Engine Selection, Modeling, and Control Development for an Extended Range Electric Vehicle

Thesis

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

By

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ABSTRACT

Increased pressure for fuel economy improvement in combination with rapid development of battery technology has brought focus to new vehicle architectures like: hybrid electric vehicles (HEV), plug-in hybrid vehicles (PHEV), and extended range electric vehicles (EREV). These architectures require different engine characteristics which must be considered during the component selection phase of a vehicle development program. Throughout the development program a variety of different engine simulation techniques can be used to increase the efficiency of the program. Current literature has shown that a wide variety of engine simulation models have been developed and applied to many different engine research problems. These models vary greatly in their fidelity and computational efficiency. The methods which are used to build and calibrate the different models require varying amounts of empirical data and model calibration effort. With a large number of model based resources available, a key question is how to select and implement models which are best targeted for a project’s goals.

When developing a new engine control strategy, some of the driving issues are cost and resource minimization and quality improvement. This thesis outlines how a model based approach was used to choose an engine and develop an engine control strategy for an
EREV. The outlined approach allowed the development team to minimize the required number of experiments and to complete much of the control development and calibration before implementing the control strategy in the vehicle. It will be shown how models of different fidelity, from map-based models, to mean value models, to 1-D gas dynamics models were generated and used to develop the engine control system. The application of real time capable models for Hardware-in-the-Loop testing and the development of an electronic throttle control strategy will also be shown.

The application of this work is the EcoCAR advanced vehicle competition. The Ohio State EcoCAR team converted a high compression ratio compressed natural gas (CNG) engine to operate as a dedicated E85 auxiliary power unit in an EREV. This conversion required the complete development and calibration of a new engine control system. With the aim of reducing cost, minimizing required resources, and maximizing the quality of the control system, the team utilized model-based development tools from The MathWorks and other sponsors to create a family of models for software-in-the-loop (SIL) testing, hardware-in-the-loop (HIL) testing, and control algorithm development.
DEDICATION

I dedicate this thesis to my family and friends. Their support allows me to follow my dreams, wherever they may take me.
ACKNOWLEDGMENTS

I would like to thank Dr. Giorgio Rizzoni who supported my graduate school application and made it possible for me to attend Ohio State University and to work on the EcoCAR program. Thanks to Dr. Shawn Midlam-Mohler whose support and guidance is invaluable to me and the EcoCAR program in general. I also thank the entire Ohio State EcoCAR team who continually amaze me with what they can accomplish.
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1.1 Motivation
The Ohio State University applied and was accepted to compete in the EcoCAR Challenge competition, a competition that started in the fall of 2008 and continues through the spring of 2011. The competition is sponsored primarily by General Motors and the U.S. Department of Energy. The goal of the competition is to redesign a 2009 Saturn VUE to increase fuel economy, decrease emissions, and maintain consumer acceptability related to features and performance.

The three year design cycle for the EcoCAR competition requires the team to design the vehicle in year one, implement the design and prove basic functionality in year two and deliver a complete production intent vehicle in year three. The fast pace timeline of the competition presents a serious challenge, especially in the first two years of competition. The timeline dictates that the team must make component and vehicle architecture decisions without a detailed knowledge of all available components, shown in Figure 1. This requirement motivates the use of model based design principles for component and vehicle architecture selection, modeling, and control development.
After researching and simulating a large variety of design options, the Ohio State team chose a series-parallel-hybrid vehicle architecture, shown in Figure 2. In this configuration, the engine is coupled to an electric motor with a clutch and the electric motor is coupled to a fixed final drive on the front axle with another clutch. An additional electric motor is placed at the rear to drive the rear wheels. With this architecture, the engine can function in a series-hybrid mode by coupling to the electric motor-generator and isolating itself from the final drive. The vehicle also has the capability of operating the engine in a parallel-hybrid mode by engaging both clutches and allowing power to be transmitted through the final drive gear. Since there is only one fixed gear ratio this mode is intended for use at highway speeds for an increase in highway fuel economy.
Both of these modes change the way the engine operates compared to a traditional vehicle. In the series-hybrid mode the engine speed is not tied to vehicle speed variation. Thus, the engine can operate at any combination of speed and load to meet a power request. In the parallel mode the engine speed is tied to the vehicle speed request. However, the transients in speed requests are reduced since speed is usually maintained more evenly during highway driving. Also, the electric motors provide an opportunity to move the engine operating load at a given speed, either by adding load to the engine or by sharing load with the engine. In addition, the vehicle architecture utilizes a large battery pack and has the ability to operate in an all electric mode for ranges of 40 to 50 miles.

Although this vehicle architecture has the ability to drive with electricity only, the long term performance and efficiency of the vehicle is dependent on the engine. When the battery’s pure electric range has been used the vehicle must turn on the engine and
operate as a hybrid vehicle. In this mode the engine must meet the average power requirements of the vehicle with energy from liquid fuel. Even though the engine operates at a reduced duty cycle in this architecture, it remains as one of the most important components of the overall design. This thesis outlines how model based design concepts were used to help select and develop an engine which provides an optimal balance of energy efficiency and emissions reduction for the EcoCAR vehicle.

1.2 Thesis Overview

The engine development process which was followed during the first two years of the EcoCAR competition is discussed in this thesis. The development is divided into three sections described below.

- Chapter 2 – Engine Selection
  - Chapter 2 outlines the analysis and decision making process which was used to choose an optimal engine for the EcoCAR Architecture.

- Chapter 3 – Engine Simulation
  - Chapter 3 describes the overall modeling approach which was used for EcoCAR engine development. A variety of engine simulations are presented along with some comparisons with experimental results.
• Chapter 4 – Electronic Throttle Control

  o Chapter 4 describes the model based control development which was used to improve the control of the selected engine’s electronic throttle.
CHAPTER 2

ENGINE SELECTION

2.1 Literature Review on Engine Selection

A variety of considerations must be made when choosing an engine for a hybrid vehicle. This section presents an overview of relevant topics from the literature that can be used to guide the decision making process. Some topics such as engine compression ratio and engine emissions are presented because they provide relevant background understand of some of the specific engines which were available to the EcoCAR team.

2.1.1 Engine Power/Load Control

Normal vehicle operation requires a combination of acceleration, deceleration and steady speed cruising. The frequency and magnitude of these accelerations can be characterized with the type of driving that is occurring, i.e. city, highway, or aggressive driving. In order to have consistent drive data for comparison, these different driving styles have been organized by researchers into driving schedules. These driving schedules describe a series of vehicle speeds at a given times during a vehicle test. These schedules allow multiple vehicles to be tested fairly against one another for the purpose of energy usage and vehicle emissions.
With knowledge of vehicle properties like mass, inertia, etc. the necessary power needed at the wheels for a particular vehicle to follow the drive schedule can be calculated. An example of this was performed in [2] and can be seen in Figure 3.

![Figure 3: Power Required for LA4 Drive Cycle](image)

This is important because it allows different strategies for delivering this power to be analyzed. It can be noted that large spikes in power and braking are needed to follow a drive cycle. One strategy proposed by [2] is to use an auxiliary power unit (APU), the engine generator combination, as a source of constant power and to use the battery or other load leveling device (LLD) as a source or sink for either extra power or excess power during the drive cycle. Considering that drive cycles require different amounts of steady state power, single point APU operation can require a duty cycle operation to maintain the battery state of charge (SOC). An example of this style of engine control can be seen in Figure 4.
Although this type of driving may be optimal for engine efficiency and emissions it was noted in [2] that the constant APU mode may not be optimal for system efficiency and could lead to a rapid loss in battery performance. This is due to the fact that although the engine is operating at a good efficiency, at times it creates excess power that the battery has to absorb. At other times the engine does not meet the vehicle's power requirement and the battery must discharge power to meet the drive cycle. Whenever power is stored and released from the battery two new efficiencies must be considered, charge and discharge efficiency. Depending on the situation these efficiencies can cause the overall system efficiency to be less than if the engine moved its operation to a lower power and lower efficiency zone and avoided creating excess power. In addition, heavy cycling of a battery has been shown to decrease the battery life. An alternative method of operating...
the APU is to keep the battery power relatively constant and to allow the APU to follow the power request; this can be seen in Figure 5.

![Figure 5: Power During APU Follower Mode [2]](image)

The benefits of operating the engine solely in efficient zones are lost with this type of strategy. Although the engine is decoupled from the mechanical drive, the engine must still vary its operating point rapidly to meet the power demands. The main advantage of this type of hybrid strategy as compared to normal transmissions is that the engine speed is decoupled from the wheel speed. Thus, APU power can be adjusted by maintaining engine speed and fluctuating the torque output of the engine.
The previous two cases represent two extremes of APU operation. In reality, the optimal operation of the engine most lies somewhere between these two cases. In this way, both the engine and the battery share the duty of meeting the power request in an optimal manner. An alternative method of operation which was proposed in [6] involves operating the engine along an optimal line; this can be seen in Figure 6.

Figure 6: Optimal Engine Operation Line Control Strategy [6]

The previous figure demonstrates that the optimal fashion of engine operation may be to only operate in the highest efficiency zones of the engine, shown in terms of brake specific fuel consumption (BSFC). This can be done in different ways that create slightly different operation lines. One can choose a required torque and find the best efficiency for each torque value, one can find a required speed and find the best efficiency for each
speed, or one can follow iso-power lines and find the best efficiency for each power. With series APU hybrids, the engine speed is decoupled from the mechanical drive which makes an optimal operation line based on power possible and often most appropriate.

Another method is proposed in opposition to the optimal operation line in [1] where the best operation should be found at a single operation point. However, this method was proposed for use in a planetary gear hybrid power train, which is essentially a parallel design. In that design the single operation point may make more sense because the transmission can function in an infinitely variable gear ratio mode. Depending on the chosen engine and powertrain, the characteristics of the engine may make different control strategies more optimal. [2] shows that certain engines have very narrow efficiency zones which may lead to a single operating point strategy. Other engines have broad efficiency zones which make operation line strategies optimal, shown in Figure 7.
Figure 7: Examples of Fuel Consumption vs Power Rating in Different Engines [2], where (a) shows a narrow peak and (d) shows a broad efficiency zone.

Another issue related to the APU control is how to control the engine operating point, since it is a combination of controlling the engine throttle and electric generator torque or speed. An example of a typical engine operation of a series-hybrid electric military all terrain vehicle during an urban drive cycle was shown in [14] and can be seen in Figure 8.
The figure shows that during driving in large fluctuations in engine operating point can occur. As was suggested in all of the surveyed literature, it is desirable to limit this variability in the operation of a series-hybrid. Since both the engine and electric motor in a series generator are both torque controlled systems, a feedback loop for the desired speed of the coupled system must be developed. The method proposed in [14] for controlling the engine operation is the use of two independent sliding mode controllers, one for the engine and one for the generator. Sliding mode controllers were selected because they function well for single input systems of large order. They also function well in non-linear systems because they are discontinuous. In this way, a possible method for controlling the engine operation point could be to use a sliding mode controller to control the speed of the generator and another controller could be used to control the throttle of the engine. This most nearly resembles the common method of controlling engine operating points during dynamometer testing. A more basic control system could
use proportional, integral, derivative (PID) feedback control on the speed of the electric by varying motor torque maintain speed. Other feed-forward predictors of engine torque can be used to adjust the throttle to achieve the requested engine torque. An illustration of the much tighter engine operation point control accomplished with these methods can be seen in Figure 9.

![Engine Efficiency Map (HOT)](image)

Figure 9: 2 Sliding Mode Control Engine Operation [14]

One method for controlling the system operation of a hybrid is proposed in [26]. This method is called Equivalent Consumption Minimization Strategy (ECMS). In this method the controller looks at the entire system efficiency in the form of a non-linear penalty function and performs minimization of this function to find the best operating point for the APU. An example of resulting ECMS engine operation in a parallel-hybrid can be seen in Figure 10.
One of the main advantages of this control method is the ability to add in additional system consideration to the cost function. With appropriate weighting applied to the various characteristics considered in the cost function, the best operating strategy can be determined at each point in time. An example on how the previous engine operation map varies when weight is added to minimizing the engine’s NOx emissions is shown in Figure 11, where the operation points move to lower loads which align with contours of reduced NOx production.
It can be noted that the control strategy shifts to move the engine operation points to lower load points at slightly higher rpm in a an attempt to minimize the effect of NOx emissions on the overall cost function. This control strategy seems promising because it provides the best opportunity to consider the largest variety of operating parameters. One drawback is that it seems to be computationally intense as it is finding the minimum of a larger non-linear cost function.

2.1.2 Engine Compression Ratio

An important consideration when designing or selecting an engine for a particular application is the compression ratio of the engine. The compression ratio of an engine relates the maximum volume of a cylinder at the bottom of the piston stroke to the cylinder clearance volume at the top of the piston stroke, shown in equation (1).
Where \( b \) is the bore diameter of the cylinder, \( s \) is the stroke of the piston, and \( V_c \) is the clearance volume. Compression ratio is important because it has been shown to have a direct effect on the peak efficiency of an engine, an example of how efficiency increases with increasing compression ratio is shown in Figure 12.

