One-Dimensional Air System Modeling of Advanced Technology Compressed Natural Gas Engine

Thesis

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Abstract

With oil prices always being on the rise CNG is emerging as a cheaper and cleaner alternative. This study was carried out to predict the performance characteristics of a 2012 Honda Civic CNG engine which is the only dedicated OEM passenger vehicle for CNG in USA. With CNG having lower CO₂ emissions, higher octane number and being roughly half the price compared to gasoline, it is truly important to develop this emerging CNG market.

In this thesis, a 1–D engine modeling software, GT-Power, was used to model a 2012 Honda Civic CNG engine. Experimental data of that particular engine running on liquid E-85 was available through the EcoCAR 2 competition team’s engine data. The 1-D model was calibrated and validated for accurate predictions of mass air flow with E-85 as a baseline and then validated for CNG engine data, which was available from chassis dyno runs of the vehicle under study. The PI controller developed was able to predict mass air flow values within ±1.5 g/s for most part-load cases after controlling the manifold pressures within ±0.5 kPa.

Actual turbocharger maps for compressors and turbines were used during the matching procedure in order to select the best fit for the engine from a range of available products. The baseline model developed was modified to easily account for different engine configurations and the quantities of volumetric efficiency, air-per-cylinder,
manifold pressure, and mass air flow were observed for comparison between different configurations and checking the accuracy of the model wherever experimental data was available.

It was seen that the results obtained followed proper physical trends with the boosted direct injection engine showing the maximum air per cylinder, which directly relates to brake torque. The air per cylinder of the DI-boosted engines was found to be maximum and was used as the normalization quantity for comparisons. The naturally aspirated cases also predicted proper results with the volumetric efficiency of the direct injection being more than the regular port fuel injection for CNG fuel and lesser than the volumetric efficiency of the liquid fueled E-85 engine.

The maximum air per cylinder (APC) of the CNG DI-boosted engine found to be 5% more than maximum air per cylinder of the CNG PFI-boosted engine, 108% more than maximum APC of the CNG DI engine and 132% more than the maximum APC of the CNG PFI engine. Similarly the maximum atmospheric referenced volumetric efficiency of the CNG DI-boosted engine found to be 5%, 34% and 49% more than maximum volumetric efficiencies of CNG PFI-boosted, CNG DI and CNG PFI engines respectively.
Dedication

I dedicate this thesis to my parents and my sister, without whom my graduate school journey would have been impossible to complete.
Acknowledgements

I would like to thank Dr. Shawn Midlam-Mohler for his constant guidance and patience when things weren’t working out for me. His excellent advising and attention to details have helped me in topics extending far beyond this thesis.

I would also like to extend my gratitude to Dr. Giorgio Rizzoni, Dr. Fabio Chiara and Dr. James Durand, who gave me this unique opportunity to work on the Smart@CAR Consortium project which has expanded my knowledge of IC Engines tremendously.

Finally I would like to thank all my friends who helped me during my course of graduate studies at The Ohio State University in the Center for Automotive Research (CAR). From teaching me simple functions in Matlab and GT-Power to giving me frequent lifts from home to CAR and back, every single person has contributed towards this effort.
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Commonly Used Symbols and Abbreviations

MAF – Mass Air Flow
MAP – Manifold Pressure
$\eta_v$ - Volumetric Efficiency
IAT – Intake Air Temperature
WOT – Wide open throttle
RPM – Rotations per minute
BSFC – Brake Specific Fuel Consumption
GT–ISE – Gamma Technologies – Integrated Simulation Environment
CNG – Compressed Natural Gas
1 – D – One Dimensional
3 – D – Three Dimensional
$C_d$ - Coefficient of Discharge
PI – Proportional Integral.
TPS – Throttle Position Sensor
OBD – On Board Diagnostics
AFR – Air to Fuel Ratio
APC – Air per Cylinder
CHAPTER 1:

INTRODUCTION

1.1) Motivation

With the demand for vehicle emissions to improve growing every day, automotive companies are always looking for alternate sources of fuel for engines and improvement of those technologies. Compressed Natural Gas (CNG) has emerged as one such alternative in the recent past with Asian, South American, and European countries developing large vehicle fleets around CNG.[1]

CNG is a promising alternative fuel comprising primarily of Methane with traces of Ethane and other gaseous constituents. Natural Gas has a 25% lower energy-specific carbon content. It is comparatively cheap and can be used in the important mid-size passenger car segment [4]. While methane is primarily responsible for the combustion characteristics, the secondary species may also affect the Lower Heating Value (LHV), Stoichiometric Air Fuel Ratio (A/FST or AFR), and density of the fuel along with mixture burning speed, octane quality, etc [1].

