of the dynamic simulation and not prescribed a priori” [26]. A high-level view of the top-level of the vehicle model, driver model and controller showing the interfaces between them is shown in Figure 2 below.

Figure 2: High-level view of model top level

The outputs of each of the three subsystems were all fed in to a Bus Creator block and fed back into each of the subsystems after passing through a Memory block. The Bus Creator was used to organize signals, while the Memory block was used to avoid algebraic loop errors. Additionally, a block diagram version of the top-level interface between the driver model, vehicle model and controller are shown below in Figure 3.

Figure 3: High-level view of model top level block diagram

From Figure 3, the driver inputs are fed to the controller, which sends commands to the vehicle model. The vehicle model feeds the vehicle speed back to the driver, which results from the vehicle model dynamics. The details of the individual vehicle model components, as well as the driver model are listed in the following sections.

3.3 Vehicle Model

3.3.1 Vehicle Model Overview

The vehicle model was broken into three sections, which included: a Communication section, an Extended Supervisory Controller section and the Plant. The details of the Communication and Extended Supervisory Controller will not be covered, these were modeled for signal routing purposes to match the developed vehicle. A high-level view of the vehicle model is shown in Figure 4.

Figure 4: Vehicle model high-level view

The vehicle model plant section consisted of the following subsystems: vehicle glider and controller, HV battery and controller, EM and inverter/controller, transmission and controller, torque converter, engine and controller, fuel tank and controller, a HV DC-DC converter, a HV charger, a HV air conditioner, and a HV junction box. The details of the HV air conditioner and HV charger will not be discussed as they had no bearing on the tested vehicle operation in this thesis. The individual subsystems were connected via Simscape™ connections, and between several subsystems were the Flexible Shaft Simscape™ component. Additionally, connected on the top level between the transmission subsystem and a Flexible Shaft component was a Solver Configuration block. A high-level view of the plant section and the connections between the plant subsystems is shown in Figure 5.

Figure 5: High-level view of vehicle model top level

A zoomed in view of the left and right portions of the vehicle model top level view are shown below in Figure 6 and Figure 7.

Figure 6: High-level view of vehicle model top level left portion

Figure 7: High-level view of vehicle model top level right portion

The subsequent values for each of the Simscape™ component blocks are listed in 8.1 Model Top-Level Component Variables. Additionally, a block diagram view of the model plant is shown below in Figure 8.

Figure 8: High-level view of vehicle model top level block diagram

In Figure 8, the various commands are sent from the hybrid supervisory controller (HSC), while the vehicle speed is feedback for the driver and results from the simulation dynamics.

3.3.2 Vehicle Glider Model

The vehicle glider portion of the model consisted of two parts: a controller for signal routing to match the vehicle, and a vehicle dynamics model labeled vehicle glider model. The details of the controller will not be covered. A high-level view of the body and controller are shown in Figure 9.

Figure 9: Vehicle glider model and controller

The vehicle glider portion of the model consists of a front and rear set of wheels, a set of shafts, a rear differential, and a vehicle body. All components in this portion of the model were taken from the Simscape™ portion of the Simulink® library. The wheel sets were modeled using the Tire (Simple), Double-Shoe Brake, Rolling Resistance blocks, Ideal Rotational Motion Sensor, Simulink-PS Converter and PS-Simulink Converter blocks. The front axle and rear half shafts of the vehicle were modeled using Inertia blocks. The rear differential was modeled using the Differential block, and the vehicle body was modeled using the Vehicle Body block. A high-level view of the components and the connections between them is shown in Simulink® in Figure 10.

Figure 10: Vehicle glider model top level

A view of the model for the front and rear wheel Subsystems is shown in Figure 11 and Figure 12.

Figure 11: Front wheels model

Figure 12: Rear wheels model

The subsequent values for each of the Simulink[®] component blocks are listed in 8.2 Vehicle Glider Component Variables. The friction and viscous losses of the drivetrain were ignored as part of the drivetrain model, which reduced the accuracy of the energy consumption. This accuracy reduction likely reduced the expected fuel consumption and emissions predictions, as the vehicle components likely output less power over the cycle to meet the drive trace.

3.3.3 High Voltage Battery Model

The HV battery model consisted of three parts: the HV battery controller labeled as BMS, the junction box and the HV battery dynamics model labeled ESS. The HV battery controller and junction box will not be discussed in detail, because these components were only responsible for tactical control development purposes. A high-level view of the HV battery model, the subsequent controller, the junction box and connections between them are shown in Figure 13.

Figure 13: High voltage battery model, controller and junction box

The HV battery model, titled ESS in Figure 13, consisted of several lookup tables with HV battery parameters given by A123 Systems, a sponsor of the EcoCAR3 competition. The given parameters consisted of internal HV battery resistance and open circuit voltage as a function of both HV battery SOC and temperature. The current drawn from the HV battery is the sum of the inverter/EM and DC-DC converter power draw. The following equation was used to calculate the HV battery SOC using these parameters:

Equation 1

$$SOC_{Curr} = \frac{\int I dt}{Ah} + SOC_{Init}$$

In Equation 1, SOC_{Curr} represents the updated SOC value, I is the current being drawn from the HV battery, Ah is the HV battery capacity, and SOC_{Init} is the initial SOC at the beginning of the simulation. To calculate the HV battery voltage the following equation was applied:

Equation 2

$$V = I * R + OCV$$

In Equation 2, V represents the HV battery voltage, I is the current being drawn from the HV battery, R is the internal HV battery resistance determined from the lookup tables provided, and OCV is the open circuit voltage (OCV). Equation 2 was derived from a simple equivalence circuit shown in Figure 14 below.

Figure 14: High voltage battery equivalence circuit

The temperature was held constant at 40 °C for the purposes of this work, which reduced the accuracy of the estimated energy consumption in the model results. A high-level view of the HV battery model in Simulink® is shown below in Figure 15.

Figure 15: High voltage battery model high-level view

A view of the voltage and current calculation is shown in Figure 16.

Figure 16: High voltage battery model current and voltage calculations

The switch in Figure 16 was used for tactical control development purposes. A view of the HV battery SOC calculations are shown in Figure 17.

Figure 17: High voltage battery model SOC calculation

A view of the HV battery resistance and OCV lookup table subsystem is shown in Figure 18.

Figure 18: High voltage battery model resistance and open circuit voltage lookup tables

The data in the lookup tables in Figure 18 will not be shared, because it is data produced by a non-disclosure agreement with A123 Systems. A view of the HV battery current limits lookup table subsystem is shown in Figure 19.

Figure 19: High voltage battery model current limits lookup tables

The current limits lookup table was used for signaling to the Hybrid Supervisory Controller the current limitations of the HV battery. The data in the lookup tables in Figure 19 will not be shared, because it is covered under a non-disclosure agreement with A123 Systems. The subsequent values for each of the Simulink[®] component blocks that will be shared are listed in 8.3 High Voltage Battery Component Parameters.

3.3.4 Electric Motor and Inverter Model

The EM and inverter model consisted of two parts: a controller and an EM and inverter dynamics model. The controller will not be discussed in detail as it was for tactical control development purposes not covered in this thesis. A high-level view of the EM and inverter model along with the controller are shown in Figure 20.

Figure 20: Electric motor and inverter model and controller

The EM and inverter dynamics model consisted of both manufacturer specific data and some Simscape™ components. The manufacturer specific data consisted of lookup tables indicating the maximum available torque output from the EM and the system efficiency of the EM and inverter. The efficiency lookup table data accounts for rotor iron losses, other rotor losses, stator iron losses, moving losses and I^2R losses [44]. The Simscape™ components used included the following: Ideal Torque Source, Mechanical Rotational References, Inertia, Ideal Rotational Motion Sensor, Simple Gear, Simulink-PS Converter, and PS-Simulink Converter. The lookup table data including the maximum EM torque and the efficiency varied with respect to the speed and torque of the EM in the data provided. The maximum EM torque was used to saturate the torque request from the HSC and this saturated value was sent to the Ideal Torque Source block, which was connected to the driveline via the Simple Gear block. The Simple Gear block modeled the gearbox coupling found in the WVU EcoCAR3 Camaro, and the Inertia block modeled the inertia from the EM found in the Camaro as well.

The following calculations were performed to determine the amount of current drawn from the HV battery:

Equation 3

$$\text{Charge } I = \frac{\tau * \omega}{V} \eta$$

Equation 4

$$\text{Discharge } I = \frac{\tau * \omega}{V} \frac{1}{\eta}$$

In Equation 3 and Equation 4, I represents the current drawn from the HV battery to operate the EM, τ is the torque output by the EM, ω is the EM's angular speed, V is the voltage input to the inverter, and η is the system efficiency. The same efficiency data was used for both charge and discharge. A high-level view of the EM and inverter model in Simulink® are shown in Figure 21.

Figure 21: Electric motor and inverter model high-level view

A view of the EM current calculations is shown in Figure 22.

Figure 22: Electric motor and inverter model current calculations

A view of the dynamics portion of the EM and inverter model are shown in Figure 23.

Figure 23: Electric motor and inverter model view of dynamics

The subsequent values for each of the Simulink[®] component blocks are listed in 8.4 Motor and Inverter Component Parameters.

3.3.5 Transmission and Torque Converter Model

The transmission and torque converter model consisted of three parts: the transmission dynamics model, the torque converter dynamics model and the transmission and torque converter controller. The controller for these components will not be discussed as it was used for tactical control development including PRNDL shifting and signal routing. A high-level

diagram of the three parts of the transmission and torque converter model is presented in Figure 24.

Figure 24: Transmission and torque converter model and controller

The transmission model used a few Simscape™ components, which included a custom gear box given by MathWorks®, two Disk Friction Clutches, a mechanical rotational reference, an Ideal Torque Sensor, and an Ideal Rotational Motion Sensor. The gearbox component received the gear ratio from a Multiport Switch block, which changed the gear ratio according to the received gear request from the HSC. The Disk Friction Clutch blocks were used to model the neutral gear and park gear, which actuated according to the gear request.

A high-level view of the transmission model subsystem in Simulink® is shown in Figure 25 and Figure 26.

Figure 25: Transmission model left portion

Figure 26: Transmission model right portion

The torque converter model consisted of primarily a series of Simscape™ components including the following: a Torque Converter block, a Synchronizer block, Flexible Shaft blocks, Ideal Torque Sensor blocks, Ideal Rotational Motion Sensor blocks, Mechanical Rotational Reference blocks, a Simulink-PS block, and PS-Simulink blocks. The converter

lock-up mechanism results from logic in the controller, which actuates above a set vehicle speed. A view of the torque converter model is shown in Figure 27.

Figure 27: Torque converter model

The subsequent values for each of the Simulink[®] component blocks are listed in 8.5 Transmission and Torque Converter Component Parameters.

3.3.6 Engine Model

The engine model was comprised of two parts: the engine model and the controller. The controller will not be discussed in detail in this report, because it was developed for tactical control purposes and is not relevant to the work presented in this thesis. The engine dynamics model was developed for EcoCAR3 teams to use by MathWorks[®]. It combines Simscape[™] tools with manufacturer data. The engine dynamics model was split into four sections: communication, startup logic, engine mechanical dynamics model, and engine thermodynamic model. A high-level view is shown in Figure 28.