![Diagram](image)

Figure 12: Change in Thermal Efficiency with Increasing Compression Ratio [18]

The figure shows the effect of compression ratio on different experimental engines, KT and CN, on both the indicated and brake efficiency as compared to the predicted increase
based on a standard fuel-air cycle. Considering this relation, one would believe that all engines would be designed with a compression ratio of 15 to 16. However, most production spark ignited gasoline engines have compression ratios of 9 to 11. Some of the key limitations that require lower compression ratios are knock limitations of gasoline fuel and durability limits of engine hardware. It has been shown in the literature that by using engine hardware designed to handle high compression ratios, alternative fuels such as E85, 85% ethanol and 15% gasoline by volume, can enable the use of high compression ratios without the knock limitations of gasoline. This is illustrated by varying the fuel in a single cylinder engine and measuring the efficiency of the engine with increasing compression ratio until the knock limit of the fuel is met, shown in Figure 13.
Figure 13: Experimental Measurement of Thermal Efficiency Difference in a High Compression Ratio E85 Fueled Engine [4]

The primary factor allowing E85 to operate at higher compression ratios without knock is octane rating. The octane rating of a fuel is a measure of how the knock characteristics of a particular fuel compare to the knock characteristics of a mixture of iso-octane and heptane. An octane rating of 90, a typical value for gasoline, means that the fuel behaves like a mixture of 90% iso-octane and 10% heptane. Values over 100 are possible for fuels like compressed natural gas, ~120 octane, and E85, ~105 octane. Another fuel property, the stoichiometric air/fuel ratio, governs the mass of air that must be mixed with a unit mass of fuel for stoichiometric combustion. The air/fuel ratio for gasoline is nominally 14.7 and the ratio for E85 is nominally 9.8. This means that for a given amount of air flowing into the engine, more E85 must be used as compared to gasoline. As a result, although high compression ratio E85 engines have been shown to have higher
efficiencies, the specific fuel consumption remains higher for E85 fueled engines, shown in Figure 14.

![Figure 14: Comparison of E85 and Reformulated Gasoline Indicated Thermal Efficiency (ITE), Indicated Mean Effective Pressure (IMEP), Indicated Specific Fuel Consumption (ISFC) with Varying Compression Ratio Engines [28]](image)

Other studies have shown additional increases in efficiency with high compression ratio E85 engines by including exhaust gas recirculation (EGR) and by adjusting the average operating point of the engine from high speed low torque to high torque low speed operation points. These shifts have also been shown to allow for emissions reduction as compared to standard gasoline engines, shown in Figure 15.
Figure 15: Effect of E85 and Increased Compression Ratio on Engine Thermal Efficiency and Emissions [12]
2.1.3 Engine Emissions

Engine out emissions are a major consideration for the operation of the series-hybrid vehicle. The literature has revealed that the primary sources of engine emissions are cold start, engine start/stop and engine transients. The start and stop characteristics are especially important for the EcoCAR team since the vehicle has the ability to operate all electric and may have multiple starts and stops during a driving cycle. In [9], start-stop and transient emissions are analyzed using fast emissions sensors. An example of hydrocarbon emissions during start up can be seen in Figure 16.

![Figure 16: Engine Out and Tailpipe Out HC Emissions During a Vehicle Cold Start [9]](image)

This figure reveals that a large amount of hydrocarbon emissions are released in the first few seconds of engine start. A lower amount of emissions are seen in at the tailpipe but
they are still significant. In Figure 17 it is shown more clearly the source of these emissions during start up.

![Figure 17: Cycle Resolved HC Emissions During a Vehicle Cold Start [9]](image)

The figure shows both measured hydrocarbon emissions and in-cylinder pressure. It is shown that during the first few cycles of starting misfires can occur. These misfires are characterized by low in cylinder pressure and high hydrocarbon emissions since the fuel-air mixture is left un-burnt. In order to minimize these emissions during a drive cycle, the control strategy must either minimize start stop conditions or a significant amount of modeling and calibration work must be performed to characterize and minimize these misfiring events during startup. Although three way catalytic converters have been shown to be very effective at reducing emissions for gasoline engines, they are only effective when the engine is operating near stoichiometry and after warming to a “light off”
temperature. Significant HC emission spikes can also be seen during vehicle acceleration and gear shifts, especially before the catalyst reaches its light-off temperature. These spikes can be seen in Figure 18.

![Figure 18: Transient Emissions Before Catalyst Light Off][9]

Engine control can minimize these types of emission spikes in two ways. One way is to reduce the frequency and magnitude of engine transients while the catalyst is warming. Another possible solution is to increase the rate at which the catalyst warms. For instance, hybrid electric vehicles can elevate exhaust temperature with spark retardation. The resulting reduction in torque output can be overcome with electric torque boosting. Electrified powertrains also provide a significant energy source to power emissions treatment systems like electrically heated catalysts. Electric catalyst heating has shown potential to reduce tailpipe emissions. A plot of exhaust temperature versus time with varying heating temperatures can be seen in Figure 19.
This rapid rise in catalyst temperature is good because it allows the catalyst to light-off much more quickly. This could decrease the emissions penalty due to increased cold starts in plug-in hybrid vehicles. The vehicle control will need to be optimized based on the weighting factors for emissions reduction and efficiency improvement. A plot of the reduction in peak and average CO emissions for an electrically heated catalyst at different temperatures can be seen in Figure 20.
The literature has also shown that emissions are a major problem for diesel engines, especially NOx emissions. It has been shown that oxidation catalysts function well to reduce CO and THC emissions because of the diesel’s lean burn operation. It has also been shown that diesel particulate filters (DPF) are very effective at removing soot from diesel emissions. One of the most prevalent after treatment systems for NOx reduction on modern diesel engines is the selective catalytic reduction (SCR) catalyst. Typically the SCR reacts with injected urea to form ammonia which reacts with the NOx in the engine out emissions to reduce the tailpipe NOx emissions. Significant reductions in NOx emissions have been found using SCR systems, an example of typical NOx conversion efficiencies vs. SCR temperature is shown in Figure 21.
2.2 Vehicle Load Comparison

As was shown in the literature, vehicle component sizing and selection is primarily driven by the expected load cases for the component. One way of predicting the load on the engine during both steady state operation and transient driving is to use a backward looking vehicle model. This type of model predicts the forces acting on the vehicle at each time step of a drive cycle and allows for various properties like vehicle power and wheel torque requirements to be calculated. Equations (3) (4) express how the sum of forces acting on the vehicle, $F_{total}$, can be used to calculate vehicle power, $P$, and wheel torque, $T_w$, where $V$ is the vehicle velocity and $R_w$ is the tire radius.

\[ F_{total} = F_{Aero} + F_{rolling} + F_{grade} + F_{inertia} \] (2)
The total force acting on the vehicle is computed by first calculating the aerodynamic, rolling resistance, road grade and vehicle inertial forces. Equation (5) is used to calculate the aerodynamic resistance of the vehicle where $\rho$ is the density of air, $C_d$ is the vehicle’s coefficient of drag, and $A$ is the frontal area of the vehicle.

$$F_{\text{Aero}} = \frac{1}{2} \rho C_d A V^2$$

Equation (6) calculates the rolling resistance of the vehicle by multiplying the normal force of the vehicle on the road by the coefficient of rolling resistance, $C_r$. The normal force is calculated by using the effective mass of the vehicle, $m$, gravitational acceleration, $g$, and the angle of the road surface, $\alpha$. The effective mass of the vehicle includes the mass of a trailer if one is attached.

$$F_{\text{rolling}} = mg C_r \cos(\alpha)$$

Equation (7) calculates the additional longitudinal force added on the vehicle by the grade of a road surface.
Equation (8) calculates the inertial force of the vehicle by doing a backwards approximation of the vehicle acceleration, \( \frac{dv}{dt} \), by using the vehicles current speed and the next vehicle speed specified in the drive cycle.

\[
F_{inertia} = m \frac{dv}{dt}
\]  

Table 1 summarizes the assumed parameters which are used in the backward looking vehicle model described by the preceding equations. These parameters are for a 2009 Saturn VUE crossover utility vehicle.
Using the simple backward looking vehicle model the properties of three different drive cycles were analyzed, the Federal Urban Driving Schedule (FUDS), Federal Highway Driving Schedule (FHDS), and the US06 Supplemental Federal Test Procedure. These three were chosen because they represent a wide variety of driving. A combination of city driving, mild highway cruising, and aggressive/high speed driving provides representation of the many types of drivers who may use a vehicle. Since hybrids are expected to be very efficient, customers expect good fuel efficiency with mild driving similar to the FUDS and FHDS drive cycles. Typically the US06 drive cycle is used to ensure the hybridized powertrain maintains an adequate amount of performance, although other performance tests like 0-60mph acceleration are also used. However, a vehicle which provides good efficiency on an aggressive drive cycle like US06 shows a relative insensitivity of the vehicles efficiency to the drive cycle. The vehicle speed trace and histogram of the speeds for the FUDS drive cycle is presented in Figure 22.
As a representation of city driving the FUDS drive cycle has a large number of accelerations to 25-30 mph and decelerations to a stop. Some variety is provided in the drive cycle by an initial acceleration to 55 mph. From looking at the histogram a good representation of the FUDS drive cycle speed is 25 mph. A power profile and positive power histogram for the FUDS cycle was generated with the backward simulator and is shown in Figure 23.
Analysis of the FUDS power profile for the EcoCAR vehicle reveals that only a small amount of positive power is needed from the APU, 10.5 kW average power. Because of the relatively low speeds of the FUDS drive cycle this power will need to be met in a series mode with the EcoCAR architecture. The speed profile/histogram for the FHDS cycle is shown in Figure 24.
The speed profile of the FHDS is much higher than the FUDS and is more steady. This provides a good representation of highway cruising with an average speed of 50 mph. The FUDS power profile/histogram is shown in Figure 25.
The FHDS power profile reveals a higher average power requirement of approximately 18 kW, which is consistent with the higher speeds of the profile. The peak power is actually less than that of the FUDS cycle because of the lack of heavy acceleration. The speed profile/histogram for the US06 cycle is shown in Figure 26.

![Speed vs Time](image1)

![Vehicle Speed Histogram](image2)

**Figure 26: US06 Drive Cycle Profile and Speed Histogram**

The US06 cycle provides a much wider variety of vehicle speeds and accelerations. Based on its histogram a good representative speed for this drive cycle is 65 mph. Large accelerations are seen at both low and high speed in this drive cycle. This leads to much larger power requirements which are shown in the US06 power profile/histogram in Figure 27.
The large accelerations lead to a much higher average power requirement of 34 kW. The peak power requirement is much higher as well, a requirement which is considered in the sizing of the electric powertrain components. Because of primarily series nature of the EcoCAR hybrid architecture, the APU requirements for the three discussed drive cycles were compressed into single representative operating points. This was justified because of the fast paced design cycle where a full vehicle simulator for the designed architecture was being developed in parallel with the engine selection process. The chosen operating points are shown in Table 2, where the engine speeds and torques assume the vehicle is operating in parallel mode with a fixed 2.77:1 final drive ratio. The power requirements can also be met at many other operating points if the vehicle remains in series mode.
Table 2: Drive Cycle Representative Load Cases

<table>
<thead>
<tr>
<th></th>
<th>FUDS</th>
<th>FHDS</th>
<th>US06</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Speed (mph)</strong></td>
<td>25</td>
<td>50</td>
<td>65</td>
</tr>
<tr>
<td><strong>Power (kW)</strong></td>
<td>10</td>
<td>20</td>
<td>35</td>
</tr>
<tr>
<td><strong>Engine Speed (rpm)</strong></td>
<td>845</td>
<td>1688</td>
<td>2195</td>
</tr>
<tr>
<td><strong>Engine Torque (Nm)</strong></td>
<td>113</td>
<td>113</td>
<td>152</td>
</tr>
</tbody>
</table>

The backwards simulation can also be used to calculate the steady state road load requirements at a variety of vehicle speeds. This is important for understanding the steady state load which must be met by the vehicles APU in order to sustain charge in extended range operation. This type of analysis also provides a good platform for understanding how towing a trailer on a grade affects the power requirements of the vehicle. Figure 28 shows the EcoCAR vehicle power requirements for both normal operation and towing.

![Figure 28: Road Load Vehicle Power Requirement](image)
This analysis reveals a significant increase in the vehicle power requirement when towing a 1500 lb trailer on a 3.5% grade. This steady state speed analysis reveals that the APU should be sized to have a minimum continuous power rating of 50 kW in order to allow trailer towing at 70 mph and normal operation cruising at 80 mph. The vehicle power requirement can be converted into a wheel torque requirement and engine torque requirement using the radius of the wheel and the final drive ratio of the EcoCAR transmission, shown in Figure 29.

![Figure 29: Road Load Wheel and Engine Torque Requirement](image)

Plotting the requirement gives insight into the parallel mode operation requirements for the APU. The normal mode operation reveals that the engine should have a minimum of 170 Nm of torque at 2700 RPM to cruise in parallel mode. If parallel towing is a requirement, the engine must make 180 Nm at 1000 RPM to tow at approximately 45 mph.
2.3 Engine Comparison

With knowledge of the vehicle requirements for the APU, five available General Motors engines were analyzed to understand their performance characteristics. Another possible engine was assessed, a compressed natural gas (CNG) Honda engine was made available for donation if the team converted the engine to run on E85. A list of six available engines for the EcoCAR APU is shown following.

List of Considered Engines
- GM 1.3L Diesel
- GM 2.0L Diesel
- GM 1.6L Gasoline
- GM 1.8L Gasoline
- GM 2.4L Gasoline
- Honda 1.8L E85, High Compression Ratio

Since the EcoCAR competition is structured in a way to consider energy efficiency, as opposed to fuel consumption, plots of the engines’ brake efficiency vs. operating conditions were created to better understand the benefits of each engine. In order to understand how the engines would operate as a series APU, a plot of best engine efficiency vs. power rating was created for each considered engine. Similar to the ideas from the literature, this type of analysis reveals information not only about the overall efficiency but about the appropriateness of the engine for different control strategies. For example, engines with relatively flat efficiency vs. power curves are better suited to best efficiency line operation. Also, if the final drive ratio is correctly chosen, a flatter efficiency line may lend itself better to efficient parallel operation. Engines with narrow efficiency curves are more well suited for series only, single operating point duty cycling.
operation. The plots for the two diesel engines are shown in Figure 30, where the considered power range was limited from 10kW to 60kW based on the single point FUDS load case and the maximum steady state towing power.