The chemical make-up of the fuel yields the benefit of lesser CO2 emissions, on the order of 20-30% as compared to gasoline or diesel which are a mixture of hydrocarbons yielding much lower average hydrogen to carbon ratios.
CNG also adds benefits to the combustion process in an engine. Its high octane number allows the engine to be operated in higher compression ratios which leads to a higher thermal efficiency and thus a higher fuel economy rating. The power of the engine can also be raised by the use of effective boosting techniques.

But due to gaseous nature of the fuel, it displaces more air than comparable vapor particles of gasoline, therefore when the fuel is injected into the manifold it has a detrimental effect on volumetric efficiency. A higher stoichiometric air to fuel ratio of CNG engines also suggest that gasoline engines are able to produce more energy for a given quantity of charge [1]. Since, there is a direct relation between the charge ingested to power produced in an engine, there is a need for overcoming the low volumetric efficiency of CNG engines. Some technologies such as Direct Injection (DI) and boosting have been explored in greater detail in this thesis.

The Ohio State University, Center for Automotive Research (OSU CAR) has been focusing on researches related to advanced propulsion systems, advanced engines and alternate fuels for reduced fuel consumptions and emissions for nearly 2 decades. The SMART@CAR Industrial Consortium is one such venture for eminent members of the automotive industry to come together and invest in the common goal of better transportation system for the future.

The Compressed Natural Gas project is a part of this consortium where a stock CNG vehicle (2012 Honda Civic) was characterized experimentally for the purpose of understanding its behavior under various operating conditions. For the complete understanding of an engine, having a detailed model which can be modified to suitably
imitate its behavior is a must. This thesis provides the details on how such a model was
developed using a 1-D Simulation software, GT-Power. The thesis also outlines how the
air path of the model was validated on basis of engine data available from the EcoCAR 2
team.

1.2) Overview of Experimental Characterization

The experimental data for the CNG engine in this thesis is obtained from a Honda
Civic vehicle stock CNG vehicle. The vehicle was provided to the Center for Automotive
Research (CAR) by the Honda-OSU Partnership.

The experimental setup, including the building of a shaft encoder, calibration of
an in-cylinder pressure transducer, installation of thermocouples, and mounting of other
various sensors have been carried out by a fellow graduate student David Hillstrom. The
engine data is being acquired from 2 sources – The On – Board Diagnostics (OBD) port
and by means of a Labview program collecting real – time information from all the
sources mentioned above.

1.3) Thesis Overview

The development of the 1-D simulation model over the duration of the second
year of the project is discussed in this thesis. The development is divided in 5 sections
described below:

- Chapter 2: Literature Review on Engine Modeling
  - This section contains an overview of literature from technical papers,
    journals and an analysis of the various methods of improvement in
efficiencies and performance characteristics of a CNG engine.
• Chapter 3: Baseline GT-Power Model Development and Improvements
  o This section goes over the goals of the model and the need to upgrade various parts of the model based on some preliminary results at WOT and part-load conditions.
  o This section also goes over the successful implementation of a controller which validates the air-flow data by using the experimental data from EcoCAR engine.

• Chapter 4: Development of CNG Models in GT-Power.
  o The Effect of CNG on the model is explored in this section. Advanced engine technologies like DI and Boosting are also explored in different combinations.

• Chapter 5: Simulation Results and Comparisons.
  o The results of running simulations on different fuels and engine configurations are discussed in this section. The comparisons of various engines based on their performance in a simulated environment is also explored.

• Chapter 6: Conclusions and Future Works.
  o This section discusses accuracy of the air path obtained and effects of various engine configurations on the engine breathing characteristics along with possible reasons for any errors. Improvements and possible future work on the 1-D model are also discussed here.
CHAPTER 2:

LITERATURE REVIEW ON ENGINE MODELING

A wide variety of considerations were made while modeling the Internal Combustion Engine on GT-Power. This section provides an overview of the relevant topics from the literature that was used to guide the process of design and modification of the engine models. Some basic concepts have also been visited in this section to provide the relevant background information to understand the fundamentals of I.C. Engines.