Figure 28: Engine dynamics model high-level

The communication portion will not be discussed in detail, as it was for the purposes of signal routing only. The startup logic and engine mechanical dynamics model portion consisted of various SimscapeTM components including: a Disk Friction Clutch block, an Ideal Torque Source block, an Ideal Rotational Motion Sensor block, a Simulink-PS block, a PS-Simulink block, an Inertia block, and Mechanical Rotational References. The Disk Friction Clutch block along with the subsequent components in the engine startup logic section were used to emulate an engine startup. The components in the engine mechanical dynamics model section modeled the torque from the engine, as well as the engine inertia. The subsequent speed of the engine was found using the Ideal Rotational Motion sensor and fed into the engine thermodynamics model. A view of the engine startup logic and engine mechanical dynamics model sections is shown below in Figure 29.

Figure 29: Engine mechanical dynamics model and startup logic

The engine thermodynamics model was split into three subsystems: the throttle body, the intake manifold and the core engine. A high-level view of the engine thermodynamics model is shown in Figure 30.

Figure 30: Engine thermodynamics model high-level

The individual subsystems in the engine thermodynamics model were connected via Simscape™ connections, and between the throttle body and core engine sections are several Simscape™ components. The Simscape™ components in Figure 30 include the following: a Solver Configuration block, a Constant Area Pneumatic Orifice block, a Pneumatic Atmospheric Reference block and a Gas Properties block.

The throttle body subsystem consisted of a transfer function, a lookup table and a series of Simscape™ components. The Simscape™ components that were in the throttle body model include the following: a Variable Area Pneumatic Orifice block, a Pneumatic Mass & Heat Flow Sensor block, a Simulink-PS block, and PS-Simulink blocks. A view of the components that were used in the throttle body model is shown in Figure 31.

Figure 31: Throttle body model

The intake manifold model consisted of a couple of Simscape™ components including: a Perfect Insulator block and a Constant Volume Pneumatic Chamber block. The connections between these blocks is shown in Figure 32.

Figure 32: Intake manifold model

The core engine model was split into two sections the first being the engine efficiency section and the second being the engine fuel, torque and emissions section. A high-level view of the core engine subsystem is shown in Figure 33.

Figure 33: Core engine model high-level

The engine efficiency section contained volumetric efficiency look-up table data, a few calculations to determine intake port flowrate, a low pass filter and several Simscape™ components. The volumetric efficiency data was formulated from steady state testing and does not account for transient effects. The Simscape™ components included the following: a Pneumatic Pressure & Temperature Sensor block, a Controlled Pneumatic Flow Rate Source block, a Pneumatic Absolute Reference block, a Simulink-PS Converter block and PS-Simulink Converter blocks. A view of the components in the engine efficiency section and how they were connected is shown in Figure 34.

Figure 34: Engine efficiency section core engine model

The engine fuel, torque and emissions section included several look-up tables with torque, fuel flow, and emissions data. The emissions look-up tables included HC, CO, NO_x and CO₂ data. It is important to note that the fuel and emissions data was taken from steady state engine operation, therefore much of the information regarding the effects of engine transients are lost. Transient engine operation is likely to impact the HC and CO emissions projections, while the CO₂ emissions are likely to be very well correlated with the fuel consumption and are likely more accurate. The switches and product block were used for modeling the engine start-up. The engine fuel, torque and emissions section are shown below in Figure 35.

Figure 35: Engine fuel, torque and emissions look-up tables

The subsequent values for each of the Simulink[®] component blocks are listed in 8.6 Engine Component Variables.

3.3.7 Fuel Tank Model

The fuel tank model consisted of two portions the fuel tank model and the fuel tank controller. The controller will not be discussed in this thesis, as it was only used for tactical level control development purposes. A high-level view of the fuel tank model and the fuel tank controller are shown in Figure 36.

Figure 36: Fuel tank and fuel tank controller model

The fuel tank model was used to evaluate the fuel level of the vehicle. The following equation was used to model the fuel level:

Equation 5

$$FL_{Curr} = \frac{FL_{Init} * V - \int \dot{v} dt}{V}$$

In Equation 5 FL_{Curr} is the current fuel level, $\dot{v}$ is the volumetric flow rate of fuel, V is the total volumetric capacity of the tank and FL_{Init} is the initial fuel level of the tank. A high-level view of the fuel tank model is shown in Figure 37.

Figure 37: Fuel tank model

The subsequent values for each of the Simulink[®] component blocks are listed in 8.7 Fuel Tank Component Variables.

3.4 Driver Model

The driver model consisted of several subsystems including: a startup, shutdown and driving subsystem, along with additional logic to complete the start-up and shut-down procedure matching the vehicle controls. The details of the start-up and shut-down procedures will not be discussed in this thesis, as they are negligible to driving operation. A high-level view of the entire driver subsystem is shown in Figure 38.

Figure 38: Driver model high-level view

In the driving subsystem the APP and brake pedal position (BPP) control of a driver was modeled as a function of the current vehicle speed and the desired vehicle speed. The driving subsystem was split into several sections including: the communication section, the cycle

timer section, the drive cycle error section, and the automatic driver section. A high-level view of the driving subsystem is shown in Figure 39.

Figure 39: Driving subsystem high-level view

The cycle timer section was used to determine how far in time the simulation moved. The cycle time was used by the drive cycle error section to determine the desired vehicle speed from a look-up table containing the drive cycle. The current vehicle speed was then subtracted from the desired vehicle speed to determine the drive cycle error. A view of the cycle timer and drive cycle error sections is shown Figure 40.

Figure 40: Driving subsystem drive cycle timer and error section

A Proportional Integral Derivative (PID) controller was used to adjust the APP and BPP from the calculated cycle error. Saturation blocks were used to filter out when acceleration and braking were desired, which was determined by the sign of the PID output. If the PID output was positive the saturation to the APP signal allowed for the value to pass, and if the PID output was negative the saturation to the BPP signal was passed then multiplied by negative

one to indicate a positive pedal excursion. A view of the automatic driver section is shown in Figure 41.

Figure 41: Driving subsystem automatic driver section

The values for the automatic driver PID were tuned manually over several drive cycles.

3.5 Controller

3.5.1 Controller Overview

The control algorithm was comprised of two primary portions which were the high-level and low-level controls. The high-level portion of the control algorithm was primarily developed for determining the optimal control values such as the gear selection and the torque split values between the EM and IC engine. The low-level portion of the control algorithm was developed for tactical component-level control. The details of the low-level portion of the control algorithm will not be discussed, as it does not apply for the purposes of this thesis.

A diagram representing the control flow from the high-level portion of the control algorithm is shown in Figure 42.

Figure 42: Control algorithm high-level block diagram

In the following sections the subsystems of the high-level portion of the control algorithm are discussed including: Driver Torque Demand Interpreter, Component Torque Limits, Gear Selection and Torque Split Algorithm (TSA).

3.5.2 Driver Torque Demand Interpreter

The Driver Torque Demand Interpreter subsystem operated by receiving the driver inputs and reinterpreting them as driver demanded wheel torque. The driver inputs included APP and BPP. The APP and BPP signals were used as a function of the vehicle velocity to determine the driver demanded wheel torque from look-up tables. The values in the look-up tables were determined offline and later calibrated online in-vehicle. The calculated driver demanded wheel torque value was supplied to the TSA.

3.5.3 Component Torque Limits

The Component Torque Limits subsystem determined the maximum and minimum torque values from both the EM and IC engine. The torque limits were determined via lookup tables and calculations using both component state feedback and component limit feedback, such

as HV battery current limits. The torque limit values that were calculated were passed to the TSA.

3.5.4 Gear Selection

The shift schedule consisted of two look-up tables, one of which was the upshift table, while the other was the downshift table. The upshift and downshift tables were functions of vehicle speed and APP, and the output of these tables were processed to determine the current gear command to be sent to the transmission controller to be actuated. The table data for both the upshift and downshift tables were determined offline. The optimal gear for all vehicle speed and APP were determined by evaluating the engine power loss and efficiency at each respective vehicle speed and APP using the following fitness equation:

Equation 6

$$F = W_1 * \eta_E + W_2 * \left(1 - \frac{P_{Loss,IC}}{P_{Loss,ICmax}}\right)$$

In Equation 6 η_E is engine efficiency, $P_{Loss,IC}$ is the engine power loss, $P_{Loss,ICmax}$ is the maximum engine power loss, and W_1 and W_2 are weighting coefficients. The upshift and downshift tables were found with different weighting values to offset them from one another. The work that was done to develop the upshift and downshift tables was performed by Connelly [45].

3.5.5 Torque Split Algorithm

The TSA consisted of several parts including engine start logic, transmission drive gear engagement logic and a search algorithm coupled with a cost function for real-time torque split selection via cost-function minimization.

The engine-start logic and transmission drive gear engagement logic consisted of look-up tables that evaluated the HV battery SOC and the driver demanded wheel torque to determine if an engine start or shutdown was necessary and subsequently transmission drive gear

engagement or disengagement. In the case of an engine start due to the driver demanded wheel torque being above the EM available torque the engine was started and the transmission drive gear engaged to supply the additional torque required. Additionally, if the HV battery SOC fell to a set target SOC, around 35%, then the engine turned on and gear engaged to sustain charge, unless the SOC rose above an upper limit, around 39%, in which case the engine shut off. The values for this portion of the algorithm remained constant for this study.

The search algorithm used was a golden section search algorithm. A golden section search algorithm uses a successive bracketing scheme, on a function, that is based on the golden ratio [46]. The use of the golden ratio in the golden section search algorithm adds efficiency to the search and guarantees successive reduction in search space size at the rate of the golden ratio. The golden section search algorithm was used to minimize a cost function to find the optimal engine torque. The optimal engine torque was found within the engine torque bounds, and subsequently the EM torque was found within the EM torque bounds such that the driver demanded wheel torque was met. A flowchart illustrating the golden section search minimization to find the engine torque is shown below in Figure 43.

Figure 43: Golden section search minimization flowchart

In Figure 43 i is an indexing term for the iterations, the x terms are the probed points, the $f(x)$ terms are the probed points' costs, and Φ is the golden ratio term. The minimum and maximum bounds were determined first and were set as the x_1 and x_4 terms, followed by the x_2 and x_3 terms being determined by the golden ratio. All the x terms were fed through the

cost function to determine a cost and the x_2 and x_3 terms were compared for determining the next set of bounds and the next probing point. The process was continued for a set number of iterations for programming simplicity purposes at 14 iterations. A full description of the cost functions used are presented in the following sections.

3.5.5.1 Cost Function Version 1

The approach for the development of the cost function was to attempt a minimization of power-loss while meeting the driver demanded wheel torque and sustaining charge. The first attempted cost function is shown in the following equation:

Equation 7

$$P_{Loss} = W_1 * P_{Loss,EV} + W_2 * P_{Loss,IC} + W_3 * P_{Loss,CS}$$

In Equation 7, P_{Loss} is the final cost value, $P_{Loss,EV}$ is the power loss associated with the HV battery, EM and inverter, $P_{Loss,IC}$ is the power loss associated with the engine, $P_{Loss,CS}$ is a charge sustaining cost, and $W_{1:3}$ were weights applied to each individual cost. $P_{Loss,CS}$ does not represent actual power loss.