The diesel engines show promise with good efficiency and high torque. Typically, the peak torque of the diesel engines occurs low in the speed range of the engine which lends itself well to the possibility of parallel towing. Based on the parallel mode vehicle load analysis from the previous section it seems that the 2.0L diesel engine may be too large
to serve the needs of the vehicle in an efficient manner. Although the peak efficiency is
good, the peak torque is very high and may not be useful with the vehicles selected gear
ratio. The 1.3L diesel seems well suited to the requirements of the vehicle powertrain.
The plots for the three gasoline engines are shown in Figure 31.
Figure 31: Gasoline Engine Efficiency Plots
The plots of the gasoline engine efficiency reveal that the 1.6L engine may be too small for the intended vehicle. The peak torque is relatively low and the peak in the efficiency vs. power curve means that the engine would be less than optimal for parallel operation. The 1.8L and 2.4L seem to be adequate in terms of peak torque; however the efficiency of the 1.8L is less than desirable. The efficiency vs. power plot for the 2.4L is extremely flat in the range of powers considered. In fact, it is so flat that it seems the engine was either designed and/or calibrated to have a flat efficiency vs. power curve. Overall the 2.4L seems like a good option, the primary drawback of this engine is the extra size/weight as compared to smaller gasoline engines.

Considering the fast paced vehicle design cycle and the lack of available data for an E85 Honda engine, the efficiency maps for the 1.8L high compression ratio engine were created by modifying the GM 1.8L gasoline engine data to shift the peak efficiency to roughly 39%, as was shown possible in the literature. This was done by making the assumption that the amount of air entering the cylinder at each operating point is nominally the same. Thus, the efficiency increase is realized as an increase of torque output for the same energetic combination of air and fuel. This follows logically from the idea presented in the literature that the primary gain in efficiency in high compression ratio E85 engines comes from the increase in stroke. The longer stroke allows for a more effective conversion of cylinder pressure to torque output. The other primary efficiency increase comes from the decrease in exhaust temperature from the latent heat of vaporization of ethanol, which leads to decreased thermal losses. The approximated efficiency maps for the Honda 1.8L engine are shown in Figure 32.
Examining the predicted efficiency plot of the 1.8L high compression ratio E85 engine reveals behavior very similar to the 1.3L Diesel. The peak torque of the engine seems to be adequate and the shape and magnitude of the efficiency vs. power plot is very good for the power range associated with the EcoCAR vehicle. These somewhat qualitative assessments reveal that the two lead contenders for the EcoCAR APU are the 1.3L Diesel and the 1.8L E85 engine.

The single point drive cycle load cases which were established in Table 3 were next used to analyze the efficiency of the different engine options for specific drive cycles. The efficiencies for the FHDS and US06 were analyzed as if the engine was operating in both the series and parallel modes. The efficiencies for the FUDS drive cycle were only analyzed for the series mode because the speed of the drive cycle does not allow for significant parallel operation. In order to highlight the benefits of the parallel operation
mode the efficiencies were analyzed based on the actual engine operating point efficiency as well as the overall hybrid system efficiency. It was assumed that the hybrid system efficiency is series mode is 80%, based on two motors and inverters with 92% and 97% efficiencies respectively. The parallel mode transmission efficiency was assumed to be 98%. The results for both analyses are shown in Table 3 and Table 4 respectively.

Table 3: Single Point Load Case Engine Operation Efficiency

<table>
<thead>
<tr>
<th>Engine Operating Point Efficiency</th>
<th>Series Mode</th>
<th>Parallel Mode</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>FUDS</td>
<td>FHDS</td>
</tr>
<tr>
<td>1.3L D</td>
<td>0.340</td>
<td>0.376</td>
</tr>
<tr>
<td>2.0L D</td>
<td>0.345</td>
<td>0.375</td>
</tr>
<tr>
<td>1.6L G</td>
<td>0.330</td>
<td>0.341</td>
</tr>
<tr>
<td>1.8L G</td>
<td>0.300</td>
<td>0.325</td>
</tr>
<tr>
<td>2.4L G</td>
<td>0.330</td>
<td>0.355</td>
</tr>
<tr>
<td>1.8L E85</td>
<td>0.345</td>
<td>0.381</td>
</tr>
</tbody>
</table>

Table 4: Single Point Load Case Hybrid Operation Efficiency

<table>
<thead>
<tr>
<th>Hybrid System Efficiency</th>
<th>Series Mode Efficiency</th>
<th>Parallel</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>FUDS</td>
<td>FHDS</td>
</tr>
<tr>
<td>1.3L D</td>
<td>0.271</td>
<td>0.299</td>
</tr>
<tr>
<td>2.0L D</td>
<td>0.275</td>
<td>0.299</td>
</tr>
<tr>
<td>1.6L G</td>
<td>0.263</td>
<td>0.271</td>
</tr>
<tr>
<td>1.8L G</td>
<td>0.239</td>
<td>0.259</td>
</tr>
<tr>
<td>2.4L G</td>
<td>0.263</td>
<td>0.283</td>
</tr>
<tr>
<td>1.8L E85</td>
<td>0.275</td>
<td>0.303</td>
</tr>
</tbody>
</table>
After considering the hybrid system efficiencies the 1.3L Diesel and 1.8L E85 engines seem to be the best choice based on their parallel mode operating efficiency for both the FHDS and US06 drive cycles. Also, both engines have similar efficiencies in series operation. Another way to compare the performance of these two engines is to overlay the parallel mode torque loads from normal vehicle operation and towing, shown in Figure 33.

![Figure 33: Parallel Operation Comparison, 1.3L Diesel vs. 1.8L E85](image)

The normal operation vehicle load torque lines up well with the peak efficiency zones of both engines. The primary difference is that the peak torque of the 1.8L E85 engine is not sufficient to perform charge sustaining parallel towing. The 1.3L Diesel has a narrow range of speeds where parallel towing is possible. In the EcoCAR competition fuel consumption is not measured for the towing tests which makes this a difference with minimal importance since both engines are capable of towing in series mode.
The engine out and tailpipe emissions are a major difference in gasoline/E85 engines and diesel engines. The relative emissions between the 1.3L Diesel and 1.8L E85 engine were approximated by using steady state emissions data provided for the 1.3L Diesel and 2.4L Gasoline engine. For a fair comparison the emissions were converted from mass flow rates, \(\frac{g}{\sec}\), to specific quantities, \(\frac{g}{kWh}\). The engine out emissions for both types of engines are compared by the histograms in Figure 34.
Figure 34: Engine Out Emissions, Gasoline vs. Diesel
The engine out emissions are better for the diesel engine but are well above the allowable amounts for tailpipe emissions. As discussed earlier, the emissions for the gasoline engine can be reduced by the use of a three way catalyst. The primary factor affecting the performance of three way catalysts is the engine’s equivalence ratio, where CO and THC emissions are reduced with lean equivalence ratios and NOx is eliminated with rich equivalence ratios. The equivalence ratio map for the 2.4L Gasoline engine and the assumed catalyst conversion efficiency curve, based on data derived from experiments performed on a GM LE5 engine at Ohio State University, are shown in Figure 35.

![Equivalence Ratio Map and Catalyst Conversion Efficiency](image)

**Figure 35: Gasoline Equivalence Ratio Map and Catalyst Conversion Efficiency**

As discussed earlier, diesel emissions are controlled by a variety of aftertreatment components. CO and THC emissions are handled well with an oxidation catalyst because diesel engines use lean equivalence ratios. NOx emissions see almost no reduction from the oxidation catalyst for the same reason. The NOx emissions for this analysis are assumed to be processed by an SCR system. It has also been shown that particulate
emissions can be handled very reliably with the use of a DPF and for this reason soot emissions are not calculated for APU selection purposes. The conversion efficiency of both the oxidation catalyst and SCR is primarily affected by the catalyst and SCR temperatures. It was assumed that these temperatures are primarily a function of the steady state exhaust temperature of the diesel engine. Thus, the conversion efficiency of the after treatment system can be calculated by using the exhaust temperature map and the conversion efficiency map shown in Figure 36, SCR data based on [8] and oxidation catalyst data derived from experiments performed on a GM R425 DOHC diesel engine at Ohio State University.

![Exhaust Gas Temperature Map and After treatment Conversion Efficiency](image)

**Figure 36: Exhaust Gas Temperature Map and After treatment Conversion Efficiency**

The conversion efficiencies were applied to both the gasoline and diesel engine out emissions and significant reductions in all three emissions categories were seen. The diesel shows slightly better performance with CO and THC emissions but the gasoline engine emissions are comparable. The biggest difference is seen in the tailpipe NOx emissions, where the gasoline emissions are significantly less, shown in Figure 37.
Figure 37: Tailpipe Emissions, Gasoline vs. Diesel
2.4 Engine Selection

Based on the comparisons made in the previous section the 1.8L high compression ratio engine was chosen to function as the APU for the EcoCAR vehicle, shown in Figure 38. Preliminary estimates of this engine's performance show a good balance of engine efficiency and tailpipe emissions. This balance is critical because of the diverse set of scoring criteria and vehicle tests used in the EcoCAR competition. Other factors influencing the decision include the availability of engine hardware. Although both the 1.3L Diesel and 1.8L E85 engines were made available at no cost to the team by GM and Honda respectively, Honda was able to provide two engines at a much earlier delivery date. This allowed the team to have one engine dedicated to dynamometer testing and one for final use in the vehicle. In addition, the need to convert the Honda engine to run on E85 allows for a greater educational experience because a new engine control system needed development. A picture of the selected engine is shown in Figure 38.

Figure 38: Honda 1.8L CNG Engine [31]
CHAPTER 3
ENGINE SIMULATION

3.1 Introduction

3.1.1 Motivation for Modeling
As discussed earlier, the engine which was selected for use in Ohio State EcoCAR comes delivered with a fuel system and engine calibration for CNG. Since the preferred fuel for the Ohio State team is E85, the fuel delivery system must be modified to deliver E85 and the engine must be completely recalibrated to work with the new fuel. No data from the original calibration was made available and no simulation model was provided. However, a reasonable amount of experimental data was found in [31].

In addition, the recalibration effort must occur over a relatively short time schedule, less than six months, shown in Figure 1. It must also be performed with a single engine and dyno setup with a small number of students to design, perform, and analyze results of relevant experiments. For this reason, it is desirable to limit the amount of experiments which must be performed by developing a variety of engine simulations which can perform virtual experiments.
3.1.2 Key Challenges

The key challenges based on the specifications of the engine are the large number of experiments which must be performed to calibrate an engine with variable valve timing (VVT) and a variable length intake. In the specific case of the Honda CNG engine, this effort is reduced somewhat by the fact that both variations are discrete choices.

The VVT system gives the calibrator the choice of a high output cam or a delayed closure cam profiles. This choice modifies both the lift profile and timing of one intake valve. The profiles of the other three valves, two exhaust and one intake, remain the same during both choices. The main purpose of adding the delayed closure cam to the engine is to reduce the effective compression ratio and volumetric efficiency of the engine in certain operation points. This allows the throttle body to open more during those operation points and thus reduce the amount of pumping losses in the engine.

The variable length intake system is used in the engine to increase the amount of available torque at high engine speeds. This is done by taking advantage of engine acoustics through the use of a tuning valve in the intake. The operation of the intake tuning valve system can be seen in Figure 39.
The tuning valve provides two choices in the effective length of the intake. The shorter path works better for higher speed operation and the longer path works better for lower speed operation. This is based on matching the resonance of the intake system to the speed of the engine.

### 3.1.3 Goals for Modeling

The lack of an engine calibration and the overall combination of two sets of valve timing and two sets of intake length provides the possibility of having to essentially calibrate the engine four times. With a lack of manpower and time to perform an in depth full factorial calibration, the OSU team plans to use engine modeling to achieve the following goals:

- To better understand the dynamics of the engine
- To create a minimized design of experiment (DOE) for calibrating the engine
- To increase the density of the calibration mesh based on simulation rather than experimental results
3.1.4 Model Types

During the model based development process, three main types of engine models were developed, black-box models, mean-value models, and 1-D gas dynamics models. These models vary both in their fidelity and their computation time. This trend is generalized in Figure 40.

![Figure 40: Model Computation Time vs. Model Fidelity](image)

The black-box models which were developed are input/output models which predict engine performance based on different types of inputs, e.g., engine speed, torque request, etc. These models capture the engine performance without a model structure based on knowledge of physical phenomena involved in the engine system. These models functioned as lookup tables and resulted in very fast computation times.
Different mean-value models were developed, some for SIL and some for HIL testing. Similar to the black-box models, the mean-value models are reliant on performance data which is found externally. However, these models do include some knowledge of the engine processes in the structure of the model as well as some physical engine parameters like displacement and intake manifold volumes. These models are capable of calculating how air dynamics affect the mean engine operation from cycle to cycle. Of the three model types used, these provide an intermediate value of both fidelity and computation time.

The 1-D gas dynamics models were developed using Gamma Technologies GT-Power software. GT-POWER is a commercial engine modeling software which is capable of calculating 1-D gas dynamics. The use of this type of software is becoming increasingly common among major automotive manufacturers. In terms of computation intensity this type of model falls between a mean-value model on the low end and a 3-D CFD model on the high end. These were the most physically based models developed. The model structure accounts for intake and exhaust geometries with a 1-D discretization along the air flow path. These models also account for in-cylinder processes with port flow models and combustion models. These models provide the highest amount of fidelity but also the longest computation times of the models considered.

Because of the varying model structures, the different models were each used for different purposes. The properties of the black-box models were particularly well suited for vehicle development studies like fuel consumption prediction. The mean-value
models worked well for more advanced vehicle studies, control development, and for HIL testing. The 1-D models functioned well for more detailed component level studies.

3.2 Literature Review
Current literature has shown that a wide variety of engine simulation models have been developed and applied to many different engine research problems in [10],[17],[18],[27], and [30]. These models vary greatly in their fidelity and computational efficiency [18]. The methods which are used to build and calibrate the different models require varying amounts of empirical data and model calibration effort [11]. It has also been shown in [7] that a variety of models, from advanced 1-D to simpler mean-value simulations, are useful through all stages of the vehicle design process. In addition, [25] provides background into the benefits of coupling GT-POWER simulations with other post processing techniques to refine the performance of engines with flexible valve actuation systems. The following portion of the literature review focuses on the topic of coupling mean-value engine simulations with hardware-in-the-loop simulators to aid in engine control development.