2.1) One-Dimensional Gas Exchange Models

1-D modeling is a mathematical representation of a component and its dynamics in the physical world. The Gas Exchange process includes a lot of phenomena that makes it rather difficult to design a high fidelity model with analytical methods. Firstly, the dynamic behavior of gases in intake and exhaust manifolds have to be studied because spatial and temporal variations of thermodynamic properties along the pipes are of great importance.

1-D models, as the name suggests, take into consideration only changes in a single direction. It is normally in the direction of flow of the working fluid. 1-D models lose their credibility in the zones next to a cylinder and during in-cylinder processes as the phenomena of charge mixing, swirls and combustion are highly complex and need to be analyzed in 3-D instead of a simplified 1-D process for more accurate results. These
models need shorter calculation times than 3-D models but also need empirical constants to calculate the more complex phenomenon. An example of model classification applied to an intake manifold is illustrated in the image below.

![Diagram of model classification applied to intake manifold modeling](image)

Figure 1: Example of Model Classification Applied to Intake Manifold Modeling [10]

In [5], a comparison of 1-D and 3-D modeling has been done for a 350 cc, 2-stroke engine. It uses a crude mesh solving Euler’s equations for the intake and exhaust pipes and a 0-D model for the cylinder. Later, with imposed boundary conditions and a refined mesh, a 3-D approach is used to calculate flows inside the cylinder as well. It is seen for various configurations, the errors between 1-D and 3-D models is very less and
the major quantities such as pressures and mass flow rates, match very well as shown below.

Figure 2: Air-flow Matching of 1-D and 3-D Models of a 2-Stroke Engine[5]

The above literature shows that for the purpose of measurement and calibration of air-flow in an engine model, the use of a 1-D model with lesser computation time is a good approach. The Engine specifications of the 1.8L Honda Civic considered for the 1-D model in GT-Power are given in the table below.
<table>
<thead>
<tr>
<th>Parameter</th>
<th>Description</th>
<th>Value</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>$V_d$</td>
<td>Displacement Volume</td>
<td>1.8</td>
<td>Liters</td>
</tr>
<tr>
<td>$V_{im}$</td>
<td>Intake Manifold Volume</td>
<td>5.145</td>
<td>Liters</td>
</tr>
<tr>
<td></td>
<td>Compression Ratio</td>
<td>12.5:1</td>
<td>Unitless</td>
</tr>
<tr>
<td>$\eta_v$</td>
<td>Volumetric Efficiency</td>
<td>Calculated</td>
<td>Unitless</td>
</tr>
<tr>
<td>$P_{im}$</td>
<td>Manifold Air Pressure</td>
<td>Measured/Simulated</td>
<td>kPa</td>
</tr>
<tr>
<td>$m_a$</td>
<td>Mass Air Flow</td>
<td>Measured/Simulated</td>
<td>g/s</td>
</tr>
<tr>
<td>$N$</td>
<td>Engine Speed</td>
<td>Measured</td>
<td>RPM</td>
</tr>
</tbody>
</table>

Table 1: Honda Civic Engine Specifications [18]

2.2) CNG Engine Performance

Methane is a promising energy source regarding limited availability of conventional fossil fuels and increasing demand for efficient propulsion systems. The low C/H ratio of methane compared to conventional fuels leads to lower CO$_2$ emissions and CNG combustion concept allows energy-efficient operation of downsized engines.[6]

It has already been mentioned in Chapter 1, that the volumetric efficiency of a CNG engine is significantly lower than a gasoline engine, which effects the torque output of the engine as lesser air flows into the cylinder per unit time. Volumetric efficiency is
defined as overall effectiveness of the air intake and exhaust system of a 4-stroke cycle
by analyzing the engine as an air pumping device.[9] The expression of volumetric

efficiency of a 4-stroke engine is given as in equation

\[ \eta_v = \frac{2 \dot{m}_a}{\rho_a V_d N} \]  

(2.1)

Where \( \dot{m}_a \) is the mass air flow, \( \rho_a \) is the density of air, \( N \) is engine speed and
‘\( V_d \)’ is the displacement volume of the engine [9]. An experimental study done on a
single cylinder Bajaj – Kawasaki CNG 100 cc engine (4 – Stroke), performed by [20],
illustrates the trend of lower volumetric efficiency for a CNG compared to a gasoline fuel
on the same engine.