The $P_{Loss,EV}$ cost was determined off-line as a function of EM torque and speed. To do this the HV battery resistance and open-circuit voltage were first set to be constant values, by averaging them in the look-up table data supplied for each. Then, Equation 2, Equation 3, and Equation 4 were used to solve for the current and voltage at each EM torque and speed point evaluated. Finally, the following equation was used:

Equation 8

$$P_{Loss,EV} = |I_{Batt}V_{Batt} - \tau_{Mot} * \omega_{Mot}| + [I_{Batt}^2 * R_{Batt}]$$

In Equation 8, I_{Batt} is the HV battery current, V_{Batt} is the HV battery voltage, τ_{Mot} is the EM torque, ω_{Mot} is the EM speed, and R_{Batt} is the HV battery resistance.

The $P_{Loss,IC}$ was determined off-line as a function of engine torque and speed. This was done using engine data taken from the MathWorks[®] supplied engine model. The efficiency was used as was the engine torque and speed to find the power loss at each of the torque and speed points. These values were then renormalized by subtracting the power loss at the point of maximum efficiency from all points and taking the absolute value of these, according to the following equation:

Equation 9

$$P_{Loss,IC} = |P_{Loss,Map,IC} - P_{Loss,Eff,IC}|$$

In Equation 9, $P_{Loss,Map,IC}$ is the raw power loss calculated and $P_{Loss,Eff,IC}$ is the power loss at the most efficient point of the engine. The purpose of the normalization was to incentivize efficient operation of the engine during over the road charging operation. The normalization incentivizes efficient charging by putting the most efficient engine operating points as the lowest cost in the engine power loss table.

For simplicity, both the $P_{Loss,EV}$ and $P_{Loss,IC}$ tables were then renormalized to the maximum value contained within them, which kept them in the range of zero to one making it easier when adding in other costs. Since both the $P_{Loss,EV}$ and $P_{Loss,IC}$ tables were normalized to a single maximum value contained within them, they still operate on the same playing field and should be able to be weighted equally.

The $P_{Loss,CS}$ term, which does not represent actual power loss, was developed to be a calibratable means of sustaining charge and is shown in the following equation:

Equation 10

$$P_{Loss,CS} = \left((SOC_{Curr} - SOC_{Target}) * K_{Norm} \right)^{K_{Exp}} * \left(-\frac{\tau_{Mot,Pos}}{\tau_{Mot,Max}} \right)$$

In Equation 10, SOC_{Curr} is the current SOC of the HV battery, SOC_{Target} is the target SOC for the HV battery, K_{Norm} is a normalization term to set the upper and lower bounds of HV battery SOC, K_{Exp} is an exponential term to create a region of lesser influence closer to the HV battery target SOC in the form of a sigmoid function, $\tau_{Mot,Pos}$ is the EM torque value being evaluated, and $\tau_{Mot,Max}$ is the maximum possible EM torque.

3.5.5.2 Cost Function Version 2

Through the testing of the cost function shown in Equation 7, it was determined that an additional cost should be added to reduce the engine torque transients produced by Equation 7, the details of this testing are discussed in the Results and Discussion section. The following cost function was developed:

Equation 11

$$P_{Loss} = W_1 * P_{Loss,EV} + W_2 * P_{Loss,IC} + W_3 * P_{Loss,CS} + W_4 * P_{Loss,Trns}$$

All terms in Equation 11 are the same as in Equation 7, except the new terms W_4 and $P_{Loss,Trns}$. In Equation 11, $P_{Loss,Trns}$ is an engine torque transient cost term, and W_4 is a weight applied to the engine torque transient cost term. $P_{Loss,Trns}$ is not a term representing actual power-loss. It was proposed that the engine torque transient cost add cost around the current engine operating point and thus for the $P_{Loss,Trns}$ term the following equation was developed to be calculated online:

Equation 12

$$P_{Loss,Trns} = (Sat(\tau_{Eng,Pos} - \tau_{Fdbk}) * K_{Norm})^2$$

In Equation 12, $\tau_{Eng,Pos}$ is the potential engine operating point being evaluated, τ_{Fdbk} is the engine torque feedback, Sat is a saturation applied within calibratable bounds deemed

K_{Sat} , and K_{Norm} is a calibratable normalization applied. A diagram of the parameters and the impacts they have on the cost is shown in Figure 44.

Figure 44: Engine torque transient cost diagram

3.5.5.3 Cost Function Version 3

A final version of the cost function was determined to be necessary from the testing detailed in Results and Discussion. The change that was made was the value used for the τ_{Fdbk} term value in Equation 12. Originally the value was from the engine torque feedback signal sent from the engine controller as is shown below in Figure 45.

Figure 45: Engine torque transient feedback scheme 1

The value that was used instead was the feedback of the previous command, rather than the feedback of the operating engine torque, as is shown in Figure 46.

Figure 46: Engine torque transient feedback scheme 2

The intent behind this change was to add cost around the current set point rather than add cost around the current engine operating point. Essentially, it was thought that this would allow the cost function to determine if moving to the proposed minimum value was worth the engine torque transient that it would require. The results of these changes are detailed in the Results and Discussion section.

3.6 Vehicle Dynamometer and Emissions Testing Setup

3.6.1 Testing Facility

The vehicle was tested at the Center for Alternative Fuels Engines and Emissions (CAFEE), a non-profit research center that works extensively on emission reduction research [47]. The CAFEE team and facilities have been used for a wide array of research projects involving the Department of Energy and the Department of Transportation. Some such research was performed on the emissions and fuel economy of different powertrain configuration buses for comparison by Wayne et al. in [48], and by Nix et al. in [49]. Another example of research performed at the CAFEE facilities was on particulate matter emissions from heavy duty vehicles and was studied by Wu et al. in [50] and by Littera et al. in [51]. The CAFEE facility and team have been involved in numerous contributing research efforts and were instrumental in the testing for this thesis.

Testing was done in the CAFEE light-duty chassis dynamometer test cell, consisting of the Title 40 CFR, Part 1066-compliant Horiba® 4WD Vulcan II emission chassis dynamometer,

shown in Figure 47, along with a Title 40 CFR, Part 1065 compliant Constant Volume Sampling (CVS) emissions sampling system for vehicles with spark-ignited and compression-ignited engines [52].

Figure 47: Horiba® 4WD Vulcan II emission chassis dynamometer

The Horiba® 4WD Vulcan II emission chassis dynamometer was controlled at CAFEE via software provided by Horiba® that allows for a range of standardized test cycles as well as customized cycles to be implemented. The CAFEE emissions sampling system allows for emissions data collection of all parameters considered for this thesis including: CO₂, CO, NO_x and THC. The CAFEE emissions sampling system complies with Title 40 CFR, Part 86, Part 1065 and Part 1066 [52].

3.6.2 In-Vehicle Hardware

To bring the algorithm into the vehicle, the algorithm was compiled in Simulink® and flashed on an ETAS ES910.3-A prototyping and Interface Module using the ETAS INCA environment. Additionally, CAN data was collected at 1 Hz sampling frequency via a Vector VN1630 log module for post-processing.

3.7 Results Calculations

An analysis of energy consumption was performed from the data collected and required the calculation of the following: electrical energy use, fuel energy use, SOC corrected fuel economy, and gasoline equivalent fuel economy.

The electrical energy use was calculated with Equation 13 below.

Equation 13

$$E_{elec} = \int V * I dt$$

Where, V is the HV battery voltage and I is the HV battery current. The fuel energy use was determined using Equation 14 below.

Equation 14

$$E_{fuel} = \int \dot{v} dt * LHV_{E85}$$

Where, $\dot{v}$ is the volumetric flowrate of the E85 used and LHV_{E85} is the lower heating value (LHV) of E85. The values used for LHV_{E85} was 83,600 BTU/Gal [53]. The SOC corrected fuel economy was found using Equation 15 below.

Equation 15

$$FE_{SOCCorr} = \frac{Dist}{\left(E_{fuel} + \frac{E_{elec}}{0.35}\right) * \frac{1}{LHV_{E85}}}$$

Where, $Dist$ is the distance traveled over the cycle, the 0.35 is an estimated conversion efficiency from fuel to electricity. The 0.35 efficiency value was determined by approximating the energy conversion from fuel to electricity at the most efficient engine operating point by using the engine efficiency, motor and inverter efficiency, and battery I^2R losses. The gasoline equivalent fuel economy was ascertained from Equation 16 below.

Equation 16

$$FE_{SOCCorr,GasEq} = FE_{SOCCorr} \frac{LHV_{Gas}}{LHV_{E85}}$$

Where, LHV_{Gas} is the lower heating value of gasoline, the LHV_{Gas} value used was 115,400 BTU/Gal [53].

Additionally, an analysis of the emissions data was performed to determine the distance specific emissions total for each of the emissions and required Equation 17 below.

Equation 17

$$Em_{Tot} = \frac{\int \dot{Em} dt}{Dist}$$

Where, $\dot{Em}$ is the instantaneous emissions mass rate of the emissions species.

An analysis of the “busyness” of the engine torque command was also performed. The developed metric, engine torque transient metric, required Equation 18 below.

Equation 18

$$\tau_{Busy} = \frac{\sum_{i=1}^{j-1} |\tau_{Eng,i+1} - \tau_{Eng,i}|}{T_{EngOn} * T_S}$$

Where, j is the number of points in the set being evaluated and τ_{Eng} is the engine torque, T_S is the sample period and T_{EngOn} is the amount of time the engine was on. When this equation was applied to the CAN data collected, the τ_{Eng} value was updated on the CAN bus at 12ms intervals, or approximately 83.3 Hz, and was sampled at a 1 Hz sampling rate. Equation 18 was also used to evaluate the “busyness” of the APP signal, and this metric was called the APP busyness metric. When evaluating the APP busyness metric, the APP signal was used rather than engine torque and the engine on time value used was the same.

Analysis was performed on the correlation between different variables using the MATLAB® corrcoef function, which returns a matrix of the Pearson correlation coefficients, a measure

of linear dependence between variables [54]. The Pearson correlation coefficient follows Equation 19 below.

Equation 19

$$\rho(A, B) = \frac{1}{N-1} \sum_{i=1}^N \left(\frac{A_i - \mu_A}{\sigma_A} \right) \left(\frac{B_i - \mu_B}{\sigma_B} \right)$$

Where, σ is the standard deviation and μ is the mean.

Additionally, cross-correlation was used to determine a lag between the signals collected with a maximum correlation, to compensate for lag time associated with delay in system effects and signal measurement. The MATLAB® xcorr function was applied, which measures the similarity between a signal and another shifted signal [55].

An analysis of the match of the drive cycle with the actual cycle driven was performed. The coefficient of determination was used and was calculated with Equation 20 below.

Equation 20

$$R^2 = 1 - \frac{\sum_{i=1}^n (y_i - \hat{y}_i)^2}{\sum_{i=1}^n (y_i - \bar{y})^2}$$

Where, $\hat{y}_i$ are the values of the actual cycle driven, y_i are the values of the desired drive cycle and $\bar{y}$ is the mean value of the desired drive cycle [56].

Calculations were also performed to determine the fuel consumption determined from measured emissions. A carbon balance was performed using the Equation 21 below [57].

Equation 21

$$E_{fuel,CB} = \frac{((0.817 * C_{HC}) + (0.429 * C_{CO}) + (0.273 * C_{CO2}))}{1713} * LHV_{E85}$$

Where, $E_{fuel,CB}$ is the fuel consumption from carbon balance, C_{HC} is the amount of HC emitted in grams, C_{CO} is the amount of CO emitted in grams, and C_{CO2} is the amount of CO₂ emitted in grams.