Typical mean-value engine models can be broken into three main sections; the air path, the fuel system, and the combustion chamber. Where the combination of air and fuel flow into the combustion chamber is used to predict a mean-value torque for the combustion cycle. Within these sections, the model can be broken down into component models including the throttle, intake manifold, fuel injectors, etc. An example of the
interconnection of these component models is shown in Figure 41. The figure reveals a variable air flow path architecture which includes many components found in a variety of engines, *i.e.* turbocharger, intercooler, EGR valve, *etc.*. These inclusions allow a user to subtract components to represent their particular system.

Figure 41: Mean-value Engine Model Air Flow Path [27]

Although mean-value engine models predict the average torque of the engine during a cycle, the use of torque shaping functions allows these medium fidelity models to provide torque dynamics indicative of more detailed combustion models. These dynamics allow for modeling misfire which can be used to tune fault diagnostic systems, Figure 42.
In general, the dynamic states which can be included in a mean-value engine model lend themselves well to interacting with the input/output (I/O) connections of modern electronic controllers. A diagram of typical controller I/O for a modern gasoline direct injection (GDI) engine is shown in Figure 43.
In order to increase the speed and robustness of engine control unit (ECU) testing many companies have turned to hardware-in-the-loop testing. This refers to a loop where the I/O of the engine controller is simulated in a hardware simulator and connected directly to the engine controller through physical connections, shown in Figure 44. This loop allows the engine controller to function as it does when connected to an actual engine or in an actual vehicle. This allows the software and hardware to be validated as a system.
This also provides the opportunity to take difficult to model systems out of the simulation and to use the actual plant. Figure 45 reveals how the actual electronic throttle can be connected to the ECU and simulation model simultaneously, where the ECU controls the position and the model predicts airflow based on the measured position. The inclusion of high speed time processors can also allow the engine model to simulate the high speed pulses of the crank and cam position sensors. In addition, the injection and ignition pulses can be captured and be used to control the fuel system and torque prediction systems of the mean-value model.
Even with coarsely calibrated engine simulations, this connection of engine control hardware with simulated engine states can allow the engine controller to “believe” that it is operating in a normal state. This allows the user to insert faults like a stuck throttle condition or missed cam sensor pulse and gauge the ability of the ECU to detect and remediate these faults.

3.3 Modeling Process

Before the team began to develop models and perform experiments on the engine, a modeling process was defined. By defining a process up-front, the team was not only able to understand the task better but was also able to understand what needed to be done in each stage of development. The team first looked at how models had been developed in previous iterations of the collegiate vehicle design competition. This process can be seen in Figure 46.
This linear process of model development allows different design groups to develop different models for different purposes. Each group works separately to develop the experimental requirements for each model’s calibration. Also, each group can pass information to the other about how changes to the individual models may affect the others. This process does allow for some level of parallel model development; however, the team found that the process is too segmented for a small collegiate design group. Also, since the models are developed with independent experimentation, the team recognized that this type of process may lead to unnecessary amounts of repeated experimentation.

A more optimal process was found by moving away from a linear process and looking for a circular process. The process that the team decided to use is illustrated in Figure 47. This method works by dividing development into two processes, modeling and experimentation. There are also two loops in the process. An external loop connects the modeling process to the experimentation process. An internal loop connects the data generated by the different model types to one another.
The key to this process is that all of the models are developed at the same time with the same information. For instance, the team enters the process with some amount of data about the system and then follows the progression in Figure 48 as many times as is necessary.

1. Use known data to develop models
2. Use data generated by high fidelity models to fill in gaps in data coverage for lower fidelity models
3. Perform simulations to understand which portions of the models need improvement
4. Generate a request for more data and send to the experimentation process
5. Pass generated data to the modeling process

Figure 48: Steps in Circular Modeling Process
Following this strategy allowed the different groups in the team to begin working at the same time and for all of the models to develop simultaneously. Also, sharing generated data in the modeling loop allowed the team to minimize the number of times that it was necessary to use the external experimentation loop. A more detailed diagram of the team’s specific sequence of model development is shown in Figure 49.

![Figure 49: OSU Model Development Process](image)

As shown in Figure 49 the team started with a limited amount of data about the engine geometry and experimental performance characteristics of similarly sized engines. This
information was used to generate both a GT-POWER simulation as well as a black-box model. The GT-POWER simulation was then used to generate simulation based version of both a mean-value and black-box model. The second round of data collection and experimentation inserted more detailed engine geometry data as well as a limited number of preliminary experimental data points. This was used to increase the accuracy of the previous models and spawn a new generation engine models. The following sections will describe more in depth some of the model development and model results with different types of models.

### 3.4 Black Box Models

The black-box engine models developed during year one and two of the EcoCAR competition are essentially lookup tables. Based on certain input conditions, the models lookup different output states from a static table. The lookup values are derived from steady state operation values which can be determined experimentally or estimated from similar engines or higher fidelity engine simulations. The engine efficiency tables which were used in the engine sizing section of this thesis are an example of these types of table lookups. Based on certain engine operating speeds and torques, the associated efficiency and/or fuel consumption can be determined directly from an efficiency/fuel consumption surface.

The primary use of these models is to estimate fuel consumption in a vehicle simulator. Integrating the black-box engine model in to the team’s EcoSIM vehicle simulator allows
time based fuel consumption to be estimated with engine torque demand and engine speed provided as inputs to the model. Engine speed is determined externally from this model because the vehicle architecture allows the engine to be coupled to either the front electric motor or the front electric motor and the vehicle through a single speed transmission. The engine torque demand is determined by the vehicles supervisory control strategy, ECMS. This strategy takes into consideration conditions like driver torque request and battery SOC to determine the optimal engine torque request.

The black-box model structure is fairly simple and is best explained by Figure 50. With torque command as an input, the dynamics of torque request to production are simulated as a torque filter and torque saturation block. This lumps in the dynamics of throttle positioning, manifold air dynamics, etc. into a filter. The torque saturation block ensures that the torque of the engine does not exceed the maximum and minimum torque limits. The output of the dynamic section of the model outputs an engine achieved torque which is tied in with the other components of the vehicle. The fuel consumption of the engine can then be looked up based on the engine achieved torque and the imposed engine speed.
Figure 50: Black-box Engine Model Calculation Method

An example of how the engine torque dynamics are represented by the filter is shown in Figure 51. This response was achieved by using a first order filter with a 0.1 second time constant.

Figure 51: Black-box Engine Torque Output vs. Command
Based on the torque output and imposed engine speed the fuel consumption at each time step is looked up from the steady state fuel consumption map. The engine operation points for two drive cycles are shown in Figure 52.

The engine operation points in Figure 52 are shown overlaid on the fuel consumption table which was predicted for the 1.8L E85 engine in the engine sizing section of this thesis. The engine operation points reveal that the initial ECMS strategy chooses to operate the engine at discrete engine speeds. Future implementations should provide a wider of variety of engine speeds which will maximize the efficiency of engine operation. This data was used for initial fuel consumption estimation and was later updated when empirical data for the actual engine was available in year two, shown in Figure 53.
Both the predicted fuel consumption curves and the measured curves are very similar because fuel consumption is related to iso-power curves through mass air flow, air/fuel ratio, and thermal efficiency. The primary difference is that the peak torque is much lower than originally predicted. This forces the engine operation points to be compressed within the lower peak torque values.

3.5 1-D Gas Exchange models

3.5.1 Early Development GT-POWER Model

The initial approach for high fidelity engine modeling was to create GT-POWER model based on data available in the engine service manual, from physical measurements and from published literature. The model was also validated against a small amount of experimental results available in published literature. This physically based model was
then be used to extrapolate more characteristics of the engine and to create a design of experiments (DOE) for the actual engine calibration effort.

The physical model was created by taking measurements from the actual engine components. Complicated shapes were approximated with simpler geometries with matching volumes. The model was modified from a GT-POWER four cylinder engine template to suitably represent the engine system geometry and the specific engine features which have been discussed earlier.

The model can be broken up into three main sections, which follow the flow of air from inlet to exhaust. The three model sections are as follows: the air intake system, the engine cylinders, and the exhaust system. The components that make up these sections are described in more detail following.

3.5.1.1 Air Side Model

The overall layout of the air intake side of the model can be seen in Figure 54. For this section the air starts at an ambient condition, flows through an air filter box, through the electronic throttle body, into the intake manifold, through the intake runner / tuning valve system, and into the intake ports.
The air filter box can be seen in Figure 55. The model which was created to represent this component is fairly simple. An inlet port flows into an expanding section that which expands to the size of the air filter. The air filter is represented by a tunable orifice. The orifice discharge coefficient can be tuned to represent the pressure drop through the air filter. After the orifice, there is a converging section which goes from the area of the filter to the area of the outlet. The outlet section goes through a nozzle which is where the
MAF sensor is located in the actual physical hardware. An extra pipe section follows the MAF section and connects to the electronic throttle. Overall the air box section of the model was found to have little effect on the model result since it is upstream of the throttle body.

![Air Filter Box](image)

Figure 55: Air Filter Box

The electronic throttle body can be seen in Figure 56. The stock throttle model from GT-POWER was used for this section of the air model. The model primarily consists of the area of the throttle plate as well as a definition for how the discharge coefficient varies with varying throttle angles. An auto tuning PID controller was added to control throttle angle so the model can be run to meet different MAP pressure set points. The controller thus adjusts the throttle angle to meet the desired MAP pressure for each test point.
Figure 56: Electronic Throttle

The intake manifold’s variable length runner system, shown previously in Figure 39, allows multiple paths for air to flow in the intake based on control logic. The air path can flow through a shorter path by opening a set of tuning valves in the intake or can flow a longer path when the valves are closed. A model of this system was developed by modeling both flow paths and by inserting an orifice to represent the tuning valve system. The model of the discretized intake manifold for each intake runner as well as the model for how the tuning valve connects to the intake runner can be seen in Figure 57.
The intake manifold is modeled by four identical volumes and each volume has four ports. Two ports connect the volumes in series with one another. The other two ports represent the air paths for the long runner system or the tuning valve system. The long runner is a pipe section that connects in a y-junction with the short runner pipe and the tuning valve. The tuning valve is simulated with an orifice connection whose discharge coefficient can be set to one or zero to represent opening and closing the tuning valve respectively.

The intake port model was modified directly from the GT-POWER intake port template. The model consists of a y-junction and two intake port runners with the appropriate
geometry assigned to each component. The fuel injectors are attached to the y-junction to closely model how fuel is injected into the port fuel injected engine.

### 3.5.1.2 Exhaust Side Model

The exhaust side of the model can be seen in Figure 58. The model primarily consists of exhaust ports, an exhaust manifold, a three way catalyst, an exhaust pipe, and a muffler.

![Figure 58: Exhaust Side Model](image)

The exhaust ports which were used in the model are modified directly from the GT-POWER template with the appropriate geometry for the engine. The exhaust manifold for the Honda engine is somewhat unique in that it is directly integrated into the engine’s
head. Thus, the exhaust side of the head acts as a collector with a single port connecting to the catalyst. This can be seen in Figure 59.

Figure 59: Exhaust Manifold

Without a typical exhaust manifold, the head integrated manifold was modeled by using a single y-junction with four inputs, one from each exhaust port junction, and one outlet which leads to the catalyst. This can be seen in Figure 60. The effective length was set differently for the inner and outer cylinders to represent the differences in length from the cylinders to the exhaust outlet.
The exhaust catalyst and catalyst model can be seen in Figure 61. The model consists of three primary sections. The inlet is modeled as a y-junction. The catalyst model consists of the two sections of the catalyst with a y-junction that represents the empty volume between the two catalyst bricks. The outlet of the second catalyst brick is connected to another y-junction which connects to the exhaust pipe. The catalyst section of the model is modified from the stock GT-POWER catalyst to represent the actual cell density of the brick.
The remainder of the exhaust system is based purely on the stock GT-POWER model template. The exhaust pipe was modeled based on the diameter of the catalyst outlet and the approximate length of a vehicle. The muffler model was directly copied from the template. A final exhaust orifice connects the system back to ambient conditions so that the back pressure of the exhaust system can be tuned if necessary.

3.5.1.3 Other Models

The cylinder geometry information was copied from documentation about the Honda engine in both [20] and [31]. The valve lift profiles of the model were generated based on information in [31], shown in Figure 62. This figure shows how the delayed closure
cam remains open well into the compression stroke of the engine. This reduces the effective compression ratio of the engine as well as the volumetric efficiency.

Figure 62: Valve Profiles [31]

The heat release profiles for the combustion section of the model were generated based on information in [31], shown Figure 63.
Once the model was assembled based on physical measurements and documentation about the engine, simulations were run to calibrate the model and to gain insight into the dynamics of the engine. The calibration was performed by running the model first with CNG as the fuel for the engine. This was done since the most detailed experimental results from [31] were performed using CNG. This section will outline how the model was calibrated with CNG and how the model was then run on E85 to better understand the characteristics of mode selection in the engine control strategy.

3.5.1.4 1-D Model Calibration

Some of the torque curves from [31] can be seen in Figure 64. These torque curves show how the engine performs with the high output cam selected and with the tuning valve turned on only at high speeds. It can be seen that significant gains in torque can be found
at high engine speeds when the tuning valve is opened. Other curves were referenced from the paper to look at the performance of the engine during delayed valve closure.

A primary goal of tuning a GT-POWER model is to match the “tuning peaks” which show up in the maximum torque curve of the engine. As briefly described earlier, these peaks and valleys in the maximum torque curve are directly related to the speed of engine operation. Varying speed causes pressure pulses in the intake system to come in and out of phase with the intake valve opening period. These pressure pulses can either help or hurt the ability of the engine to bring air into the cylinder. The primary driver of where the peaks and valleys occur in the maximum torque curve is the length of the intake runners.
The lengths of the two sections of runners, the long section and the short section, were varied to make the initial model’s tuning peaks more closely align with the experimental data. The initial model showed that the tuning peaks where generally focused at lower engine speeds than the experimental results. This is indicative of the lengths of the runners being set too long. Since the runner lengths were measured in a rather imprecise method, applying tape to the outside of the intake air path and then measuring the tape length, it was deemed reasonable to adjust the runner lengths to by up to 20 percent shorter.