Figure 3: Volumetric Efficiency Comparison Sketch [20]
Thanks to CNG's high knock resistance, the compression ratio of the engine can be increased for better combustion efficiency compared to the standard gasoline configuration. As the estimated Research Octane Number (RON) of methane is about 130, compared to 95-98 for premium European unleaded gasoline fuels, the engine can operate at Maximum Best Torque (MBT) spark advances in continuous stoichiometric conditions whatever the load and engine speed [7].

However, under 10x atmospheric pressure, which is brought about by a typical engine compression stroke, the peak laminar flame speed of gasoline is 65% faster than that of CNG. The laminar flame speed of CNG is highest at stoichiometry, therefore there is little incentive to enrichen the mixture, whereas the peak for gasoline resides at an excess air ratio of about 0.9.

Figure 4: Laminar Flame Speed Comparison [8]
Lower flame speed coupled with lesser volumetric efficiency of the air intake system acts as a major disadvantage for the CNG variant when being compared to gasoline counterparts. Many methods have been explored for overcoming the problem of lower torque outputs of CNG engines. Direct Injection, Geometrical modifications, Dual/Bi-fuel technology, Hydrogen integrations and turbocharging are some of the methods which have been explored. A summary of these efforts is shown in [3].

This thesis goes into details of only Direct Injection and Turbocharging of a CNG engine via the means of a GT-Power model. In Direct Injection (DI) engines, combustion gets affected by change in in-cylinder air motion and Fuel injection system characteristics [12] whereas in Port Fuel Injection (PFI) engines, effects of charge cooling, wall wetting and turbulence of the air in the manifold have a significant impact on the engine performance.

With CNG DI engines, fuel can be injected through a high pressure line to produce similar or higher brake power than a gasoline engine [11]. The different types of combustion systems are illustrated in Figure 4. It can be explained that each combustion system has unique features to reflect specific strategies of mixture preparation, combustion control and emissions reduction. However, all systems have a common goal of achieving substantial fuel economy improvement while simultaneously achieving large reductions in engine output and tailpipe emissions [11].
The experimental comparison of gasoline PI, CNG – BI and CNG – DI engines for same displacement volume (1597 cc) showed that the CNG – DI engine produced 4% higher brake power at 6000 rpm as compared to original gasoline fueled engine. The CNG – BI engine produces maximum power of 57 kW at 5500 rpm which is 23% lower than CNG-DI engine’s peak power (at 6000 rpm) [11]. A point to be noted about these results is that the CNG – BI and Gasoline engines operated on a compression ratio of 10:1 whereas the dedicated CNG – DI worked on compression ratio of 14:1. Figure 6 shows the torque curves of the above discussed experimental analysis at wide open throttle (WOT) operating conditions.

Figure 5: Types of Combustion Systems for CNG Fuel [11]
Another study carried out to compare the performance of PFI and DI engines for CNG in [6] shows the advantage of DI in terms of brake mean effective pressure (BMEP). The graph illustrating this is shown below.

Figure 6: Torque Performance Comparison of Various Engine Types at WOT [11]

Figure 7: CNG-DI and CNG-PFI Performance Comparisons [6]
Thus, the advantage and practicality of CNG–DI engines can be seen by this analysis. The technique of modeling of a CNG–DI engine from the baseline EcoCAR engine model in GT-Power is discussed in detail in Chapter 4.

2.3) **Turbocharger Effects on the Engine.**

The objective of exhaust gas turbo charging applied to an internal combustion engine is to increase the density of air in the intake manifold of the engine in order to allow for injecting a larger mass of fuel per stroke thus increasing the production of work on the engine crankshaft.

The increase of the density of the working medium is achieved by means of pre-compressing the air before it reaches the combustion chamber by using a radial compressor coupled, on the same shaft, to a radial turbine. The reason for the choice of this specific class of machines for internal combustion engine applications is related to their to have a high mass flow rate of working fluids while allowing for a quite high pressure ratio increase in a single stage [13]. A rough schematic of a turbocharger arrangement for an IC Engine is shown in *Figure 8*. 
The matching of a turbocharger to an IC Engine requires a certain amount of experimental data. In order to obtain a proper compressor, one should have a particular set of data points for which the turbocharger needs to be designed for. This is because a turbocharger works optimally only in a particular design region and works in a sub-optimal manner for other operating conditions. One should have the values of quantities such as maximum flow rate, maximum power and desired boost level in order to decide upon a compressor size. The boost pressure depends on the available enthalpy in the exhaust flow, which tends to increase with increase in engine speed. In addition to this, the compressor must have sufficient clearance to the surge limit in the lower speed range of the engine along with a clearance at high engine speed to the maximum compressor speed.
When matching the compressor to the engine, Watson [13] stresses the importance of some key operating boundaries in the characteristic compressor map represented in Figure 9.