4 Results and Discussion

4.1 Vehicle Model Test Results

The results from the vehicle model were determined by starting the vehicle model with 35% SOC in the HV battery pack, a full tank of E85 fuel and running the EcoCAR 4-cycle drive cycle. A plot of the EcoCAR 4-cycle drive cycle is shown in 7.1 EcoCAR 4-cycle Drive Cycle. The first set of tests that were performed evaluated the first version of the TSA cost function.

4.1.1 Cost Function Version 1 Model Test Results

The first simulation test performed was on the TSA cost function from Equation 7 along with Equation 10, which are shown below as Equation 22 and Equation 23.

Equation 22

$$P_{Loss} = W_1 * P_{Loss,EV} + W_2 * P_{Loss,IC} + W_3 * P_{Loss,CS}$$

Equation 23

$$P_{Loss,CS} = \left((SOC_{Curr} - SOC_{Target}) * K_{Norm} \right)^{K_{Exp}} * \left(-\frac{\tau_{Mot,Pos}}{\tau_{Mot,Max}} \right)$$

The weights and calibratable values that were used were determined from successive iterations of testing in the model. The power loss weights were kept equal, to keep the power loss on an even playing field. The values from the test of the cost function Version 1 are shown in Table 1 and Table 2.

Table 1: Cost function Version 1 test weights

Weights	Values
W_1	0.5
W_2	0.5
W_3	0.3

Table 2: Cost function Version 1 test charge sustaining calibratable variables

Calibratable Variables	Values
K_{Norm}	0.2
K_{Exp}	3

The resulting drive cycle and HV battery SOC from the model test are shown in Figure 48 below.

Figure 48: Cost function Version 1 vehicle speed and SOC plot, model test

From Figure 48, it can be noted that the HV battery SOC is kept within the bounds of 30% and 40%, and the SOC starts at the target of 35%. The resultant engine torque was also plotted and is shown Figure 49.

Figure 49: Cost function Version 1 engine torque and transmission gear plot, model test

From Figure 49, the presence of engine torque transients is noted. Some of the engine torque transients coincided with the changing of the transmission gear, however some of the engine torque transients occurred while the transmission gear was not changing. Some sections with engine torque transients with engine torque changes greater than 10Nm, while the transmission gear was not changing are highlighted in yellow. A zoomed in view of one of the highlighted regions in Figure 49 is shown below in Figure 50.

Figure 50: Cost function Version 1, engine torque, motor torque and APP plot, model test

From Figure 50, it is noted that a roughly 50Nm change in engine torque occurs but coincides with a subsequent drop in motor torque and the APP, as well. The algorithm is seemingly sensitive to a slight change in APP, which results in a drop in engine torque. The presence of the engine torque transients drove the development of Version 2 of the cost function in Equation 11.

4.1.2 Cost Function Version 2 Model Test Results

The testing of cost function Version 2 was performed using Equation 11 and the engine torque feedback for the τ_{Fdbk} value from Equation 12, as shown in Figure 45. Equation 11 and Equation 12 are shown below as Equation 24 and Equation 25.

Equation 24

$$P_{Loss} = W_1 * P_{Loss,EV} + W_2 * P_{Loss,IC} + W_3 * P_{Loss,CS} + W_4 * P_{Loss,Trns}$$

Equation 25

$$P_{Loss,Trns} = (Sat(\tau_{Eng,Pos} - \tau_{Fdbk}) * K_{Norm})^2$$

The weights and calibratable values used for this test were determined after successive iterations of calibration in model. The values used are list in Table 2 above, and Table 3 and Table 4 below.

Table 3: Cost function Version 2 test weights

Weights	Values
W_1	0.5
W_2	0.5
W_3	0.3
W_4	0.02

Table 4: Cost function Version 2 test engine transient cost calibratable variables

Calibratable Variables	Values
K_{Sat}	120
K_{Norm}	0.083

The resulting drive trace and HV battery SOC are shown below in Figure 51.

Figure 51: Cost function Version 2 vehicle speed and SOC plot, model test

In Figure 51, it is noted that the HV battery SOC was kept within the desired bounds and behaves similarly to that in Figure 48. The engine torque and gear were also plotted and are shown below in Figure 52.

Figure 52: Cost function Version 2 engine torque and transmission gear plot, model test

In Figure 52, the same areas are highlighted as in Figure 49, the engine transients don't seem to be reduced much at all from visual inspection. The zoomed in area from testing of cost function Version 1, shown in Figure 50, is also zoomed in below in Figure 53.

Figure 53: Cost function Version 2, engine torque, motor torque and APP plot, model test

From Figure 53 the drop in engine torque is larger than that from cost function Version 1, as is shown in Figure 50. The engine torque change in Figure 53 is around 100Nm followed by an increase in engine torque around 80 Nm, and later another large drop around 140Nm. The APP and motor torque change as well during the engine torque transients. It appears the cost function Version 2 is more sensitive to changes in the APP.

The inability to mitigate the transients with cost function Version 2 led to the development of changes applied with Version 3 of the cost function.

4.1.3 Cost Function Version 3 Model Test Results

The testing of cost function Version 3 was performed using Equation 11, and using the previous command value sent from the TSA subsystem as τ_{Fdbk} in Equation 12, as shown in Figure 46. Equation 11 and Equation 12 are shown below as Equation 26 and Equation 27.

Equation 26

$$P_{Loss} = W_1 * P_{Loss,EV} + W_2 * P_{Loss,IC} + W_3 * P_{Loss,CS} + W_4 * P_{Loss,Trns}$$

Equation 27

$$P_{Loss,Trns} = (Sat(\tau_{Eng,Pos} - \tau_{Fdbk}) * K_{Norm})^2$$

The weights and calibratable values used during the testing of cost function Version 3 are shown above in Table 2 and below in Table 5 and Table 6, and were determined after successive iterations of testing in the model. The approach for determining the calibratable values and variables involved changing the K_{Norm} term to increase or decrease the rate at which the slope of the parabola cost increases as the torque difference from the τ_{Fdbk} term increases. As the K_{Norm} value changed, the W_4 term was adjusted such that the maximum value of the $P_{Loss,Trns}$ term was around the values for the other terms to keep them on a similar playing field. The K_{Sat} term was also adjusted iteratively, and when this term changed the W_4 term was adjusted for the same reason.

Table 5: Cost function Version 3 test weights

Weights	Values
W_1	0.5
W_2	0.5
W_3	0.3
W_4	0.0005

Table 6: Cost function Version 3 test engine transient cost calibratable variables

Calibratable Variable	Values
K_{Sat}	20
K_{Norm}	0.8

The resultant drive cycle and SOC from testing of cost function Version 3 are presented below in Figure 54.

Figure 54: Cost function Version 3 vehicle speed and SOC plot, model test

In Figure 54, it is noted that the behavior of the SOC results in a higher discharge than that of the previous cost function versions, however the general trend of the SOC is similar. The higher discharge as compared to the previous cost function versions indicates the transient portion of the cost function, actively determining if a change in torque is worth the transient that it comes with. A torque increase from the rest of the cost function to increase the HV battery SOC might not occur due to this added cost. The engine torque and gear were also plotted and is shown below in Figure 55.

Figure 55: Cost function Version 3 engine torque and transmission gear plot, model test

In Figure 55 the highlighted regions are the same as that in both Figure 49 and Figure 52. The engine transients have been seemingly diminished to some degree from visual inspection as compared to both Figure 49 and Figure 52, however a zoomed in region was plotted for comparison. The zoomed in region is the same as Figure 50 and Figure 53 for cost function Version 1 and Version 2 respectively. The zoomed in figure for cost function Version 3 is shown below in Figure 56.

Figure 56: Cost function Version 3, engine torque, motor torque and APP plot, model test

From Figure 56, it is noted that the engine torque is seemingly more transient in this section as compared to the cost function Version 1. The engine torque drops around 50Nm as it did in Figure 50, but also increases later around 30Nm. From Figure 56 cost function Version 3 seems to be more sensitive to changes in the APP than Version 1. To further compare the results shown in Figure 50, Figure 53 and Figure 56, plots combining these results are shown below in Figure 57 and Figure 58.

Figure 57: Cost function comparison, engine torque and motor torque, model test

Figure 58: Cost function comparison, APP, model test

From Figure 57 and Figure 58 it is clearer the difference between the three cost functions at the specific region in time shown. Cost function Version 1 seems to respond the best to the

APP decrease at 677s, while cost function Version 2 seems to perform the worst. Cost function Version 2 produced two large engine torque changes, while cost function Version 1 produces smaller torque changes. While the visual comparison is insightful, further analysis was required to quantitatively determine the amount of transient engine torque operation from each cost function test.

4.1.4 Cost Function Version Comparison Model Results

A metric, shown in Equation 18, for analyzing the amount of torque transients was developed and applied to the engine torque. The “busyness” of the engine torque from each test is shown below in Table 7.

Table 7: Engine torque transient metric model test results

Cost Function Version #	W_1	W_2	W_3	W_4	K_{Sat}	K_{Norm}	Engine Torque Transient Metric
1	0.5	0.5	0.3	N/A	N/A	N/A	6.18
2	0.5	0.5	0.3	0.02	120	0.083	6.89
3	0.5	0.5	0.3	0.005	20	0.8	5.67

From Table 7, the engine transients were reduced from testing of cost function Version 1 to testing of cost function Version 3 by 8.25%. It is also noted that the results of testing of cost function Version 2 indicate that this version of the cost function was a worse performing algorithm in terms of torque transient reduction, with an 11.5% increase from testing of cost function Version 1. Version 2 of the cost function exacerbated the transient torque issue.

The energy consumption values for each of the tests were calculated and are shown below in Table 8.

Table 8: Energy consumption model test results

Cost Function Version #	Electrical Energy Consumption (kwh)	Fuel Energy Consumption (kwh)	Gasoline Equivalent Fuel Economy (MPGge)
1	-0.395	19.0	26.3
2	-0.429	19.0	26.4
3	-0.119	18.5	25.9

The resultant energy consumption shows a very slight increase in fuel economy from testing of cost function Version 1 to testing of cost function Version 2, and a slight decrease in fuel economy for testing of cost function Version 3. The decrease in fuel economy in testing of cost function Version 3 is likely due to the cost function no longer selecting to solely reduce power-loss and sustain charge, instead the cost function was additionally determining if the engine transient required to achieve the goals were worth it. The fuel usage in the model was determined with look-up table data that were developed with an engine operating at steady state, with data that also considers transients more accurately the results may be different.

The resultant emissions values, including CO, CO₂, NO_x, and Total HC (THC) were also determined and are shown below in Table 9.

Table 9: Emissions model test results

Cost Function Version #	CO (g/mi)	NO _x (g/mi)	CO ₂ (g/mi)	THC (g/mi)
1	16.8	6.71	322	1.82
2	16.8	6.75	323	1.82
3	15.9	6.48	313	1.78

From Table 9, the results of testing of cost function Version 1 and 2 are similar, the emissions values are either the same or are close. Testing of cost function Version 3 had the lowest emissions across the board, which is possibly due to different engine torque points selected over the drive cycle. Again, the engine look-up table data was developed with steady state engine operation, so the impacts of transient operation on emissions are not represented

accurately in the results. Additionally, the coefficient of determination for the drive cycles was calculated for each test and the values are shown below in Table 10.