With the two different sections of intake available for calibration the long section of the runner was calibrated first, this can be seen in Figure 65. The runner lengths are labeled 1 through 5, where 1 is 20% shorter and 5 is the length from the initial model. The other lengths are linearly dispersed between these two extremes. The maximum torque curve from the experimental data is also shown for comparison. As can be seen in the figure, the shorter runner lengths move the peak and valley to higher engine speeds, as was expected. With 20 percent set as a superficial maximum allowable change from the measured length, the shortest length was chosen and then the short runner was also adjusted.
The short runner was adjusted in the exact same method as the long runner and the results can be seen in Figure 66. The minimum runner length was also chosen for this case. The 20 percent decrease in the overall length of the intake shows much better agreement with the experimental results.
A separate calibration was performed on the valvetrain/combustion models by other team members and are not presented in this work. After combining the calibrated version of the physical model and the calibrated version of the valvetrain/combustion model, an overall comparison of the maximum torque curve from the model and from the experiments in [31] could be made, shown in Figure 67. The overall comparison is fairly good considering the level of available data for building and calibrating the model. The data lies mainly within 10% of the experimental results. The simulation data also shows many of the same trends which are seen in the experimental data. The tuning peaks occur in very similar locations and the tuning valve performs its function accurately in the model by increasing the maximum torque at high engine speeds.
3.5.1.5 1-D Model Results

A DOE was made in GT-POWER and run using the validated model. In order to better understand the actual volumetric efficiency and torque of the converted engine, the model was redefined to run on E85. The DOE was structured to vary manifold air pressure (MAP) and engine speed. This was performed with 10 different MAP pressures across 20 different engine speeds from 2000 rpm to 7000 rpm. This DOE was performed four times, once for each possible operating mode of the engine.

The DOE post processing tool which is built into GT-POWER was used to extract relevant data from the DOE. The volumetric efficiency vs. torque and speed for each of the engine modes was plotted. These graphs can be seen in Figure 68, Figure 69, Figure
70, and Figure 71. The high output cam can be seen to produce very high volumetric efficiencies at high loads, some even exceed 1.

![Volumetric Efficiency vs Torque & Speed](image)

**Figure 68: Volumetric Efficiency (High Output, TV Off)**

It can be seen that turning the tuning valve on in the high output condition significantly increases the speed at which max torque occurs. However; the actual maximum torque and volumetric efficiency is lower with the tuning valve on. Although not shown in these plots, the tuning valve results in large power production since the torque remains much flatter through high engine speeds.
The effect of the delayed closure cam is very significant. The maximum torque of the engine is significantly decreased and the maximum volumetric efficiently is also greatly reduced.

Figure 69: Volumetric Efficiency (High Output, TV On)

Figure 70: Volumetric Efficiency (Delayed Closure, TV Off)
The tuning valve exhibits the same trend with the delayed closure cam that was shown with the high output cam. The location of peak torque moves to high speed and the overall torque curve remains flatter.

Figure 71: Volumetric Efficiency (Delayed Closure, TV On)

Another method to look at these trends is to look at how torque varies versus MAP and speed for the different engine operation modes. These plots can be seen in Figure 72, Figure 73, Figure 74, and Figure 75.
These figures reveal how maintaining the same MAP pressure and turning the tuning valve on shifts the lines of constant torque to higher speeds.

Figure 72: Torque Production (High Output, TV Off)

Figure 73: Torque Production (High Output, TV On)
A more significant trend that is exhibited by this type of plot is the difference between torque production in the high output and delayed closure valve settings for a given map pressure. The torque production is much lower for the delayed valve closure with the same map pressure because of the drastically reduced volumetric efficiency.

Figure 74: Torque Production (Delayed Closure, TV Off)

The tuning valve is also shown to significantly change the torque production of the delayed closure cam as it also greatly affects the volumetric efficiency of the engine.
The primary goal of the early GT-POWER model was to better understand how the different possible operating modes of the engine should be best used to operate the engine most efficiently. This was done by comparing the brake efficiency for the different operating modes across the entire spectrum of available torques and speeds. The engine mode which yielded the best brake efficiency for a given torque / speed point in the engine map was chosen and plotted in Figure 76.
In general the area where each cam is most efficient is intuitive. However, the areas where the tuning valve should be turned on seem less intuitive. The majority of the map suggests that the tuning valve should be turned on with only a few areas requiring the tuning valve to be turned off.

The results from Figure 76 were summarized into generalized regions of operation in Figure 77. The model predicts that it is only necessary to turn the tuning valve off to gain torque in the mid-speed / high load section of the engine operation map. A narrow band of tuning valve off may also be favorable in the lower speed section of the delayed closure cam operation. The tuning valve off section which is predicted at high speed is likely unnecessary since the engine control will avoid operation in this region.
The previous figures were created by choosing the best efficiency mode at each torque and speed engine operating point. No criteria for a minimum difference in efficiency was set to decide whether or not to change modes. This makes the borders of the operation zones fuzzy when one considers the impact that could be seen from air transients generated during mode switching. Benefits in efficiency may be cancelled by losses in air charge uncertainty. This uncertainty during mode switching can increase emissions output and make it less desirable to switch modes often. Since the previous figures reveal no information about the net improvement in efficiency, a histogram plot of single mode and multiple mode engine operation efficiencies was generated, shown in Figure 78.
Single Mode Operation

Multi Mode Operation

Figure 78: Simulated Efficiency, Single vs. Multi Mode Operation

Plotting the change in efficiency in this manner reveals a slight shift in the distribution of efficiencies to higher values. The effect is significant but is small enough that the team could initially calibrate the engine with only single mode operation without a tremendous loss of efficiency, thereby reducing the initial calibration effort.

After the test engine was set up in the dynamometer lab the results from this section were revisited for validation purposes. A course grid of engine operation points, 10 kPa MAP increments and 500 RPM engine speed increments, was run on the dynamometer test bench for both high output cam and delayed closure cam modes. Processing the results and choosing the best efficiency modes at different engine operation points revealed a similar trend to what was found in the GT-POWER simulation, shown in Figure 79.
In general the results for both the simulated and experimental operation zone maps are very similar. The primary differences are from differences in the peak predicted torque of each mode. The experimental results show a high peak torque curve for the delayed closure mode and a lower peak torque curve for the high power mode. These two factors lead to a much narrower zone where high output cam operation is favorable. The overall
effect of the multi mode operation from experimental results is shown by the histogram plots in Figure 80.

![Histogram plots of Single Mode Operation and Multi Mode Operation](image)

Figure 80: Measured Efficiency, Single vs. Multi Mode Operation

Similar to the simulation result, it is shown that a noticeable increase in the efficiency of the engine is found with the multiple operation modes. The plots also show that good efficiency is found with the single operation mode as well. This motivated the team to avoid the complexity of tuning multiple engine operation modes in year two. The team focused efforts on tuning the single mode operation with the possibility of revisiting other modes after the engine provides full performance with a single mode.

### 3.5.2 Advanced GT-POWER Model

A more advanced GT-POWER simulation model was later generated by a visiting scholar at Ohio State, Davide Vezza. Although this work was completed by the author, a description of the model and some of the model results are included for completeness. This more advanced model captured many engine performance characteristics.
significantly more accurately and reveals some of the potential of 1-D modeling techniques.

The more advanced model, pictured in Figure 81, used advanced tools from Gamma Technologies to translate 3D geometry models of the actual intake into 1-D approximate volumes. In addition, valve discharge coefficients for both intake and exhaust valves were measured using flow bench equipment at the Ohio State Center for Automotive Research. The increased accuracy in the raw model structure and initial calibration values allowed for simpler model calibration to match experimental results.

Figure 81: Advanced GT-POWER Model Layout [29]

After synthesizing and calibrating the advanced model, plots were generated which reveal how the model more accurately predicts the Brake Mean Effective Pressure (BMEP) at
full and part load conditions. These plots are shown in Figure 82 and Figure 83 respectively.

Figure 82: Full Load BMEP Model Validation [29]
Utilization of some advanced features of GT-POWER allows for model reduction from the detailed GT-POWER model to a mean-value representation. This model uses single volumes to represent the intake and exhaust and estimates combustion using a neural network combustion model which was trained by the detailed model. A transient simulation was performed, shown in Figure 84, with both models and it was shown that the mean-value model gives similar results to the detailed model with much less computation time. This reinforces the concept presented in the modeling process section where advanced models can be used to calibrate simpler models and the resulting group of models can be used to learn more and guide future experiments more effectively than any individual model.
3.6 Mean Value Models

The mean-value model development was hastened by the donation of the dSPACE Automotive Systems Models (ASM) following the year one competition. The team was provided with a variety of automotive simulation tools. For this work, the ASM Gasoline Engine Simulation Package was used. This simulator consists of a configurable mean-value engine model with the option of combustion torque modulation. Other models were made available which include the ability to model in-cylinder processes; however, the team determined that the level of detail was not justified considering the extra effort which would be needed to calibrate such a model. The primary goal of the mean-value model was to provide an adequate representation of the engine states required to test the
functionality and fault tolerance of the electronic engine controller on the hardware-in-the-loop test bench.

3.6.1 Modeling Principles

A primary dynamic feature of the mean-value model is the lumping of the intake and exhaust into their own representative single volumes. By using the conservation of mass assumption in these control volumes, a set of flow restriction functions and compressible flow equations can be used to determine the variation of pressure, temperature, and mass flow rate in the intake and exhaust. An example of the sets of coupled equations ASM uses to describe the intake system without considering exhaust gas recirculation is shown in Figure 85.
Figure 85: Connection of Equations for Manifold Dynamics [27]

\[
\dot{m}_{\text{Throttle}} = A(\text{Pos}_{\text{Throttle}}) \cdot \dot{p}_{\text{ambient}}
\]

\[
\dot{m}_{\text{Engine}} = \frac{2}{R} \left( \frac{p_{\text{ambient}}}{p_{\text{Engine}}} \right) \dot{m}_{\text{Throttle}} \left( \frac{\lambda_{A}}{p_{\text{ambient}} \cdot p_{\text{Engine}}} \right) (\text{m}_{\text{Engine}} - \text{m}_{\text{Throttle}})
\]

\[
\begin{align*}
\dot{\text{m}}_{\text{Throttle}} &= c_{p} (\text{m}_{\text{Throttle}} \cdot \text{T}_{\text{Throttle}} - \text{m}_{\text{Engine}} \cdot \text{T}_{\text{Engine}}) - c_{v} \text{m}_{\text{Throttle}} \\
\text{p}_{\text{Throttle}} &= \frac{p_{\text{ambient}}}{\text{V}_{\text{Manifold}}} \cdot \text{m}_{\text{Throttle}} \cdot \text{T}_{\text{Throttle}}
\end{align*}
\]

- \( \dot{m}_{\text{Throttle}} \): Throttle Mass Flow Rate
- \( A \): Throttle Effective Area
- \( \text{Pos}_{\text{Throttle}} \): Throttle Position
- \( \dot{m}_{\text{Throttle}} \): Intake Manifold Pressure
- \( p_{\text{ambient}} \): Universal Gas Constant
- \( R \): Ambient Pressure
- \( T_{\text{in}} \): Ambient Air Temperature
- \( \dot{m}_{\text{Engine}} \): Cylinder Intake Valve Mass Flow Rate
- \( V_{\text{Engine}} \): Displacement Volume
- \( n_{\text{Engine}} \): Engine Speed
- \( \lambda_{A} \): Volumetric Efficiency
- \( i \): Number of Intake Events per Revolution
- \( \text{m}_{\text{manifold}} \): Mass of Air in Manifold
- \( C_{v} \): Constant Volume Specific Heat
- \( C_{p} \): Constant Pressure Specific Heat
- \( T_{\text{Throttle}} \): Throttle Out Air Temperature
- \( T_{\text{Engine}} \): Cylinder Inlet Air Temperature
- \( T_{\text{Manifold}} \): Intake Manifold Air Temperature
- \( V_{\text{Manifold}} \): Intake Manifold Volume
Many other features of the model, like combustion torque and exhaust gas temperatures, are not calculated by a fundamental assumption or equation. Rather, steady state lookup tables coupled with filters are used to simulate the dynamics of various engine states.

### 3.6.2 Model Calibration

The dSPACE ASM mean-value engine model is most easily calibrated by using the ASM Parameterization tool. This parameterization tool consists of a set of Matlab scripts which are used to convert steady state engine data into the variables required to run the mean-value model. Although a majority of the variables are calculated by these scripts, some physical properties like engine displacement and intake manifold volume must be entered manually as they cannot be derived from a set of steady state engine tests. The parameterization tool reads in data from a Microsoft Excel file which must be formatted with the variable name in the first row and the variable units in the second row. All subsequent rows are considered to be individual steady state test points. 37 variables from each test point must be stored in the columns of the spreadsheet. Values ranging from engine speed and manifold pressure to turbine speed and intercooler temperature must be entered for all models, even if some features of the model like the turbocharger have been disabled. If the features have been disabled, the data can be filled in with invalid data and the model performance will not be effected; however, since the model is built with a “universal” structure the variables will still be called when Simulink starts. An example of a small section of a model calibration spreadsheet is shown in Figure 86.
Figure 86: Sample of Mean-value Model Calibration Data

The inclusion of variable name and units in the spreadsheet allows the user to use a data linking feature in the parameterization toolbox. This gives the user the ability to tie measured variables to specific model variables which may have different names and units. Unit conversion is performed automatically for many recognized units like RPM to rad/sec. Other unit conversions can be manually input and will be recognized automatically in future model calibrations. After linking the data from the spreadsheet to specific model variables, the parameterization tool can use pre-defined Matlab scripts to calculate non-measured variables, i.e. the throttle effective area curve. For ease of use the parameterization tool has been built with a GUI shown in Figure 87.
While running the parameterization, the scripts automatically generate a variety of plots which can be used to understand how well the parameters have been fit. An example of the throttle effective area versus throttle angle parameter is shown in Figure 88.
While the throttle effective area curve plays a major factor in determining air flow into the intake manifold, the volumetric efficiency of the engine determines the air flow from the manifold to the cylinder, parameterization shown in Figure 89.
The volumetric efficiency parameterization brings up a potential problem with using the ASM model. Many of the test points shown on the parameterization surface lie far away from the surface. This makes it obvious that the model uses some type of smoothing function and/or performs a best fit to a functional form. While much of the ASM model is described well in the help files of Matlab, if the user decides to fit a different functional form they must do one of two things, modify the parameterization code or process the data separately and include the results in the final parameterization. The first option can be daunting even with well documented code as it can take a large amount of time to become familiar with someone else’s pre-written code. The second option is complicated by the fact that ASM decided not to use MAT files to store parameterized data. Rather, the parameters are written as in-line code in model initialization Matlab scripts. This is somewhat straightforward for 1D lookup tables but becomes more complicated with higher dimensional lookups.
After predicting the amount of air going into the cylinder, the model predicts the torque output of the engine as a combination of Indicated Mean Effective Pressure (IMEP) and Friction Mean Effective Pressure (FMEP). Separating torque into a summation of these two torques allows for relative modification of both IMEP and FMEP separately. This can be used to account for torque differences as functions of engine coolant temperature, in-cylinder equivalence ratio, spark timing, etc. As would be expected, a very linear fit of relative airmass to IMEP is shown in Figure 90.