![Figure 9: Compressor Air Flow Range [13]](image)

The engine speeds on which the peak torques are expected to be seen are chosen as an operating point for design. For example, in the compressor map shown below [15], engine speeds of 5000 and 7000 RPM are chosen as design points for calculation of the other quantities mentioned above.
However, while fixing upon a compressor size during the matching process, one must take care that maximum power performance is not penalized from excessive turbine operation. To avoid extremely high boost pressures and high turbocharger speeds, a bypass (or wastegate) valve is generally operated to discharge the excess turbine inlet energy.
Once the compressor is sized, turbine choice is also important. However, as turbines can operate with high efficiency over a wide range, it is more important to plot the engine operating points on the compressor map.

For the purpose of this thesis, compressor mapping is done on certain WOT cases for purely study purposes as no experimental data is available for the turbocharged engine. A fixed geometry turbocharger has been fitted in the GT-Power model and the compressor and turbine maps have been chosen from available catalogs of Honeywell [15] - [16].

More details specific to the procedure followed for turbocharger matching with respect to GT–Power as the simulating software is discussed in Chapter 4.

2.4) **Motivation for Modeling of Engine.**

As mentioned in Section 1.2), the experimental characterization gives extremely useful data to have as it gives us an idea of how CNG engines would work and understand the characteristics under a wide range of operating conditions by running of the vehicle on a chassis dynamometer. These tasks though will only be useful for the study of this particular vehicle and engine only.

This is the type of situation where software comes into the picture. GT-POWER is an industry-standard engine One Dimensional simulation tool used by leading engine and vehicle makers to model the performance of internal combustion engines. GT–POWER is based on one-dimensional gas dynamics, representing flow and heat transfer in piping and related components. At the heart of GT–POWER are two powerful software domains called GT-ISE (Integrated Simulation Environment) that builds, executes, and manages
the simulation process and, GT-POST, a post-processing tool that provides access to all the plot data generated by the simulation [17]. All simulations and results presented in this thesis are on version 7.3 of GT–Power.

As discussed in Section 2.1, 1–D Simulations take less computational time to give fairly accurate results for simpler phenomena. For the purpose of this thesis, only the air intake and turbocharging are being studied and complex 3–D phenomena of combustion isn’t being analyzed.

Having a good, reliable model which simulates the air path phenomena is a big advantage when one wishes to study different engine configurations. Running a reliable model for various engine technologies gives us the flexibility to see the effect of various parameters on the final output. Also, studying of a model takes much less effort and time than experimentally setting up of various engines on a dynamometer and obtaining data. By making modifications to obtain various configurations and operating conditions we can easily see the conditions under which an engine reaches its physical limits and observe (or predict) behaviors under operating conditions which might be risky to carry out in an engine test cell.

The decision to limit our view to airflow was taken on basis of factors due to lack of empirical data available to properly generate engine maps and thus, making it very difficult to relate the modeling values of torques obtained to the experimental values available. However, a correlation between airflow and brake torque can be seen by analyzing a quantity known as Air Per Cylinder (APC). This quantity was chosen as the normalized quantity that was chosen for comparison. APC incorporates the MAF in a
more intuitive manner by considering volumetric efficiency as well which should show us the trends we expect the model to simulate. The expression for APC is given below.

\[
APC = \eta_v(N)p_a(T_i, \rho_i)V_d = \frac{\eta_v V_d P_{im}}{R_a T_{im}}
\]  

(2.2)

Where \(\eta_v\) is volumetric efficiency, \(V_d\) is the displacement volume, \(P_{im}\) is the intake manifold pressure, \(R_a\) is the gas constant and \(T_{im}\) is the intake manifold temperature [9]. Typically the units of APC are mg/cylinder/cycle. APC directly affects the brake torque of an engine as can be seen from the following plot obtained from experimentally available engine dyno data from the EcoCAR 2 vehicle.
Figure 11: Correlation of APC and Brake Torque.

The above figure makes it clear that a study of APC from different engine models could be a reasonable estimate of engine performance without having to model the complexities of the combustion process and base our assumptions on reliable airflow data from the proposed 1–D model.