Table 10: Coefficient of determination model test results

Cost Function Version #	R ²
1	0.9997
2	0.9998
3	0.9998

In Table 10, the coefficient of determination from each test was approximately 0.999, indicating the validity of the model tests in terms of the consistency with which the cycle was met via the model driver.

4.2 Vehicle Dynamometer Test Results

To test the possible impact of the reduced engine torque transients on fuel consumption and emissions as well as validate the reduction of engine torque transients, a set of tests were performed in the vehicle-in-the-loop test environment on a dynamometer with emissions data collected. The dynamometer F0, F1 and F2 road load coefficients used were those of the stock 2016 Camaro, which resulted in data that does not match the vehicle driving over the road. This is due to limitations when performing coast downs in preparation for dynamometer testing that compensate for the losses of the dynamometer. The data from these tests was only used, however for cost function comparison purposes. The tests were performed by running the EcoCAR 4-cycle drive cycle, as shown in 7.1 EcoCAR 4-cycle Drive Cycle, twice consecutively. The first cycle acted to condition the vehicle and attempt to reduce the impacts of temperature effects on the exhaust catalysts as well as give the driver requisite time to gauge the vehicle response.

4.2.1 Cost Function Version 1 Dynamometer Test Results

The first test performed tested cost function Version 1, with the weights and calibratable values used above in Table 1 and Table 2. The resultant vehicle drive trace and HV battery SOC are shown below in Figure 59.

Figure 59: Cost function Version 1 vehicle speed and SOC plot, dynamometer test

From Figure 59, it is noted that the HV battery SOC behaved similarly to the model results shown in Figure 48, and was maintained between the designated limits.

The engine torque and transmission gear are also shown below in Figure 60 and Figure 61.

Figure 60: Cost function Version 1 engine torque and transmission gear plot part 1, dynamometer test

Figure 61: Cost function Version 1 engine torque and transmission gear plot part 2, dynamometer test

From both Figure 60 and Figure 61, engine torque transients are noted, as in the model data some of the transients coincided with transmission gear changing. The highlighted portions in Figure 60 and Figure 61 show areas of engine torque transients while the transmission gear was not changing. The two cycles show different results, which is likely due to a different starting SOC from the start of one cycle to the next and as a result different times in which the engine is on or off. A zoomed in view of the engine torque, motor torque and the APP, while the transmission gear is not changing, from one portion of this test was plotted and is shown below in Figure 62.

Figure 62: Cost function Version 1, engine torque, motor torque and APP plot, dynamometer test

From Figure 62, it is noted that the engine torque does change several times, the first of which is an approximate 130Nm drop followed by a subsequent increase of the same magnitude. The two drops in engine torque, at times 1437s and 1444s, do coincide with drops in APP and motor torque, indicating that some of the changes in engine torque perceived to be from

the cost function seemingly could be from difference in the way the driver drove the cycle, which could cause the APP differences. Additionally, at around 1467s another dip is shown in the APP, which coincides with a drop in engine torque, further indicating the possible connection between the driver inputs and the resulting engine and motor torque outcomes.

4.2.2 Cost Function Version 3 Dynamometer Test Results

The second test performed tested cost function Version 3, with the weights and calibratable values used above in Table 2, Table 5 and Table 6. Cost function Version 2 was not tested in the vehicle-in-the-loop testing environment, because it did not show as much promise as cost function Version 3, and dynamometer testing time was limited. The resulting vehicle drive trace and HV battery SOC from testing of cost function Version 3 are shown below in Figure 63.

Figure 63: Cost function Version 3 vehicle speed and SOC plot, dynamometer test

From Figure 63, the HV battery SOC was sustained between the designed limits. Also, the SOC trajectory in the second cycle for both dynamometer tests were similar, as shown in

Figure 63 and Figure 59. The SOC difference between Figure 63 and Figure 59 in the first cycle is likely due to a higher starting SOC in the cost function Version 3 test.

The engine torque values and the transmission gear were plotted and are shown below in Figure 64 and Figure 65.

Figure 64: Cost function Version 3 engine torque and transmission gear plot part 1, dynamometer test

Figure 65: Cost function Version 3 engine torque and transmission gear plot part 2, dynamometer test

From visual inspection of Figure 64 and Figure 65 the amount of engine torque transients seems to have decreased relative to Figure 60 and Figure 61. A zoomed in view of the engine torque, motor torque and the APP, while the transmission gear is not changing, from the same portion of the test as in testing of cost function Version 1, was plotted and is shown below in Figure 66.

Figure 66: Cost function Version 3, engine torque, motor torque and APP plot, dynamometer test

From Figure 66, it is noted that a change in engine torque does occur at the start of the plot and coincides with a change in APP, however at time 1435s another change in APP results in a slight increase in engine torque followed by a small decrease in engine torque. The engine torque response during this section is seemingly better than that from cost function Version 1 in Figure 62. An additional plot showing the engine torque and motor torque during this time period from both cost function Version 1 and Version 3 is shown, for further comparison, below in Figure 67 and Figure 68.

Figure 67: Cost function comparison, engine torque and motor torque, dynamometer test

Figure 68: Cost function comparison, APP, dynamometer test

From Figure 67 and Figure 68, it is noted that the APP paths are different, and cost function Version 3 seems to produce less transient engine torque. It is unclear, however to what extent

the difference in the driver inputs affected the resulting engine torque transients detected. Further analysis was performed to quantitatively compare the amount of engine torque transients.

4.2.3 Cost Function Version Comparison Dynamometer Results

To further assess the amount of engine torque transients between cost function Version 1 and cost function Version 3 testing Equation 18 was applied. The results from applying this metric are shown below in Table 11.

Table 11: Engine torque transient metric dynamometer test results both cycles

Cost Function Version #	Engine Torque Transient Metric
1	14.64
3	14.60

From Table 11, it is noted that the engine torque transients have decreased slightly from testing of cost function Version 1 to testing of cost function Version 3 across the entire cycle. It is however noted that at the beginning of testing of cost function Version 1 the HV battery SOC was at 35.5% and ended at 38%, while the HV battery SOC started at 38% for testing of cost function Version 3 and ended at 38%. The second half of the cycle was evaluated alone to compensate for this discrepancy, as well as other possible discrepancies during the conditioning cycles. The HV battery SOC at the halfway point for testing of cost function Version 1 was 39% and 38.5% for testing of cost function Version 3, which are closer in value to one another. The engine transient metric from Equation 18 was applied to the second portion of the cycle alone and the results of this are presented below in Table 12.

Table 12: Engine torque transient metric dynamometer test results second cycle

Cost Function Version #	Engine Torque Transient Metric
1	18.34
3	15.67

From Table 12, it is noted that the engine transients seem to be decreased from testing of cost function Version 1 to testing of cost function Version 3 on the second portion of the cycle, when both were started from a similar SOC value. The percent change from testing of cost function Version 1 to testing of cost function Version 3 on the second portion of the cycle is 14.6%.

The energy consumption from both cycles was also evaluated and is presented below in Table 13.

Table 13: Energy consumption dynamometer test results both cycles

Cost Function Version #	Electrical Energy Consumption (kwh)	Fuel Energy Consumption (kwh)	Gasoline Equivalent Fuel Economy (MPGge)
1	-0.473	28.9	35.3
3	-0.173	29.1	34.0

The energy consumption results show that over the course of the cycle, testing of cost function Version 1 in fact had a higher fuel economy than testing of cost function Version 3. Before testing of cost function Version 1 other tests had been performed using the engine, allowing for a warmer starting temperature for both the engine and catalyst, while before testing of cost function Version 3 the components had a longer period to cool. Along with the temperature is the discrepancy in starting SOC, and other possible discrepancies from the conditioning cycle. To evaluate the two cost functions against each other in similar conditions, the second half of the cycle was evaluated alone. The energy consumption results from the second half of the cycle alone are shown below in Table 14.

Table 14: Energy consumption dynamometer test results second cycle

Cost Function Version #	Electrical Energy Consumption (kwh)	Fuel Energy Consumption (kwh)	Gasoline Equivalent Fuel Economy (MPGge)
1	0.036	13.5	35.6
3	0.000	13.4	36.2

From Table 14, it is noted that a slight increase in fuel economy exists from testing of cost function Version 1 to testing of cost function Version 3, however the percent change is only a 1.7% increase. The calculated difference of fuel consumption was within experimental uncertainty using an expected error from CAN data measurement of 3%, which was determined from [58].

The emissions data collected from the tests were integrated using Equation 17 to determine the values as shown below in Table 15.

Table 15: Emissions dynamometer test results both cycles

Cost Function Version #	CO (g/mi)	NO _x (g/mi)	CO ₂ (g/mi)	THC (g/mi)
1	0.614	0.007	277.3	0.075
3	0.651	0.013	323.8	0.092

From Table 15, it is noted that the emissions across the board were higher in testing of cost function Version 3. However, as mentioned before the differences from the conditioning cycle likely had some effect. The accumulated emissions values were calculated for the second cycle in each test and are shown below in Table 16.

Table 16: Emissions dynamometer test results second cycle

Cost Function Version #	CO (g/mi)	NO _x (g/mi)	CO ₂ (g/mi)	THC (g/mi)
1	0.613	0.013	262.4	0.062
3	0.549	0.002	303.2	0.057

From Table 16, it is noted that the CO₂ from testing of cost function Version 1 to testing of cost function Version 3 increased by 15.6% on the second cycle, despite a fuel economy increase. From testing of cost function Version 1 to testing of cost function Version 3 the CO, NO_x and THC were all reduced on the second cycle, by 10.4%, 84.6% and 8.1% respectively. Plots showing the emissions over time versus the drive cycle for both testing of

cost function Version 1 and testing of cost function Version 3 are listed in 9 Appendix C. It is unclear whether this was the result of decreased engine torque transients.

Additionally, the fuel consumption values were calculated via a carbon balance as shown in Table 17.

Table 17: Carbon balance fuel consumption, dynamometer results, second cycle

Cost Function Version #	Carbon Balance Fuel Energy Consumption (kwh)	CAN Calculated Fuel Energy Consumption (kwh)
1	30.2	13.5
3	35.2	13.4

From Table 17, it is noted that the fuel energy consumption according to the carbon balance method shows a higher difference than from the CAN data collected. It is also noted that the difference in energy consumption between the two methods is very large. The CAN calculated fuel consumption was relied on more in this thesis.

4.2.4 Dynamometer Test Result Statistical Comparison

It is possible that the differences between testing of cost function Version 1 and Version 3 may have resulted in differences in the level of engine transients in the two tests. First, the coefficient of determination of the driven cycle for the two tests was found with respect to the drive cycle and to each other. The coefficient of determination between the two driven cycles should give some indication of the comparability between both tests. The resulting coefficient of determination values for the driven cycles with respect to the drive cycle and to each other are listed below in Table 18.

Table 18: Coefficient of determination dynamometer test results both cycles

Cost Function Version #	R ²
1	0.9880
3	0.9873
1, 3	0.9985

From Table 18, it is shown that the resulting coefficient of determination values were low, but the coefficient of determination between the two tests was at 0.9985, which is like what resulted in the model. A high coefficient of determination between the two driven cycles should indicate an elevated level of comparability. In addition, the second cycle alone was evaluated for the coefficient of determination between the two driven cycles and the results of this are shown below in Table 19.

Table 19: Coefficient of determination dynamometer test results, second cycle

Cost Function Version #	R ²
1, 3	0.9988

From Table 19, again the second cycle for both tests had a 0.9988 coefficient of determination. While 0.9988 coefficient of determination is high, a closer look at the impacts that the difference between the two driven cycles is required. The two cycles were plotted and are shown below in Figure 69.