![Mean indicated Pressure = f(Engine Speed, Relative Airmass)](image)

Figure 90: ASM Parameterization IMEP Map

A non-linear fit of FMEP is shown in Figure 91. This reveals the expected trend of increased friction at higher engine speed and lower engine temperatures.
After predicting the mean-value of engine torque for an engine cycle the user has the option to use this value or to use a modulated version of the mean torque. The modulated torque is accomplished by utilizing a torque shape function shown in Figure 92 which scales the mean-value torque as a function of crank angle. This shape function is designed in such a way that the summation of the individual cylinders’ modulated torque results in the same mean-value torque for the cycle. In this way the engine speed fluctuations due to individual cylinder combustion or even misfire can be approximated without the computational burden of an actual combustion model.
Additional lookup tables like the exhaust gas temperature are important in reproducing the types of signals an electronic engine controller expects to see without entering a fault mode, shown in Figure 93.
For software-in-the-loop testing the ASM model also calculates a variety of parameters which can be used by a simulated engine controller, a “Soft ECU.” This simulated controller allows the user to check whether the model provides reasonable behavior before moving the model to the hardware simulator. The “Soft ECU” can use parameters like the indicated torque to relative air mass fit, shown in Figure 94, to perform the torque based throttle positioning required by many modern engines.
Figure 94: ASM Parameterization Relative Air Mass vs. Torque Request Fit
3.6.3 Real Time Model Implementation

The functional software-in-the-loop mean value engine model can be combined with the EcoSIM hardware-in-the-loop simulator model. The functionality of the EcoSIM simulator is maintained by running the mean value model in parallel with EcoSIM’s black box engine model. This allows the black box model to interact with the vehicle controller and the mean value model to interact with the engine controller. The connection of the controller variables to mean value variables is facilitated by the hardware signal conditioning built into the hardware simulator. An example of the types of signals which are connected through this interface is shown in Figure 95.

![HIL Engine Simulation Connections](image)

Figure 95: OSU HIL ECU Connections

Many of the signals are handled properly by the analog, digital, and resistive inputs of the hardware simulator. These signals are sampled at the rate of real time model execution.
However, some of the signals operate at high frequencies which cannot be captured by the model execution time. These high speed signals are handled by using an embedded time processing unit (TPU). The TPU reads the engine speed from the mean value model and generates a high speed engine position vector. The position vector is used to generate pulse trains which replicate the crank and cam encoder signals. These encoder signals allow the engine controller to estimate the speed and position of the engine model. The TPU receives high speed fuel injection and spark ignition signals from the ECU and interprets the duration and positioning of these signals. The fuel injection and spark timing parameters are then passed from the TPU to the mean value model at the model execution rate. This process is shown in Figure 96.

![Figure 96: HIL Time Processing Unit](image-url)
For debugging purposes a mean value model interface was developed with the dSPACE ControlDesk software, shown in Figure 97. This HIL interface can be used to read signals from the ECU, *i.e.* fuel injection, spark timing, and relay status, and signals from the mean value model, *i.e.* MAP, MAF, and torque production. Although the primary purpose of the HIL mean value model is automated fault testing, this user interface provides useful functionality during model and test development.

As has been discussed previously, the primary purpose of the HIL mean value model is to provide a repeatable test platform for checking the fault tolerance of the engine control algorithm. This testing is conducted with the dSPACE AutomationDesk software. This software allows the user to develop repeatable test plans which can be performed batch wise on the hardware simulator. A preliminary set of test plans were developed to
demonstrate the functionality of the system, shown in Figure 98. Test plan development consists of programming in a flow chart format. The first steps carry out the basic functions required to start the simulator and initiate normal engine operation. Next, relevant variables are recorded to ensure the engine is operating normally. A fault is then inserted and relevant variables are checked to see if the engine controller has made the expected adjustments to the assigned fault. Finally a report is automatically generated to summarize the results of the tests.

<table>
<thead>
<tr>
<th>Oil Pressure Fault</th>
<th>Engine Overspeed Fault</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Start Engine</strong></td>
<td><strong>Start Engine</strong></td>
</tr>
<tr>
<td><strong>Check Fuel Relay</strong></td>
<td><strong>Check Fuel Injectors</strong></td>
</tr>
<tr>
<td><strong>Insert Oil Pressure Fault</strong></td>
<td><strong>Insert Overspeed Fault</strong></td>
</tr>
<tr>
<td><strong>Check Fuel Relay</strong></td>
<td><strong>Check Fuel Injectors</strong></td>
</tr>
<tr>
<td><strong>Create Report</strong></td>
<td><strong>Create Report</strong></td>
</tr>
</tbody>
</table>

Figure 98: Fault Insertion Tests

The automated testing feature is important because it allows future versions of engine control software to be validated, confirming the fault tolerance of the previous software revision is maintained or improved. The automated report generation feature is important because it provides a quick reference of which tests the engine controller has either passed or failed. An example of an automated test report is shown in Figure 99 where a quick summary of the test results is given at the top and more detailed test specific results are provided at the bottom.
<table>
<thead>
<tr>
<th>Test Case</th>
<th>Start Time</th>
<th>Execution Duration</th>
<th>Stop Time</th>
<th>Name</th>
<th>Creation Date</th>
<th>Author</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine OPSW Low Fault</td>
<td>2010/08/05 00:33:30</td>
<td>57.217 sec.</td>
<td>2010/08/05 00:34:28</td>
<td>EcoCAR_Fault_Simulation_Engine_2010-08-02</td>
<td>2009/11/17, 10-29-00</td>
<td>EcoCAR</td>
</tr>
<tr>
<td>Engine Overspeed Fault</td>
<td>2010/08/05 00:33:31</td>
<td>34.937 sec.</td>
<td>2010/08/05 00:34:05</td>
<td>EcoCAR_Fault_Simulation_Engine_2010-08-02</td>
<td>2009/11/17, 10-54-34</td>
<td>EcoCAR</td>
</tr>
</tbody>
</table>

Low Oil Pressure detected, fuel relay shutdown

Overspeed detected, fuel injection cut
CHAPTER 4

ELECTRONIC THROTTLE CONTROL

4.1 Introduction

Electronic throttle control is an important subject for the EcoCAR team because the chosen engine comes equipped with an electronic throttle. The electronic throttle is normally controlled by the stock Honda ECU. As discussed earlier, the replacement of the stock ECU for a rapid prototyping unit requires the replication of many control functions including electronic throttle positioning.

The recent push to significantly increase the efficiency of spark ignited engines has lead to the inclusion of engine technologies like exhaust gas recirculation (EGR), variable valve trains (VVT), variable geometry intakes and direct fuel injection. These technologies have forced vehicle manufacturers to switch to electronic throttles. Many engines that do not include or do not heavily rely on these technologies have used Bowden cable controlled throttles. These throttles provide a direct connection from the vehicles accelerator pedal and the throttle plate. This connection is made with a Bowden cable, a sheathed one way pull cable, where pushing the accelerator pedal pulls the cable and opens the throttle. Releasing the pedal allows a spring built in to the throttle body to return the throttle to a lower position. Vehicles with these systems sometimes include an
extra cable and/or actuator to control the throttle for the purpose of cruise control. Fine air adjustment for idle control is typically achieved with a secondary controlled air inlet known as an idle air controller.

The direct connection of the accelerator pedal to the throttle valve causes a direct correlation between the pedal position and the amount of air entering the engine. The advanced engine technologies discussed before can affect the amount of combustible air entering the engine without adjusting the throttle position, which affects the torque and power production of the engine. Thus, when enabling one of these technologies the throttle must automatically adjust position to maintain the correct amount of air flow and torque production so as to not disturb the driver. This is a major shift in the idea of engine control where older engine controls must look at changes in throttle position as a disturbance and adjust to function with a given airflow. Modern engines must look at changes in pedal position as a disturbance to a torque request and modify the engine operating modes and throttle position to achieve the requested torque.

Electronic throttles are mechanically similar to cable actuated throttles; however, their movement is actuated by an electric motor instead of a cable and cam system. Both types of throttles include throttle position detection systems. In cable actuated systems, the throttle position sensing is used primarily for feed-forward air prediction as well as for safety systems like fault detection. In addition to these functions, throttle position sensing is used in electronic throttle control as a feed-back for a control system that varies voltage to the electric motor to achieve a desired position. A diagram of the electronic throttle provided with the Honda engine can be seen in Figure 100.
The ability of the electronic throttle to vary its position and the resulting airflow into the engine is an important feature that not only allows for the inclusion of advanced engine technologies but also allows the removal of special systems for cruise control and idle air control. The technology is also an essential part of hybrid vehicles because it allows for the decoupling of driver torque request and engine torque production.
4.2 Literature Review on Throttle Control

Many different types of literature were researched to understand the necessary features and the state of the art in developing electronic throttle controls. This section outlines a wide variety of topics related to electronic throttle control.

Like many systems in a vehicle, electronic throttle control is considered a safety critical system. This means that a failure of this system or deviation from expected behavior in this system could lead to safety issues that vary from somewhat mild problems like engine surge to extremely dangerous situations where the throttle sticks wide open. These issues are especially important and widely discussed in part because of the heavily publicized problems that caused Toyota to recall millions of vehicles in 2010 because of safety issues with their electronic throttles.

One of the most important safety critical functions in the electronic throttle is the ability to accurately sense the position of the throttle plate. This not only allows for the feedback positioning of the throttle but also for the detection of unexpected behavior. The diagram shown in Figure 101 illustrates how the signal conditioning is performed in General Motors’ electronic throttles to insure extra levels of safety. A key safety features is that two throttle position sensors are used. These two sensors have different slopes and intercepts which relate throttle position to a voltage output. This allows for error checking between the two sensors, checks for shorts to sensor supply voltage and sensor ground, as well as for shorts between the two sensors. This detection system is very similar to the that found in the Honda electronic throttle.
Another important topic which is discussed in [8] is the subject of de-icing. One fear in throttle systems is the possibility of ice building up in the return path of the throttle. This can cause a variety of problems, from the valve becoming stuck to causing a blockage which prevents the throttle from fully closing. The method of de-icing in [8] is fairly crude and involves using the motor to hit the ice blockage with the throttle plate with the hopes of breaking up and dislodging the ice. The system of throttle de-icing in the Honda system circulates engine coolant through the electronic throttle. Knowledge of this is important because the effects of engine coolant temperature may affect the performance of the electronic control system.

Figure 102 is a diagram of the electronic throttle and accelerator pedal connections to the stock Honda ECU. By moving to the rapid prototyping controller these connections must be replicated. They include sensor supply voltage, sensor ground reference, throttle
positions sensor 1 output, throttle positions sensor 2 output, and two connections for providing voltage to the electric motor in the throttle. This voltage source must be attached to an H-Bridge in the new ECU so that the polarity of the voltage is switchable. When combined with a pulse width modulated (PWM) output this allows for bi-directional variable magnitude current to drive the throttle open and closed with varying torque.

Figure 102: Honda Civic Accelerator Pedal Position and Throttle Position Sensors Diagram [20]

A mathematical model of an electronic throttle for the purpose of developing a sliding mode controller is developed in [24]. This paper also provides a good diagram of the key components which make an electronic throttle, this diagram can be seen in Figure 103.
The diagram of the electronic throttle can be used to summarize the majority of the forces that describe an electronic throttle. By tracking the actuation path from system input to motion of the throttle plate, the system can be described as follows. Variable voltage is applied as the input to an electric motor by the uses of an H-Bridge PWM signal. Assuming the PWM switching frequency is high enough the input can be approximated as a true variable voltage input. This voltage interacts with the resistive and inductive circuitry in the motor and the backward electromagnetic field voltage due to the motor’s movement to induce a current in the electric motor. As current varies in the motor so does the torque output by the motor to the pinion gear.

The pinion gear translates it’s torque through an intermediate gear to a sector gear attached to the throttle plate. This gear connection provides the opportunity for backlash in the system as well as for additional inertia/friction. The sector gear applies its torque to
a shaft that is connected to the throttle plate and to a non-linear spring. The non-linear spring has a discontinuity at the throttle’s “limp home” position. This is the position that the throttle should settle to with no torque input from the motor and allows the engine to operate at a fixed condition in certain failure modes. Since the spring does not return to a fully unbound condition at the discontinuity, there is also a certain amount of pre-load in the spring. The discontinuity thus provides positive torque above a certain set point and negative torque below that set point. Where positive torque is considered to close the throttle and negative opens the throttle, this illustrated in Figure 104.

![Non-linear Spring Torque vs Position](image)

**Figure 104: Non-linear Spring Torque vs Position [24]**

An example of how a Matlab/Simulink model can be generated to represent the previously generated dynamics is presented in [33]. This model was generated by Visteon Corporation, which is an automotive supplier for a variety of automotive control.
products. One feature missing from this model is the backlash of the gears which has been shown to be important by another automotive supplier, Mikuni, in [16]. A Simulink diagram of the Visteon model can be seen in Figure 105.