In this thesis, a continuous time flow model has been developed by using the 1–D simulation software of GT–Power and the model results have been validated from experimental data available from the EcoCAR 2 competition vehicle engine and also the stock Honda Civic CNG vehicle. All geometric measurements of the air paths have been considered while modeling and certain simplifications have also been made in the model
in order to better troubleshooting and understand the results. All modifications and simplifications have been explained in Chapter 3.
CHAPTER 3:

BASELINE GT–POWER MODEL AND IMPROVEMENTS

The baseline GT–Power model was originally made by an OSU visiting scholar, Davidé Vezza for the EcoCAR competition. The EcoCAR and EcoCAR 2 competition vehicles used a 1.8L Honda Civic CNG engine (2008 model) which was converted to run on E-85 by their engine team. That is the reason why we could easily inherit their GT–Power model as a reference and modify it to model our 2012 Honda Civic CNG engine. Experimental data from both the E-85 and CNG engines were used as validation for the improved and final models built. The engine specifications and geometry which were already present in the model have been specified below.
<table>
<thead>
<tr>
<th>Parameter</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine Type</td>
<td>4 cylinder, In-line SI</td>
</tr>
<tr>
<td>Fuel Type</td>
<td>E85 / CNG</td>
</tr>
<tr>
<td>Bore x Stroke(mm)</td>
<td>81 x 87.3</td>
</tr>
<tr>
<td>Connecting Rod(mm)</td>
<td>137</td>
</tr>
<tr>
<td>No. of Valves</td>
<td>16</td>
</tr>
<tr>
<td>Firing Order</td>
<td>1-3-4-2</td>
</tr>
</tbody>
</table>

Table 2: Cylinder Geometry Used in GT–Power

This chapter goes over the flaws and errors in the original model’s results and the procedure applied to improve it so as to have a proper baseline model.

3.1) Goals of the Proposed Model

The main goal of the model being developed is to be a platform for studying the stock vehicle engine by producing results consistent with physical trends which we might expect from an engine under a wide range of operating conditions. This model is intended to be the platform for analysis of various CNG engine technologies. This thesis intends to calibrate and validate the engine model for airflow in the PFI configuration for both CNG and E-85 fuel. The engine model also is to be used as a platform for performing possible simple extensions so that we observe trends for several possible engine configurations. These modifications include DI–CNG, Dual/Bi–Fuel modeling, Turbocharging and Spark Timing Variations.
As mentioned before, DI and Turbocharging have been explored as modified extensions of the CNG engine model after successfully creating the baseline model with a throttle controller for E-85 fuel. The initial state in which the model was inherited is given by the following diagram.

After addressing issues regarding the airflow in the model via means of developing a throttle controller, the following model was achieved for the E–85 fuel and was also used with CNG as fuel.
This chapter goes over the procedure of building of the throttle controller showed in Figure 13 and the improvements on the airflow performance of the original model.

3.2) Preliminary Simulations

The inherited model had a few different configurations with some differences in each model. For example, there were differences in configurations of the intake manifold. The different forms of intake manifolds are shown below.
Davidé Vezza had also made some models with different valve lifts and varying lengths of the intake runner via a switch to operate the tuning valve. The tuning valve is present in the stock engine and gets activated at high speeds where a shorter air path is desired to reduce pumping losses and maintain a high volumetric efficiency. A brief sketch on how the air path changes by means of the tuning valve is shown in Figure 15.
The tuning valve is closed for all simulation purposes since the initial simulations were validated against the EcoCAR 2 engine data which did not use this valve. Initially, a simulation was run on the model with the complicated intake manifold to check whether the model behaves as expected at Wide Open Throttle (WOT). Simulations were run on both E–85 and Methane–vapor as a fuel and compared to volumetric efficiency available from experimental data. This trend is shown in the figure below and was presented in the annual SMART@CAR Industrial Consortium held in Detroit in August 2013 [14].