Figure 69: Drive cycle comparison dynamometer test results, second cycle

From Figure 69, the large acceleration and deceleration regions seem to match well, while some of the areas of close to constant speed show more variation. The large acceleration at around 1700s shows the greatest disparity of the large acceleration or deceleration regions visually. Additionally, the approximately constant speed section at around 1500s shows a large disparity as well. To determine the extent to which differing portions between the driven cycles may have affected the results, a look at the driver inputs in the regions of disparity was analyzed. The vehicle speed, APP and BPP were plotted during the period with the large acceleration at around 1700s and is shown below in Figure 70 as an example of how the driver inputs are affected by the vehicle speed.

Figure 70: Vehicle speed, APP and BPP comparison 1, dynamometer test

From Figure 70, the APP and BPP seem to match up within a few percent, but where there is a larger speed difference the APP difference is quite large at around 1735s. This shows how a seemingly small difference between the two cycles can manifest as a larger difference

in the driver inputs. The extent to which the driver input difference may have affected the resulting outputs from the cost function was also analyzed. A plot of the engine and motor torque from the same time period are shown below in Figure 71.

Figure 71: Engine torque and motor torque comparison 1, dynamometer test

From Figure 71, it is noted that at around 1735s, when the APP difference from testing of cost function Version 1 increased to a larger degree than that from testing of cost function Version 3, the resulting motor torque from testing of cost function Version 1 increases more as well. The cost function Version 1 engine torque subsequently rises quickly to add torque as well, and the shape of the engine torque during this test somewhat matches the APP input. This shows how the difference in driver input in this scenario effected the resulting torque outputs during the test. To further assess the degree to which differences in the drive cycle effected driver inputs another section was chosen for comparison of the vehicle speed, APP and BPP and a subsequent plot is shown below in Figure 72.

Figure 72: Vehicle speed, APP and BPP comparison 2, dynamometer test

From Figure 72, the vehicle speed is mostly very close, with a more notable difference between 1500s and 1510s. During these times the APP is slightly different, but overall the differences are relatively small, as compared to that which is shown in Figure 70. To see the impacts of the APP difference, the engine and motor torque were plotted and are shown below in Figure 73.

Figure 73: Engine torque and motor torque comparison 2, dynamometer test

From Figure 73, at around 1500s a drop in the engine torque from both tests is noted, which matches a decrease in the APP from both tests in Figure 72 to meet the drive trace. At 1500s in Figure 72 the APP seems to drop a larger amount for the cost function Version 1 test but results in a smaller decrease in engine torque at this instance in Figure 73. From Figure 72 following the decrease in APP at 1500s the APP subsequently increases for both tests by differing amounts, with a larger APP increase from the cost function Version 3 test. The resulting effect on the engine torque at this time in Figure 73, shows a larger torque increase from cost function Version 3 as well. This indicates that the differences in the driven cycles, while seemingly small from the coefficient of determination metric, had a greater impact on the driver inputs and subsequently the resulting engine and motor torque.

To quantitatively assess the difference between the APP and BPP signals from testing of cost function Version 1 and testing of cost function Version 3, the mean and standard deviation

for the APP and BPP signals was assessed from the second cycle for both tests. The mean and standard deviation from testing of cost function Version 1 and testing of cost function Version 3 of the APP and BPP signals is shown below in Table 20.

Table 20: Mean and standard deviation of APP and BPP, dynamometer test second cycle

Cost Function Version #	Mean APP	σ APP	Mean BPP	σ BPP
1	11.9	7.48	4.78	10.5
3	11.8	7.00	4.67	10.3

From Table 20 it is noted that the difference in the mean for both the APP and BPP is approximately within 0.1 of each other. The standard deviation shows more deviation from testing of cost function Version 1 to testing of cost function Version 3 of the APP, with a difference of around 0.5, and difference of 0.2 for the BPP standard deviation. From the standard deviation results of the APP, there was more variability of the APP in the testing of cost function Version 1. Additionally, the “busyness” of the APP was also evaluated using Equation 18, but rather than torque values in Equation 18 APP values were used, the engine on time was the same as that for the engine torque transient metric. The resulting values for both testing of cost function Version 1 and testing of cost function Version 3 are listed below in Table 21.

Table 21: APP busyness metric dynamometer test results, second cycle

Cost Function Version #	APP Busyness Metric
1	2.41
3	2.20

From Table 21, the APP busyness reduced by 8.71% from testing of cost function Version 1 to testing of cost function Version 3. It is yet unclear what difference overall the difference in the driver inputs caused in the resulting level of engine transients in the two tests.

To assess the potential effect the difference in APP input may have had on the engine torque the correlation coefficient was determined for the APP and absolute value of derivative of the APP with respect to both the engine torque and the absolute value of the derivative of the engine torque and is shown below in Table 22. The derivative values were determined using a center difference method calculation.

Table 22: Correlation coefficient between engine torque parameters and APP, dynamometer tests second cycle

	Cost Function Version 1 APP	Cost Function Version 1 abs(dAPP/dt)	Cost Function Version 3 APP	Cost Function Version 3 abs(dAPP/dt)
τ	0.315	0.112	0.300	0.109
abs(d τ /dt)	-0.011	0.418	-0.067	0.259

From Table 22, it is noted that in the testing of cost function Version 1 the absolute value of the derivative of the APP has a moderate positive correlation of 0.418 with the absolute value of the derivative of the engine torque, indicating a moderate positive correlation between the engine torque change and the APP change. Also noted is the correlation coefficient between these two values is lower in the testing of cost function Version 3 at 0.259, indicating a weak positive correlation between the engine torque change and APP change in the testing of cost function Version 3. This further indicates that the differences determined in the engine torque transients between the two tests may have been a result of the differences in the APP.

In summary, the driven cycles were shown to have a 0.9985 coefficient of determination between them. From further inspection of the seemingly minor differences in the driven cycles, it was found that the resulting differences in driver input were affected, which resulted in altered engine and motor torque values. From the testing of cost function Version 1 to the testing of cost function Version 3 the engine transient metric value was shown to decrease by 14.6%, while the APP busyness metric showed an 8.71% decrease from the testing of cost

function Version 1 to the testing of cost function Version 3 indicating the possibility that the APP busyness contributed to the engine torque transients as opposed to the cost function alone. A moderate positive correlation was shown to exist between the absolute value of the derivative of engine torque and the absolute value of the derivative of the APP in the testing of cost function Version 1, while a weak positive correlation was shown in the testing of cost function Version 3. A further assessment of the extent to which the differences in the driving conditions affected the results was performed.

4.2.5 Model Replication of Vehicle Test Conditions and Analysis

To further assess the impacts that possible differences in test conditions had on the results of the vehicle tests the test conditions were replicated on the second cycle for the testing of cost function Version 1 and the testing of cost function Version 3 in the model-in-the-loop test environment, in terms of the cycle driven and the starting HV battery SOC.

The first assessment made between the two tests is the impact that the different conditions had on the APP. The mean, standard deviation and busyness were calculated for each test and the resulting values are presented below in Table 23.

Table 23: APP metrics, model replicated vehicle tests

Cost Function Version #	APP Mean	APP σ	APP Busyness Metric
1	12.9	8.57	2.72
3	12.7	8.06	2.05

From Table 23, it is noted that both the standard deviation and busyness metric of APP, show a decrease from the testing of cost function Version 1 to the testing of cost function Version 3 indicating a reduction in APP movement from the testing of Version 1 to the testing of Version 3. The decrease in APP movement over the cycle is in line with what was noted from the vehicle test data in Table 20 and Table 21. The APP busyness showed a 24.6% decrease

from the testing of cost function Version 1 to the testing of cost function Version 3 from the model test replicating the vehicle test conditions, while the APP busyness decreased by 8.71% from the testing of Version 1 to testing of Version 3 in the actual vehicle tests.

The resulting difference in engine torque transients were calculated and is shown below in Table 24.

Table 24: Engine torque transient metric, model replicated vehicle tests

Cost Function Version #	Engine Torque Transient Metric
1	7.26
3	5.50

From Table 24, it is noted that a subsequent 24.2% decrease in engine torque transients was found. The reduction from the actual vehicle test data showed a reduction of 14.6% from the testing of cost function Version 1 to the testing of cost function Version 3 as shown in Table 12.

The resulting energy consumption and distance specific emissions were also found from the model tests emulating the vehicle test conditions. The energy consumption and emissions from these are shown below in Table 25 and Table 26.

Table 25: Energy consumption results, model replicated vehicle tests

Cost Function Version #	Electrical Energy Consumption (kwh)	Fuel Energy Consumption (kwh)	Gasoline Equivalent Fuel Economy (MPGge)
1	0.152	18.0	26.3
3	0.279	18.2	25.5

Table 26: Emissions results, model replicated vehicle tests

Cost Function Version #	CO (g/mi)	NO _x (g/mi)	CO ₂ (g/mi)	THC (g/mi)
1	15.2	6.25	296	1.67
3	14.2	6.24	299	1.64

From Table 25, the fuel economy decreased by 3.04% from the testing of cost function Version 1 to the testing of cost function Version 3 with the vehicle test conditions emulated in the model, as compared to a 1.7% increase from the vehicle test of Version 1 to the vehicle test of Version 3.

To further distinguish the contributing factors in the results for the vehicle test, the model was also run again, with the drive cycle the same for both model tests, but the starting HV battery SOC the same as that from the start of the second cycle in the vehicle tests, in which the starting HV battery SOC was different for testing of cost function Version 1 and testing of cost function Version 3. The APP was again compared from the testing of cost function Version 1 to the testing of cost function Version 3 with only the starting HV battery SOC the same as that from the vehicle test. The resulting APP mean, standard deviation and busyness were calculated and are shown below in Table 27.

Table 27: APP metrics, model replicated starting SOC only from vehicle tests

Cost Function Version #	APP Mean	APP σ	APP Busyness Metric
1	12.1	7.51	1.86
3	12.0	7.57	1.78

From Table 27, it is noted that with the drive cycle the same for both tests the APP differences between tests are smaller than with the drive cycles different. The APP busyness metric shows a 4.30% reduction with only the starting HV battery SOC emulating the vehicle test conditions, while with the starting HV battery SOC and the drive cycles emulating the vehicle test conditions a 24.6% decrease was determined. This indicates that the difference in the driven cycles resulted in a greater difference in how the driver inputs varied between the tests. The engine torque transient metric was also calculated for these tests and is shown below in Table 28.

Table 28: Engine torque transient metric, model replicated starting SOC only from vehicle tests

Cost Function Version #	Engine Torque Transient Metric
1	6.85
3	6.40

From Table 28, a 6.57% decrease was found from the testing of cost function Version 1 to the testing of cost function Version 3 with only the starting HV battery SOC emulating the vehicle test conditions, while a 24.2% decrease occurred from the testing of Version 1 to the testing of Version 3 with both the drive cycles and the starting HV battery SOC emulating the vehicle test conditions. This indicates the degree to which the difference in the APP from the tests ultimately impacts the resulting torque outputs as well. Additionally, the energy consumption and emissions values were calculated for both tests and are shown below in Table 29 and

Table 30.