Figure 105: Visteon Corporation Simplified Plant Model of Electronic Throttle [33]

Two major problems with electronic throttle modeling and model based control that become apparent in the literature and also from initial experimentation with the throttle are the non-linearity of the system and uncertainty in the plant model parameters. Non-linearity makes it difficult if not impossible to use model based control designs that are meant for linear systems. The large amount of plant parameters can make it difficult to identify the system with system level tests because many of the parameters have similar affects on the response of system. For instance, the speed of the throttle can be effected by the torque production of the throttle, the resistance of the spring, the inertia of the system, etc. Component tests should reduce the identification problem and make
identification possible. Because of this many of the high fidelity throttle models presented in the literature have been calibrated by throttle suppliers whom have access to more detailed information and testing of the sub-components in the throttle.

One way to get around the difficulties of electronic throttle modeling is to avoid the modeling step and to formalize a calibration procedure which is performed on the actual plant. A method for controller tuning which has been presented in the literature is iterative tuning feedback. This method has been shown in [19] to work well in systems that are difficult to model. The method has also been presented for the calibration of control systems for safety critical systems which cannot be run in open loop to perform system identification. Although the method was primarily designed to work with linear systems, it has been shown in the literature to work well for some non-linear systems. In particular, it was shown in [22] to work well with electronic throttle control. Figure 106 shows a closed loop system diagram from [22] in which the control parameters of $G_1$, proportional and integral control, and $G_2$, which the authors call a damping control, were optimized to give the best response for the throttle acting as the plant $G_p$. 
After applying the iterative tuning feedback method over multiple iterations the authors were able to achieve a significant improvement in the closed loop system response. This can be seen in Figure 107.

This figure brings up another important question with electronic throttle control. What type of response is desirable? Reviewing the literature, specifically [16], [22], and [33],
reveals that the preferred response in industry is a first order response to a step change. The response should have a relatively fast time constant, nominally 50-100 milliseconds, such that the throttle achieves steady state position within 0.25-0.5 seconds. As an example of how this type of tuning performs on a production controller, Figure 108 shows how Mikuni’s controller responds to a variety of inputs. [16] also confirms that the response of the controller changes when it is run on low and high performance processors. Running the control at a slower clock speed revealed an increase in step input rise time of 7%. Since many of the controllers in the literature use clock periods of 1-4 milliseconds the EcoCAR team should expect a slightly slower response because the rapid prototyping controller must run a 7 millisecond clock to handle the engine controls computational overhead.
Another type of system response is suggested in [33] where the primary control trajectory is a first order response but the steady state positioning adds a dither cycle. The idea of the dither in this control strategy is to negate some of the variability due to static friction and gear backlash. By dithering the position it decreases the chance that the throttle will be requested to step change its position from a static position. As long as the dither cycle is sufficiently shallow and high enough frequency the intake manifold dynamics should act to filter the effect of the dither on the steady state air intake through the throttle. This control strategy can be seen in Figure 109.

Figure 108: Mikuni Corporation Tuned Electronic Throttle Control Response [16]
4.3 Electronic Throttle Control Formulation

4.3.1 First Principles Model

Initially a first principles model was pursued in order to better understand the order and dynamics of the system. A diagram of the idealized throttle system can be seen in Figure 110.
This simple representation of the throttle assumes that the system mostly functions as a DC motor. The input to the system is a voltage and the output of the system is a throttle angle. The internal dynamics between the throttle and DC motor are lumped together. This linearized system accounts for the inertia of the motor and throttle, any damping from the motor and the friction of the throttle plate and the spring in the throttle. An equation is use to relate how voltage to the motor relates to current through the motor, this is based on a LR circuit approximation. A second equation uses a simple gain factor to relate motor current to motor torque. These two governing equations are shown in equations (9) and (10).

\[
\begin{align*}
(J_m + J_t) \ddot{\theta} + B_m \dot{\theta} + K_t \theta &= K_m i_a \\
L_a \frac{di_a}{dt} + R_a i_a &= V - K_e \dot{\theta}
\end{align*}
\]

These equations can be represented as equations of state in (11) and (12).
Without an easy way to obtain values for the factors used in these equations, this method was not pursued past this point. However; the knowledge that the associated dynamics of the system should roughly fit a third order system was used for system identification.

4.3.1.1 System Identification

Without knowledge of the system parameters, a test was formulated to attempt to identify the system. A chirp signal was created and implemented on the prototype controller which controls the throttle. This signal was chosen so that the system can be identified with varying frequency components. The base chirp signal can be seen in Figure 111.
A gain was also applied to the signal so that the amplitude of the signal could be varied. Two experiments were performed, each with a different amplitude. This allows one test to be used for system identification and the other for validation. In order to identify the system, the System Identification Toolbox in Matlab was used. A screenshot of this tool and implementation can be seen in Figure 112.
The system identification tool allows the user to import test data and assign variables as input and output. The user can then automatically generated state space systems which fit the results with minimum error. This allows the user to quickly analyze which order system fits the data most accurately. The best state space fit for the test data was found with a third order system. This is reassuring since the desired linear model was designed to be a third order system.

The system identification toolbox uses the notation in equations (13) and (14) to reference the form of the identified state matrices.
With this notation the best fitting state matrix values are summarized in Figure 113, the fit of the system is shown in Figure 116.

\[
x(t + T_s) = Ax(t) + Bu(t)
\]

(13)

\[
y(t) = Cx(t) + Du(t)
\]

(14)

By examining the identified state equations from Figure 113 it can be noted that they do not match the functional form identified in equations (11) and (12). This is because system identification creates state equations that match the data but do not necessarily relate to physical states.

\[
A = \begin{bmatrix}
0.80933 & 0.050053 & 0.21104 \\
-0.19464 & -0.96759 & 0.4027 \\
0.53048 & -0.29409 & 0.034731
\end{bmatrix}
\]

\[
B = \begin{bmatrix}
0.093518
\end{bmatrix}
\]

\[
C = \begin{bmatrix}
29.64 & -2.3484 & 7.8396
\end{bmatrix}
\]

\[
D = \begin{bmatrix}
0
\end{bmatrix}
\]

Figure 113: Identified State Space Parameters
4.3.1.2 System Analysis and Design

After identifying the system, the controllability and observability of the system was evaluated by calculating the rank of the controllability and observability matrices with equations (15) and (16) respectively.

\[
CRrank = rank\begin{bmatrix} B & AB & A^2B \end{bmatrix}
\]  

(15)

\[
OBrank = rank\begin{bmatrix} C & CA & CA^2 \end{bmatrix}
\]  

(16)

Both equations were calculated to be full rank and thus the system can be said to be controllable and observable. The system was then input into Simulink with a discrete state-space block so that its system performance can be compared to the actual system, the implementation is shown in Figure 114.

![Simulink State Space Implementation](image)

Figure 114: Simulink State Space Implementation
Three types of signals were used to compare the response of the simulated system to the actual system: a step input, the fitted data input, and the validation data input. The results can be seen in Figure 115, Figure 116, and Figure 117 respectively.

Figure 115: Step Response Comparison

Figure 116: Fitted Data Chirp Response Comparison
Figure 117: Validation Data Chirp Response Comparison

From this it can be seen that the simulated system gives a descent representation of the actual systems dynamics. The representation is worse at higher frequencies but this is likely due to problems with data sampling. The data sampling in this experimental setup was relatively slow and aliasing issues occur at higher frequency throttle oscillations. Also, the actual throttle has the tendency to move its mean-value operating point throughout the experiment because of the non-linearity of the system. Because of this the simulated result is shifted by the actual systems mean-value to give a better comparison. Since this non-linear system was fit with a linear system for simulation, the model was deemed to be fairly good.

Two different controls were pursued for the throttle, a linear quadratic regulator (LQR) and a proportional, integral and derivative (PID) controller. Where the LQR controller
was designed base on the linear state equations and the PID controller was tuned with the iterative tuning feedback method on the actual throttle.

The LQR control was chosen because of the ease of tuning. The LQR provides for quick tuning of on the desired response because the various weighting factors make it possible to adjust different model response variables in a logical fashion. Specifically for this digital system the ‘dlqr’ command in Matlab was used to generate desired poles based on the Q and R matrices. Initially the system was modeled with full state feedback and the Q and R matrices were tuned to find a desirable result.

The DC gain of the system was set for each resulting set of controller gains in order for the system to settle to the correct value based on a varying input. A plot of how the throttle model responds to a step-in and step-out signal can be seen in Figure 118. The smaller magnitude signals represent the states which are calculated by the identified system, these do not necessarily have physical meaning but they mathematically make the system work.
Figure 118: Full State Feedback LQR Step Response

This response was found with the following Q and R matrices with the resulting gain vector $K$ as seen in Figure 119.

$$Q = \begin{bmatrix} 10^4 & 0 & 0 \\ 0 & 1 & 0 \\ 0 & 0 & 1 \end{bmatrix} \quad R = \begin{bmatrix} 10^{-3} \end{bmatrix} \quad K = \begin{bmatrix} 8.6406 & 0.5381 & 2.2519 \end{bmatrix}$$

After the tuning of the full state feedback controller, an observer based control was also tuned because the states of the system are not measureable. Both the gain and the observer matrices were calculated with their own independent Q and R matrices. The system was simulated with the model in Figure 120.
Because an observer does not start with the actual state value it must be proven that the observer calculated state converge to the actual states in finite time. Figure 121, Figure 122, and Figure 123 illustrate that the observed states do converge quickly to the actual state values.

Figure 121: State 1 Observer State Convergence
This response was found with the following Q and R matrices with the resulting gain vector K and observer vector L shown in Figure 124 and Figure 125 respectively.
\[ Q = \begin{bmatrix} 10^4 & 0 & 0 \\ 0 & 1 & 0 \\ 0 & 0 & 1 \end{bmatrix}, \quad R = \begin{bmatrix} 10^3 \end{bmatrix}, \quad K = \begin{bmatrix} 1.0158 & 0.0535 & 0.2488 \end{bmatrix} \]

Figure 124: Observer Based Feedback LQR and Gain Matrices

\[ Q = \begin{bmatrix} 1 & 0 & 0 \\ 0 & 10^3 & 0 \\ 0 & 0 & 1 \end{bmatrix}, \quad R = [1], \quad L = [-0.0111, 0.3167, 0.1044]^T \]

Figure 125: Observer LQR and Observer Gain Matrices

4.3.1.3 Simulation Results and Discussion

The observer based model was created and used to tune the controller and observer Q and R matrices. A representation of the response from one of the better tuned models can be seen in Figure 126.
The response of the system is slower than the desired first order response with a 0.1 second time constant. However, the model does reach the desired values. The calibrated control was next migrated into the actual prototype control hardware. Comparison of the test results and the simulated results are shown in Figure 127.
Although the control seems to give a reasonable result in this limited case, the control is far from implementable. The dc gain on the prototype control must be adjusted to match individual cases and this varies based on current position and desired position. One possible solution to this problem would be the inclusion of an integral control with the LQR controller.

4.3.2 Iterative Feedback Tuning Method

Because of the problems with the model based control model identification and control implementation the iterative tuning method was used to calibrate a PID controller with the actual system serving as the plant.
4.3.2.1 Mathematical Formulation

This section outlines the mathematical formulation for the iterative tuning feedback problem of locating “optimal” gains for an electronic throttle PID controller. The formulation follows directly from the method presented in [22], which presents a direct application of iterative tuning feedback to electronic throttle control.

Starting with the control structure shown in Figure 106, equations (17) and (18) can be written to represent the closed loop system.

\[ u = G_r r - (G_1 + G_2) y \]

\[ y = G_p u \]

Iterative tuning feedback is an iterative optimization routine, thus an appropriate criterion function must be chosen in order to analyze the optimality of the control system. The criterion function shown in (19) evaluates the value of the criterion function \( J \) as a function of the squared error \( e \) and the time for the system to reach equilibrium \( T_f \). The error is defined as the difference between the desired response and the actual response.