![Graph showing volumetric efficiency trend from simulation](image)

Figure 16: Volumetric Efficiency Trend from Simulation [14]

As we can see, the simulations in GT–Power followed the predicted trend of CNG engines having lower volumetric efficiency than the same engine with a liquid fuel of E-
85. This result showed the model had reasonable fidelity and could be used as a baseline by making necessary modifications and improvements on it. This saved us the exercise of having to build an entire GT–Power engine model from scratch and taking measurements of all engine components. The complicated intake manifold was generated by GEM–3D, a GT–Suite tool, which can import CAD files and approximate their shapes into an equivalent 1–D geometry to be used in GT–ISE. For the sake of possible needs of future, the simpler intake manifold geometry was adopted henceforth. This simplified manifold made it easier to track the air path and is similar in volume to the actual intake manifold as specified in Table 1. This can be seen by switching to the ‘Flow Component Scale View’ in GT–ISE and observing the volume amount written at the bottom (appears as 4.8L in Figure 17).
With the simpler manifold, further simplifications such as the modification of the intake runners were also done. The lengths of the air path were measured from the setup of the engine dynamometer so as to be able to simulate the air path under conditions similar to the experimental data. A snapshot of how the model looked in GT–Power is shown and labeled below.
An initial analysis for the model accuracy was done at wide open throttle (WOT) as the air path does not have to go through changing discharge coefficient of the throttle for various part–load conditions. There were 10 WOT cases or data points available for study and the first set of simulations which were run gave us the following results for mass airflow.
<table>
<thead>
<tr>
<th>RPM</th>
<th>MAF(Expt.), g/s</th>
<th>MAF(Simulation), g/s</th>
<th>Error %</th>
</tr>
</thead>
<tbody>
<tr>
<td>1078.3</td>
<td>12.7</td>
<td>13.6</td>
<td>-7.09</td>
</tr>
<tr>
<td>1308.1</td>
<td>15.8</td>
<td>16.52</td>
<td>-4.42</td>
</tr>
<tr>
<td>2157.2</td>
<td>27.8</td>
<td>28.09</td>
<td>-1.17</td>
</tr>
<tr>
<td>2416.9</td>
<td>31.2</td>
<td>31.26</td>
<td>-0.26</td>
</tr>
<tr>
<td>2576.8</td>
<td>32.8</td>
<td>33.26</td>
<td>-1.4</td>
</tr>
<tr>
<td>2776.4</td>
<td>34.9</td>
<td>35.9</td>
<td>-2.83</td>
</tr>
<tr>
<td>2876.4</td>
<td>37.0</td>
<td>37.41</td>
<td>-1.1</td>
</tr>
<tr>
<td>3225.9</td>
<td>40.9</td>
<td>43.86</td>
<td>-7.1</td>
</tr>
<tr>
<td>3655.4</td>
<td>49.5</td>
<td>49.82</td>
<td>-0.57</td>
</tr>
<tr>
<td>4075.0</td>
<td>57.4</td>
<td>57.79</td>
<td>-0.7</td>
</tr>
</tbody>
</table>

Table 3: MAF Error Analysis at WOT

The above table shows that the model works reasonably well with low %Error values for mid-range RPMs, but has higher errors at low engine speeds and at a single data point around 3225.9 RPM. To explore this matter further, a fine RPM sweep was conducted on the WOT condition where the model was made to run in 25 RPM increments from 1000 to 4175 RPM. We saw that the model deviates from the experimental data near 3225.9 RPM but due to lack of experimental data points in the vicinity of the point, we can speculate that the model is experiencing some sort of
resonance near that point. The plot showing the comparison of experimental and simulation MAF values is shown below along with the fine RPM sweep.

![Plot showing comparison of experimental and simulation MAF values](image)

**Figure 19: Air Flow Analysis Curve at WOT**

In the same set of simulations the brake torques were also measured and compared to the experimental data we had from the engine dyno. Shown below in *Table 4* and *Figure 20* are the table and graph for brake torque comparison of the engine.
<table>
<thead>
<tr>
<th>RPM</th>
<th>Brake Torque (Expt.), N-m</th>
<th>Brake Torque (Simulated), Nm</th>
<th>Error %</th>
</tr>
</thead>
<tbody>
<tr>
<td>1078.3</td>
<td>137.46</td>
<td>117.15</td>
<td>14.78</td>
</tr>
<tr>
<td>1308.1</td>
<td>141.22</td>
<td>120.5</td>
<td>14.67</td>
</tr>
<tr>
<td>2157.2</td>
<td>147.92</td>
<td>130.96</td>
<td>11.46</td>
</tr>
<tr>
<td>2416.9</td>
<td>149.55</td>
<td>130.58</td>
<td>12.68</td>
</tr>
<tr>
<td>2576.8</td>
<td>147.77</td>
<td>130.52</td>
<td>11.67</td>
</tr>
<tr>
<td>2776.4</td>
<td>150.07</td>
<td>131.05</td>
<td>12.68</td>
</tr>
<tr>
<td>2876.4</td>
<td>151.52</td>
<td>131.62</td>
<td>13.13</td>
</tr>
<tr>
<td>3225.9</td>
<td>149.46</td>
<td>137.73</td>
<td>7.84</td>
</tr>
<tr>
<td>3655.4</td>
<td>153.79</td>
<td>136.81</td>
<td>11.03</td>
</tr>
<tr>
<td>4075.0</td>
<td>155.43</td>
<td>140.51</td>
<td>9.60</td>
</tr>
</tbody>
</table>