Table 29: Energy consumption results, model replicated starting SOC only from vehicle tests

Cost Function Version #	Electrical Energy Consumption (kwh)	Fuel Energy Consumption (kwh)	Gasoline Equivalent Fuel Economy (MPGge)
1	0.096	17.1	27.0
3	0.359	16.7	26.5

Table 30: Emissions results, model replicated starting SOC only from vehicle tests

Test #	CO (g/mi)	NO _x (g/mi)	CO ₂ (g/mi)	THC (g/mi)
1	15.0	6.04	291	1.68
2	14.3	5.88	283	1.64

From Table 29, a 1.85% decrease from the testing of cost function Version 1 to the testing of cost function Version 3 was determined with only the starting HV battery SOC emulating the vehicle test conditions, while with both the drive cycles and the starting HV battery SOC emulating the vehicle test conditions a 3.04% decrease was found.

Finally, the vehicle model results with the drive cycle the same and starting HV battery SOC the same was already discussed above in 4.1 Vehicle Model Test Results. The APP metrics were not discussed above and are shown below in Table 31.

Table 31: APP metrics, original model conditions

Cost Function Version #	APP Mean	APP σ	APP Busyness Metric
1	11.3	7.80	1.48
3	11.2	7.85	1.46

From Table 31, it is noted that the APP busyness metric decreased by 1.35%. A full comparison of all the differences between the tests of the cost function Version 1 and Version 3 was developed and is shown below in Table 32.

Table 32: Percent change between tests

Tests	% Change Between Tests						
	APP Busyness Metric	Torque Transient Metric	Fuel Economy	CO	NO _x	CO ₂	THC
Vehicle Tests	-8.71%	-14.60%	1.70%	-10.40%	-84.60%	15.60%	-8.10%
Model Tests – Emulated Drive Cycles and Starting SOC	-24.60%	-24.20%	-3.04%	-6.58%	-0.16%	1.01%	-1.80%
Model Tests – Emulated Starting SOC	-4.30%	-6.57%	-1.85%	-4.67%	-2.65%	-2.75%	-2.38%
Model Tests	-1.35%	-8.25%	-1.52%	-5.36%	-3.43%	-2.80%	-2.20%

The negative values indicate percent decrease, while a positive indicates percent increase from the cost function Version 1 test to the cost function Version 3 test. From Table 32, it is noted that the APP busyness metric percent change tends to scale with the engine torque transient metric percent change, again indicating the connection between the driver inputs and the resulting engine torque transients. From the model simulations, it appears a sizable

portion of the engine torque transient difference is due to APP differences. With the driven cycles from the vehicle test and the starting HV battery SOC emulated in the model testing a much larger change in the APP busyness is shown over the model test with just the starting HV battery SOC emulated, and subsequently the same is true for the engine torque transient metric. This indicates that the driven cycle had a much larger impact on the driver input differences than the starting HV battery SOC, which resulted in a larger difference between the engine torque transient metric. With both the drive cycles the same and the starting HV battery SOC the same the APP busyness difference is even smaller, and subsequently the resulting engine torque transient metric difference is altered as well. Both the difference in the driven cycle and the starting HV battery SOC were shown to have an impact on the APP busyness, as well as the resulting engine torque transients, with the driven cycle showing more impact. Finally, if it is assumed that the differences in the engine torque transient scaled down proportionally with the model results and using the equation as follows:

Equation 28

$$-4.84\% = \frac{-14.60\%}{-24.20\%} * -8.25\%$$

The percent change from the testing of cost function Version 1 to the testing of cost function Version 3 of a fictive vehicle test emulating the same drive cycle and starting HV battery SOC as the model would be a 4.84% reduction from testing of Version 1 to Version 3, indicating a 4.84% reduction possibly attributable to the cost function change alone.

4.2.6 Correlation Between Measured Engine Effects and Engine Operation

To determine what potentially impacted the fuel economy and the emissions the correlation coefficient between the engine speed, engine torque, absolute value of the derivative of engine speed, absolute value of the derivative of engine torque with respect to the fuel rate,

the CO rate, the CO₂ rate, the NO_x rate and the THC rate was determined on the second cycle using the data from both tests. The derivative values were determined using a center difference method calculation. The values for correlation coefficient are shown below in Table 33.

Table 33: Correlation coefficient between engine parameters and measured effects, second cycle

	ω	τ	$\text{abs}(d\omega/dt)$	$\text{abs}(d\tau/dt)$
$\dot{m}_{\text{fuel}}$	0.901	0.943	0.156	0.205
CO	0.499	0.444	0.140	0.168
CO ₂	0.818	0.774	0.034	0.120
NO _x	0.059	0.027	-0.009	-0.003
THC	0.080	0.085	-0.005	0.008

From Table 33, it is noted that the fuel consumption and all emissions show a positive correlation to some degree with torque and speed of the engine. The fuel consumption rate and CO₂ rate show the strongest positive correlations with the torque and speed of the engine, which is expected, while CO is a slightly weaker positive correlation with engine speed and torque. NO_x and THC emissions showed a very weak positive correlation with engine speed and torque. Also, the fuel consumption, CO and CO₂ emissions show a weak positive correlation with the absolute value of the derivative of the torque, which corresponds to transient torque operation. While this does not show causality, it suggests that the impacts on CO, CO₂ and fuel consumption could be affected by the reduction in transient operation.

A further assessment of the potential connection between the absolute value of the derivative of the torque and the fuel consumption and emissions was performed using cross correlation to determine the lag between signals that produced the highest correlation between them. The resulting lag determined from the cross correlation is shown below in Table 34.

Table 34: Cross correlation lag between absolute value of derivative of engine torque and measured effects

	Cost Function Version 1 abs($d\tau/dt$)	Cost Function Version 3 abs($d\tau/dt$)
$\dot{m}_{\text{fuel}}$	-2	-5
CO	-8	-10
CO ₂	-7	-10
NO _x	596	-315
THC	-5	-4

From Table 34, it is noted that a small lag is shown to exist between the fuel consumption, CO, CO₂, and THC with respect to the absolute value of the derivative of the engine torque, suggesting a potentially higher correlation than suspected from just a raw correlation comparison. The lag was likely due to a combination of delay of the system effect as well as measurement delay. The values for NO_x were quite large, which likely indicates random completely uncorrelated data. The correlation coefficient with the delay included was found for the fuel consumption, CO, CO₂ and THC with respect to the absolute value of the derivative of the engine torque and is shown below in Table 35.

Table 35: Correlation coefficient with delay between absolute value of derivative of engine torque and measured effects

	Cost Function Version 1 abs($d\tau/dt$)	Cost Function Version 3 abs($d\tau/dt$)
$\dot{m}_{\text{fuel}}$	0.257	0.213
CO	0.382	0.406
CO ₂	0.249	0.207
THC	0.150	0.185

From Table 35, it is noted that with the delay included from Table 34 a stronger correlation between the absolute value of the derivative of the engine torque, which corresponds to the engine torque transients, is shown with respect to fuel consumption, CO, CO₂ and THC. The correlation for CO is a moderate positive correlation with the delay included, suggesting a possibly more significant connection between the CO produced and the engine torque transients than suspected with the previously calculated correlation.

5 Conclusion and Recommendations

The objectives of this study included the development of a base cost function as part of a torque split algorithm (TSA) and an improved cost function that reduced engine torque transients for a P3 parallel Plug-in Hybrid Electric Vehicle (PHEV). Additionally, the objectives included model-in-the-loop and vehicle-in-the-loop testing of the cost function versions developed. Three versions of the cost function were developed as part of the TSA, and testing was performed. The first version was the base version over which improvement was intended to be made. The second cost function resulted in an 11.5% increase of engine transient operation, while the third cost function version resulted in an 8.25% decrease of engine transient operation from model-in-the-loop testing over the base cost function. To assess the ability of cost function Version 3 to reduce the transient torque operation in vehicle, dynamometer tests were performed comparing the cost function Version 1 and Version 3. Dynamometer testing of cost function Version 3 resulted in a 14.6% reduction of engine torque transients over cost function Version 1, however a substantial portion of this was likely attributable to differences in the way the vehicle was driven as well as different starting conditions. The degree to which the reduction in engine torque transients was alone attributable to the cost function differences was estimated as a 4.84% reduction, by comparing to the model data, however this is only an estimate.

The impacts that the cost function Version 3 had versus cost function Version 1 in terms of fuel economy and emissions was estimated from model data. The fuel economy was estimated to decrease by 1.52%, however the engine model uses lookup tables that ignore the possible effects of engine torque transients. Additionally, the CO, NO_x, CO₂ and THC were estimated to decrease by 5.36%, 3.43%, 2.80% and 2.20% respectively, again the torque transient effects were ignored in the lookup tables for the engine model.

To assess the possible impacts that the engine torque transients may have had on emissions and fuel consumption test data from vehicle dynamometer testing was evaluated. The fuel economy difference with the cost function over the base was a 1.70% increase, which is within the 3% margin of experimental uncertainty from CAN data based fuel economy calculations. The CO, NO_x, and THC distance specific mass rates were reduced by 10.40%, 84.60% and 8.10% respectively from the test of cost function Version 1 to the test of cost function Version 3. The CO₂ distance specific mass rate decreased by 15.60% from the cost function Version 1 test to the cost function Version 3 test. The degree to which the fuel economy and emissions were due to the engine transient operation was evaluated using the correlation coefficient. The highest correlation to changes in engine torque was CO, which showed a moderate positive correlation at 0.382 and 0.486 correlation coefficient for the cost function Version 1 test and cost function Version 3 test respectively.

It is recommended that more dynamometer testing be performed in the future with further varying of the parameters. Also, it is recommended that work be done in the future to control the cycle more closely, with possibly a shorter and simpler cycle. Additionally, that application of similar engine torque transient reduction methods be applied to A-ECMS methods in the future and tested for the implications of this adaptation in terms of fuel economy and emissions. Finally, it is recommended that more work be done to assess the benefits of applying said adaptation, through use of engine dynamometer testing to more accurately determine the impacts and their causes.