As a result of changing the controller parameters \( \rho \) the error will change and the effect of the control changes can be evaluated by looking at the criterion function.
\[ J(\rho) = \frac{1}{2T_f} \int_0^{T_f} e^T(t, \rho) dt \]

(19)

With the desire of minimizing the criterion function the gradient of equation (19) with respect to \( \rho \) can be taken and is represented in equation (20).

\[ \frac{\partial J(\rho)}{\partial \rho} = \frac{1}{T_f} \int_0^{T_f} e(t, \rho) \frac{\partial e(t, \rho)}{\partial \rho} dt \]

(20)

The iterative tuning parameters can thus be calculated as shown in equation (21). This equation uses a step size \( \gamma \) and Hessian \( H \) to determine the magnitude and direction of the change in the parameter vector \( \rho \).

\[ \rho(k + 1) = \rho(k) - \gamma \gamma H^{-1}(k) \frac{\partial J(\rho)}{\partial \rho}(k) \]

(21)

The Hessian \( H \) can be calculated with the approximation shown in (22).

\[ H = \frac{1}{T_f} \int_0^{T_f} \left( \frac{\partial e}{\partial \rho} \right) \left( \frac{\partial e}{\partial \rho} \right)^T dt \]

(22)

The key to this optimization method is to calculate the quantity \( \frac{\partial e}{\partial \rho} \) and is accomplished through closed loop experimentation with iterative tuning feedback. For simplicity, the term \( \frac{\partial}{\partial \rho} \) is represented in the following equations by \( \gamma \). The derivatives of equations
(17) and (18) with respect to the parameter vector \( \rho \) are written as equations (23) and (24)

\[
\begin{align*}
  u' &= G_1 ' r - (G_1 ' + G_2 ') y - (G_1 + G_2 ) y' \\
  y' &= G_\rho u'
\end{align*}
\]

Modification of equation (24) yields the same functional form of equation (17), this is shown in equation (25).

\[
\begin{align*}
  u' &= G_1 \left( G_1 ' r - \frac{(G_1 ' + G_2 ')}{G_1} y \right) - (G_1 + G_2 ) y'
\end{align*}
\]

This form is important because it suggests that if \( \frac{(G_1 ' + G_2 ')}{G_1} y \) is used as the input to the system the output of the system will be \( y' \), shown in Figure 128.
This is important because this term is directly related to $\frac{\partial e}{\partial \rho}$ as is shown in equation (26).

$$e' = r' - y' = -y'$$

(26)

An alternative experiment is proposed where $y$, the output of a normal experiment, is used as the input to a new gradient experiment. The output of this experiment is labeled as $y^*$, shown in Figure 129.
With an assumption of linearity from input to output, which we know is not true because of the non-linear plant, the $y'$ can be calculated with the output of the normal and gradient experiments, equation (27).

$$y' = \left( \frac{G_1'}{G_1} y - \frac{(G_1' + G_2')}{G_1} y^* \right)$$  \hspace{1cm} (27)

From this the $\frac{\partial e}{\partial \rho}$ term can be calculated as shown in equation (28) and Figure 130.

$$e' = r' - y' = -y' = \left( \frac{(G_1' + G_2')}{G_1} y^* - \frac{G_1'}{G_1} y \right)$$  \hspace{1cm} (28)

![Diagram](image)

Figure 130: Calculation of $\frac{\partial e}{\partial \rho}$ [22]

By defining controller $G_i$ as a feedback controller with proportional and integral control, its transfer function is expressed by equation (29).

$$G_i(\rho) = K_p + \frac{K_i}{s}$$  \hspace{1cm} (29)
Defining controller $G_2$ as a feedback controller with a damping term and a low pass filter, its transfer function can be expressed by equation (30).

$$G_2(\rho) = \frac{K_D s}{0.01s + 1}$$  \hspace{1cm} (30)

Assigning the proportional, integral, and damping terms in the following manner: $\rho_1 = K_p$, $\rho_2 = K_I$, $\rho_3 = K_D$, and using equations (29) and (30) allows the transfer functions from equation (28) to be derived. These transfer functions can be seen in equations (31) and (32).

$$\frac{1}{G_1} \frac{\partial (G_1 + G_2)}{\partial \rho} = \begin{bmatrix} \frac{s}{K_p s + K_I} \\ \frac{1}{K_p s + K_I} \\ \frac{s}{s^2} \\ \frac{(K_p s + K_I)(0.01 s + 1)}{(K_p s + K_I)(0.01 s + 1)} \end{bmatrix}$$ \hspace{1cm} (31)

$$\frac{1}{G_1} \frac{\partial G_1}{\partial \rho} = \begin{bmatrix} \frac{s}{K_p s + K_I} \\ \frac{1}{K_p s + K_I} \\ \frac{s}{0} \end{bmatrix}$$ \hspace{1cm} (32)

The formulas derived in this section are complete in terms of estimating required parameters for the iterative tuning feedback method. The equations can be used with a
series of normal and gradient experiments to iteratively improve the performance of the closed loop control system.

4.3.2.2 Implementation

The iterative tuning feedback method was implemented by inputting a step response to the request of the closed loop system using the actual rapid prototyping controller and electronic throttle. The requested position and actual measured positions were recorded, an example of the raw data from a first iteration of a normal experiment is shown in Figure 131.

The raw data was then processed to isolate the 2 seconds following the initiation of the step input, shown in Figure 132.
Next, the processed data from the normal experiment is used as the input to a gradient experiment and the data is processed similarly, shown in Figure 133.

![Figure 133: Isolated Data for Gradient Experiment](image)

The output of the normal experiment is compared to a desired reference trajectory shown in Figure 134.
From this the error is calculated versus time, shown in Figure 135.

The $\frac{\partial e}{\partial p}$ terms can be calculated using equation (28) and the values can be plotted versus time for each of the proportional, integral, and damping terms in Figure 136, Figure 137, and Figure 138 respectively.
Figure 136: $\frac{\partial e}{\partial p_1}$, Partial Derivative of Error with Respect to Proportional Gain vs. Time

Figure 137: $\frac{\partial e}{\partial p_2}$, Partial Derivative of Error with Respect to Integral Gain vs. Time
Figure 138: $\frac{\partial e}{\partial p_3}$, Partial Derivative of Error with Respect to Derivative Gain vs. Time

4.3.2.3 Results

The method outlined previously was used for four different input profiles, a 5% to 20% step, a 5% to 10% step, a 20% to 5% step, and a 10% to 5% step. Each profile resulted in a different set of optimal gains. The relevant calculated optimization variables for the 5% to 20% step input are summarized by Table 5.
Table 5: 5% to 20% Iterative Tuning Result

<table>
<thead>
<tr>
<th>Iteration</th>
<th>Test Gains</th>
<th>$J$</th>
<th>$\frac{\partial J(p)}{\partial p}$</th>
<th>$H$</th>
<th>$\gamma$</th>
<th>Output Gains</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>[5]</td>
<td>1.0881</td>
<td>[0.2523, 0.0342, -2.3545]</td>
<td>[0.1216, -0.0004, -0.6416, -0.0004, 0.0009, -0.0375, -0.6416, -0.0375, 115.8750]</td>
<td>0.25</td>
<td>[4.4426]</td>
</tr>
<tr>
<td></td>
<td>[20]</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>[10.5237]</td>
</tr>
<tr>
<td></td>
<td>[0.1]</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>[0.0989]</td>
</tr>
<tr>
<td>2</td>
<td>[4.4426]</td>
<td>0.6799</td>
<td>[0.3693, 0.0563, -2.7027]</td>
<td>[0.1489, 0.0003, -1.2037, 0.0003, 0.0119, 0.0973, -1.2037, 0.0973, 69.1369]</td>
<td>0.25</td>
<td>[3.8307]</td>
</tr>
<tr>
<td></td>
<td>[10.5237]</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>[9.3506]</td>
</tr>
<tr>
<td></td>
<td>[0.0989]</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>[0.0997]</td>
</tr>
<tr>
<td>Final</td>
<td>[3.8307]</td>
<td>0.2433</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>[9.3506]</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>[0.0997]</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Improvement of the system response was found by slightly decreasing the proportional term and cutting the integral term in half. Hardly any change was necessary for the damping term. The change in system response versus tuning feedback iterations is shown in Figure 139.
The relevant calculated optimization variables for the 5% to 10% step input and the improvement in system response are shown in Table 6 and Figure 140 respectively. For this input it was found that the best response comes with a slight increase in integral term and slight decrease in proportional term. Once again, little change was needed for the damping term.

Figure 139: 5% to 20% Impulse Response Improvement
Table 6: 5% to 10% Iterative Tuning Result

<table>
<thead>
<tr>
<th>Iteration</th>
<th>Test Gains</th>
<th>$J$</th>
<th>$\frac{\partial J(\rho)}{\partial \rho}$</th>
<th>$\frac{\partial J(\rho)}{\partial \rho}$</th>
<th>$\gamma$</th>
<th>Output Gains</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>5</td>
<td>20</td>
<td>0.0525 [0.0091, -0.0067, -0.1421]</td>
<td>[0.0138, -0.0000, 0.0024]</td>
<td>80.25</td>
<td>4.8416 22.5540 0.1009</td>
</tr>
<tr>
<td>2</td>
<td>4.8416</td>
<td>22.5540</td>
<td>0.0346 [0.0300, 0.0020, 0.1152]</td>
<td>[0.0196, -0.0001, 0.0082]</td>
<td>0.1</td>
<td>4.6879 22.3145 0.1006</td>
</tr>
<tr>
<td>Final</td>
<td>4.6879</td>
<td>22.3145</td>
<td>0.0619 [0.0300, 0.0020, 0.1152]</td>
<td>[0.0196, -0.0001, 0.0082]</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Figure 140: 5% to 10% Impulse Response Improvement

Starting with the optimal gains which were found for the 5% to 10% step the optimization was also run for the step down cases. Similar to the step up cases, the step
down cases see large decreases in integral gain for large steps and small increases in integral for small steps. The relevant calculated optimization variables for the 10% to 5% step input and the improvement in system response are shown in Table 7 and Figure 141 respectively. The relevant calculated optimization variables for the 20% to 5% step input and the improvement in system response are shown in Table 8 and Figure 142 respectively.

Table 7: 10% to 5% Iterative Tuning Result

<table>
<thead>
<tr>
<th>Iteration</th>
<th>Test Gains</th>
<th>$J$</th>
<th>$\partial J(\rho) / \partial \rho$</th>
<th>$H$</th>
<th>$\gamma$</th>
<th>$\gamma$ Output Gains</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>4.6879, 22.3145, 0.1006</td>
<td>0.1255</td>
<td>[0.0325, 0.0008, -1.2689]</td>
<td>[0.0126, 0.0000, 0.1374; 0.0000, 0.0001, -0.0200; -0.1374, -0.0200, 111.7572]</td>
<td>0.25</td>
<td>4.0632, 21.1255, 0.1025</td>
</tr>
<tr>
<td>2</td>
<td>4.0632, 21.1255, 0.1025</td>
<td>0.0900</td>
<td>[-0.0044, -0.0082, -0.0627]</td>
<td>[0.0150, -0.0000, -0.2661; -0.0000, 0.0009, -0.0443; -0.2661, -0.0443, 136.6771]</td>
<td>0.1</td>
<td>4.1031, 22.0952, 0.1029</td>
</tr>
<tr>
<td>Final</td>
<td>4.1031, 22.0952, 0.1029</td>
<td>0.0455</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
Figure 141: 10% to 5% Impulse Response Improvement

Table 8: 20% to 5% Iterative Tuning Result

<table>
<thead>
<tr>
<th>Iteration</th>
<th>Test Gains</th>
<th>$\mathcal{J}$</th>
<th>$\frac{\partial \mathcal{J}(\rho)}{\partial \rho}$</th>
<th>$H$</th>
<th>$\gamma$</th>
<th>Output Gains</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>4.6879 22.3145 0.1006</td>
<td>1.1399</td>
<td>0.2552 0.0323 -9.9045</td>
<td>0.1468 -0.0005 -0.6026 -0.0005 0.0007 -0.1753</td>
<td>0.1753 612.5063</td>
<td>0.25</td>
</tr>
<tr>
<td>2</td>
<td>4.2117 10.2205 0.1007</td>
<td>0.7910</td>
<td>0.3487 0.0087 -8.9474</td>
<td>0.1462 -0.005 -1.5366 -0.0005 0.0057 -0.2164</td>
<td>0.2164 613.7500</td>
<td>0.25</td>
</tr>
<tr>
<td>Final</td>
<td>3.6362 9.8641 0.1028</td>
<td>0.5383</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
Using the iterative tuning feedback method has shown that a simple control structure can be designed to give a first order response for the non-linear electronic throttle plant. However, the particular non-linearities of the system require different controller gains for varying inputs. A solution which has been implemented in the rapid prototyping controller is to schedule the proportional and integral gains versus the throttle positioning error. This has been shown to handle different size step inputs; however, it’s performance depends on whether the throttle passes through the return spring set point, shown in Figure 143. Since the controller was optimized with step inputs that pass through the spring set point of 8%, the control performs well in those test cases. Test cases above the set point exhibit overshoot because of excessive integral gain.

Figure 142: 20% to 5% Impulse Response Improvement
Motion Around Spring Set point

Motion Above Spring Set point

Figure 143: Gain Scheduled Controller Performance
CHAPTER 5

CONCLUSIONS AND FUTURE WORK

5.1 Engine Selection

Based on the EcoCAR vehicle architecture a variety of possible engine choices were explored to choose the best option based on efficiency and tailpipe emissions. The requirements of the engine were analyzed by looking at the steady state load vs. vehicle speed profile and by simulating vehicle requirements for urban, highway, and aggressive drive cycles. The dynamic requirements from the drive cycle analysis were simplified into single point representative operating conditions. This was justified because the primarily series-hybrid vehicle architecture allows engine transients to be minimized. Based on the efficiency, the two best candidates were a 1.8L high compression ratio E85 engine and a 1.3L Diesel engine. The 1.8L E85 engine was the final choice because of its minimal output of tailpipe NOx emissions. The choice of a dedicated E85 engine allowed for the use of higher compression ratios and for higher efficiency operation. This efficiency increase helps narrow the fuel consumption gap between gasoline and E85 fueled engines and could increase the viability of ethanol based fuels.
5.2 Engine Simulation

Various engine simulation models were developed by the Ohio State EcoCAR team during the first two years of the competition. Simple black box models were used for tasks like engine selection and software-in-the-loop vehicle simulation. Advanced 1-D engine simulations were used to understand the dynamics of the engine variable length intake and variable intake valve actuation, which decreased experimentation and calibration effort. Mean value engine models were developed to link the engine controller to a hardware simulator. This enabled control software fault mitigation testing before moving control strategies to the dynamometer and/or vehicle. The integrated circular modeling and experimentation process presented in this section of the thesis has the potential to decrease experimentation and calibration effort in many different automotive development programs.

Future work with the advanced GT-POWER model could include validating a model for the delayed valve closure intake valve profile. The current model does not function well with this profile, likely because of large amounts of reverse flow. Also, the creation of an engine failure modes and effects analysis (FMEA) will enable the creation of additional fault cases validation. The FMEA can also drive the development of fault detection and remediation systems in the engine controller.

5.3 Electronic Throttle Control

A variety of techniques were used to explore the electronic throttle control problem. As was shown in the literature, the non-linearity of the throttle increased the difficulty of
creating an accurate throttle simulator. Best results were found by using a proportional, integral, and damping control architecture which was tuned with an iterative feedback tuning method. This method showed good results for tuning control parameters to achieve desired throttle response with a limited number of experimental iterations. However, the non-linearity of the system required gain scheduling of the control parameters to achieve good response to varying step input profiles.

Future work could take the ideas presented in this section of the thesis to combine a feed-forward model based control with the proportional, integral, damping controller. This may cancel some of the non-linearity and allow a single set of optimal control gains to be selected.