6 References

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7 Appendix A

7.1 EcoCAR 4-cycle Drive Cycle

Figure 74: EcoCAR 4-cycle drive cycle

7.2 UDDS Drive Cycle

Figure 75: UDDS "505" portion

8 Appendix B

8.1 Model Top-Level Component Variables

Table 36: Flexible shaft (between engine and torque converter subsystems)

Tab	Variable	Value	Units
Shaft	Stiffness	4.00E+05	N*m/rad
	Inertia	0.0895	kg*m^2
	Damping ratio from internal losses	0.01	
	Number of segments	1	
Viscous Bearing Losses	Viscous friction coefficient at base	[0 0]	N*m/(rad/s)
Initial Conditions	Initial shaft angular deflection	0	rad
	Initial shaft angular velocity	0	rpm

Table 37: Flexible shaft (between transmission and vehicle glider subsystems)

Tab	Variable	Value	Units
Shaft	Stiffness	4.00E+05	N*m/rad
	Inertia	0.01	kg*m^2
	Damping ratio from internal losses	0.01	
	Number of segments	1	
Viscous Bearing Losses	Viscous friction coefficient at base	[0 0]	N*m/(rad/s)
Initial Conditions	Initial shaft angular deflection	0	rad
	Initial shaft angular velocity	0	rpm

Table 38: Solver configuration

Variable	Value
Consistency tolerance	0.001
Sample time	0.005
Nonlinear iterations	3
Delay memory budget [kB]	1024
Filtering time constant	0.001

8.2 Vehicle Glider Component Variables

8.2.1 Vehicle Glider Top-Level Component Variables

Table 39: Differential (8L45 differential)

Tab	Variable	Value	Units
Main	Crown wheel located	To the right of centerline	
	Carrier to driveshaft teeth ratio	2.77	
Meshing Losses	Friction model	No meshing losses – Suitable for HIL simulation	
Viscous Losses	Sun-carrier and driveshaft casing viscous friction coefficients	[0 0]	N*m/(rad/s)
Inertia	Carrier Inertia	0	kg*m ²
	Planet gear inertia	0	kg*m ²

Table 40: Inertia (right/left halfshaft inertia)

Variable	Value	Units
Inertia	0.001	kg*m ²

Table 41: Inertia (front axle inertia)

Variable	Value	Units
Inertia	0.01	kg*m ²

Table 42: Rolling resistance

Variable	Value	Units
Tire pressure	2.5E+5	Pa
Alpha	-0.003	
Beta	0.97	
Coefficient A	0.04	
Coefficient B	0.0002	s/m
Coefficient C	0	s ² /m ²
Velocity threshold	0.001	m/s

Table 43: Vehicle body

Variable	Value	Units
Number of wheels per axle	2	

Horizontal distance from CG to front axle	1.4	m
Horizontal distance from CG to rear axle	1.6	m
CG height above ground	0.5	m
Frontal area	2.154	m ²
Drag coefficient	0.372	
Gravitational acceleration	9.81	m/s ²

8.2.2 Vehicle Glider Wheel Component Variables

Table 44: Double-shoe brake

Tab	Variable	Value	Units
Geometry	Drum radius	150	mm
	Actuator location radius	125	mm
	Pin location radius	125	mm
	Pin location angle	15	deg
	Shoe beginning angle	5	deg
	Shoe span angle	120	deg
Friction	Viscous friction coefficient	0	N*m/(rad/s)
	Contact friction coefficient	0.35	
	Angular velocity threshold	10	rad/s

Table 45: Tire (simple)

Variable	Value	Units
Rolling radius	0.34	m
Compliance	No compliance – Suitable for HIL simulation	
Inertia	No inertia	

8.3 High Voltage Battery Component Parameters

Table 46: HV battery capacity

Variable	Value
Battery Capacity (Plant.ESS.Battery_Pack_Capacity)	39.2

8.4 Motor and Inverter Component Parameters

Table 47: Inertia (motor inertia)

Variable	Value	Units
Inertia	0.0295	kg*m ²

Table 48: Simple gear (gearbox)

Tab	Variable	Value	Units
Main	Follower to base teeth ratio	2.52	
	Output shaft rotates	In same direction as input shaft	
Meshing Losses	Friction model	No meshing losses – Suitable for HIL simulation	
Viscous Losses	Viscous friction coefficients at base and follower	[0 0]	N*m/(rad/s)

8.5 Transmission and Torque Converter Component Parameters

8.5.1 Transmission Component Parameters

Table 49: Transmission gear values

Variable	Value
Transmission Gear 1 (Plant.Trans.L45.gr1)	4.62
Transmission Gear 2 (Plant.Trans.L45.gr2)	3.04
Transmission Gear 3 (Plant.Trans.L45.gr3)	2.07
Transmission Gear 4 (Plant.Trans.L45.gr4)	1.66
Transmission Gear 5 (Plant.Trans.L45.gr5)	1.26
Transmission Gear 6 (Plant.Trans.L45.gr6)	1
Transmission Gear 7 (Plant.Trans.L45.gr7)	0.85
Transmission Gear 8 (Plant.Trans.L45.gr8)	0.66

Table 50: Disk friction clutch (neutral)

Tab	Variable	Value	Units
Geometry	Force active region	Define effective radius	
	Effective torque radius	130	mm
	Number of friction surfaces	4	

	Engagement piston area	0.001	m ²
	Directionality	Bidirectional	
Friction	Friction model	Fixed kinetic friction coefficient	
	Kinetic friction coefficient	0.5	
	Static friction coefficient	0.55	
	De-rating factor	1	
	Clutch velocity tolerance	0.001	rad/s
	Engagement threshold pressure	100	Pa
Viscous Drag	Viscous drag torque coefficient	0	N*m/(rad/s)
Initial Conditions	Initial state	Unlocked	

Table 51: Disk friction clutch (parking pin)

Tab	Variable	Value	Units
Geometry	Force active region	Define effective radius	
	Effective torque radius	130	mm
	Number of friction surfaces	4	
	Engagement piston area	0.001	m ²
	Directionality	Bidirectional	
Friction	Friction model	Fixed kinetic friction coefficient	
	Kinetic friction coefficient	0.3	
	Static friction coefficient	0.35	
	De-rating factor	1	
	Clutch velocity tolerance	0.001	rad/s
	Engagement threshold pressure	100	Pa
Viscous Drag	Viscous drag torque coefficient	0	N*m/(rad/s)
Initial Conditions	Initial state	Unlocked	

8.5.2 Torque Converter Component Parameters

Table 52: Flexible shaft (impeller side)

Tab	Variable	Value	Units
Shaft	Stiffness	2.00E+05	N*m/rad
	Inertia	0.108	kg*m^2
	Damping ratio from internal losses	0.01	
	Number of segments	1	
Viscous Bearing Losses	Viscous friction coefficient at base	[0 0]	N*m/(rad/s)
Initial Conditions	Initial shaft angular deflection	0	rad
	Initial shaft angular velocity	0	rpm

Table 53: Flexible shaft (turbine side)

Tab	Variable	Value	Units
Shaft	Stiffness	2.00E+05	N*m/rad
	Inertia	0.012	kg*m^2
	Damping ratio from internal losses	0.01	
	Number of segments	1	
Viscous Bearing Losses	Viscous friction coefficient at base	[0 0]	N*m/(rad/s)
Initial Conditions	Initial shaft angular deflection	0	rad
	Initial shaft angular velocity	0	rpm

Table 54: Torque converter

Tab	Variable	Value	Units
Torque Characteristics	Speed ratio vector	[0.0, 0.5, 0.6, 0.7, 0.8, 0.87, 0.92, 0.94, 0.96, 0.97]	
	Torque ratio vector	[2.232, 1.5462, 1.4058, 1.2746, 1.1528, 1.0732, 1.0192, 0.9983, 0.9983, 0.9983]	
	Capacity factor parameterization	Ratio of speed to square root of impeller torque	
	Capacity factor reference speed	Always impeller speed	

	Capacity factor vector	[12.2938, 12.8588, 13.1452, 13.6285, 14.6163, 16.2675, 19.3503, 22.1046, 29.9986, 50.0]	(rad/s)/(N*m)^0.5
	Interpolation method	Linear	
	Extrapolation method	Linear	
Dynamics	Model transmission lag	No lag - Suitable for HIL simulation	

Table 55: Synchronizer

Tab	Variable	Value	Units
Dog Clutch	Torque transmission model	Friction clutch approximation - Suitable for HIL and linearization	
	Maximum transmitted torque	1000	N*m
	Clutch teeth mean radius	200	mm
	Maximum engagement speed	Inf	rad/s
Cone Clutch	Contact surface maximum diameter	150	mm
	Contact surface minimum diameter	100	mm
	Cone half angle	12	deg
	Friction model	Fixed kinetic friction coefficient	
	Kinetic friction coefficient	0.3	
	Static friction coefficient	0.35	
	Velocity tolerance	0.001	rad/s
	Threshold force	1	N
Detent	Peak force	500	N
	Notch width	3	mm
	Viscous friction coefficient	0.1	N/(m/s)
	Friction to peak force ratio	0.01	
	Friction velocity threshold	0.05	m/s
Shift Linkages	Linkage travel direction	Positive shift linkage displacemenet engages clutch	
	Ring-hub clearance when dog clutch disengaged	5	mm
	Tooth overlap to engage	3	mm
	Tooth height	10	mm
	Ring-hub clearance when cone clutch disengaged	3	mm

	Hard stop at back of shift linkage	Hard stop when fully disengaged	
	Dog clutch ring stop stiffness	1000000	N/m
	Cone clutch at ring stop stiffness	1000000	N/m
	Dog clutch ring stop damping	1000	N/(m/s)
	Cone clutch at ring stop damping	1000	N/(m/s)
	Viscous friction coefficient	100	N/(m/s)
Initial Conditions	Initial state	Both clutches unlocked	
	Dog clutch initial shift linkage position	0	mm
	Cone clutch initial shift linkage position	0	mm

8.6 Engine Component Variables

8.6.1 Engine Top-Level Component Variables

Table 56: Inertia (engine inertia)

Variable	Value	Units
Inertia	0.0895	kg*m ²

Table 57: Solver configuration (engine thermodynamics)

Variable	Value
Consistency tolerance	1E+9
Sample time	0.005
Delay memory budget [kB]	1024
Filtering time constant	0.001

Table 58: Constant area pneumatic orifice (exhaust system)

Variable	Value	Units
Discharge coefficient, Cd	0.8	
Orifice area	0.0081	m ²

Table 59: Gas properties (ambient air properties and states)

Variable	Value	Units
Orifice area	0.0081	m ²

Specific heat at constant pressure	1005	J/kg/K
Specific heat at constant volume	717.95	J/kg/K
Dynamic viscosity	0.00001821	s*Pa
Ambient pressure	293.15	kPa
Ambient temperature	293.15	K

8.6.2 Throttle Body Component Variables

Table 60: Variable area pneumatic orifice (throttle throat)

Variable	Value	Units
Discharge coefficient, Cd	1	
Minimum area	1.00E-12	m ²

Table 61: Transfer fcn (ETC actuator dynamics)

Variable	Value
Numerator coefficients	[1]
Denominator coefficients	[0.04 1]

8.6.3 Intake Manifold Component Variables

Table 62: Constant volume pneumatic chamber (intake manifold volume)

Variable	Value	Units
Chamber Volume	3.4	l

8.6.4 Core Engine Component Variables

Table 63: Transfer fcn (LP filter)

Variable	Value
Numerator coefficients	[1]
Denominator coefficients	[0.001 1]

8.7 Fuel Tank Component Variables

Table 64: Fuel tank capacity

Variable	Value
Plant.FuelTank.Capacity	7

9 Appendix C

9.1 Vehicle Test Results Emissions Traces Cost Function Version 1

Figure 76: Vehicle test results cost function Version 1 CO rate and vehicle speed

Figure 77: Vehicle test results cost function Version 1 HC rate and vehicle speed

Figure 78: Vehicle test results cost function Version 1 HC rate and vehicle speed, zoomed

Figure 79: Vehicle test results cost function Version 1 NOx rate and vehicle speed

Figure 80: Vehicle test results cost function Version 1 NOx rate and vehicle speed, zoomed

Figure 81: Vehicle test results cost function Version 1 CO₂ rate and vehicle speed

9.2 Vehicle Test Results Emissions Traces Cost Function Version 3

Figure 82: Vehicle test results cost function Version 3 CO rate and vehicle speed

Figure 83: Vehicle test results cost function Version 3 HC rate and vehicle speed

Figure 84: Vehicle test results cost function Version 3 HC rate and vehicle speed, zoomed

Figure 85: Vehicle test results cost function Version 3 NO_x rate and vehicle speed

Figure 86: Vehicle test results cost function Version 3 NO_x rate and vehicle speed, zoomed

Figure 87: Vehicle test results cost function Version 3 CO₂ rate and vehicle speed