Table 4: Brake Torque Analysis at WOT
Due to the low error in Figure 19, the simulated brake torque curve in Figure 20 should be following the trend of the experimental data. This suggests an incorrect combustion and/or friction model is currently present in the engine cylinder. However, since we are not concerned with the combustion model for this thesis, we will be using brake torque or BMEP values for reference and will be focusing almost exclusively on variables related to air induction.

3.3) Modification of the Throttle Body for Part Load Operations

After having verifying the WOT performance of the engine, the part–load conditions were looked in greater detail. Due to very significant errors in results, many improvements were made along the way towards the development of the model. Firstly,
we ran all the 120 part–load condition cases through our model. All these cases had various values of engine speed and throttle angle. A comparison of the experimental and simulated MAF values are given in the figure below.

We can clearly see that there exists some type of saturation for a range of low throttle points at approximately 5 g/s. This clearly pointed towards an issue with the throttle body. It was found that the original model was using discharge coefficient values dissimilar to the experimentally obtained values. The figure below shows the original discharge coefficients used in the inherited model in both forward and reverse directions.
Figure 22: Discharge Coefficient in Original Model

Having the above profile for discharge coefficient generated many significant errors in MAF, which are illustrated in the figures below.

Figure 23: Airflow Error Trend w.r.t Throttle
Since both engine speed and throttle angle are inputs to the GT–Power model, a plot showing the trend of error in MAF with engine speed was also documented and is shown below. It can be observed that the % Error in MAF is large over the entire RPM range.

![Plot showing MAF error analysis with original throttle body](image)

**Figure 24: MAF Error Analysis with Original Throttle body**

The Coefficient of Discharge for the throttle valve is found by just renaming a few quantities of the orifice flow equation and is given by the following expression,
\[ m_{th} = \frac{C_D A_{th} p_0}{\sqrt{R T_0}} \left( \frac{p_{im}}{p_0} \right)^{\gamma/2} \left\{ \frac{2}{\gamma - 1} \left[ 1 - \left( \frac{p_{im}}{p_0} \right)^{(\gamma - 1)/\gamma} \right] \right\}^{1/2} \] (3.1)

Where \( A_{th} \) is the throttle plate open area, \( p_0 \) and \( T_0 \) are the upstream pressure and temperature (ambient in this case), \( p_{im} \) is the downstream pressure (Intake manifold in our case) and \( C_D \) is the discharge coefficient. The above equation is valid when the pressure ratio is below the critical ratio of 0.528 [9]. For pressure ratio higher than that, the throttle is choked and the airflow is given by,

\[ m_{th} = \frac{C_D A_{th} p_0}{\sqrt{R T_0}} \gamma^{1/2} \left\{ \frac{2}{\gamma + 1} \right\}^{(\gamma + 1)/(\gamma + 2)} \] (3.2)

The values for all other quantities were available from the EcoCAR experimental data, the values for \( C_D A_{th} \) and \( C_D \) were calculated. The plot for the experimental discharge coefficients for all part load conditions is given.
Since discharge coefficients need to be monotonically increasing with throttle opening, the points which did not fit this trend were eliminated and a cubic polynomial was fit onto the values of $C_D$. The resulting curve was then used to generate the new $C_D$ curve in GT–Power. The fitted curve to the experimental $C_D$ values is shown in the following figure.
Figure 26: Curve Fitted Experimental Discharge Coefficients.

By use of the above $C_D$ values we can see a drastic improvement in the values of MAF as the saturation we see in Figure 21, are not present anymore as illustrated below.
However even though the airflow response was improved, there still was a major error in the MAF. A major reason for this could be that the Manifold Pressure (MAP) values that were being generated by the model were incorrect. The corresponding % Error plots for MAF and MAP are given in figures below.
Figure 28: Error Analysis of MAF with New Throttle Body

Figure 29: Error Analysis of MAP with New Throttle Body
The figures above thus showed that there was a need to get proper values of MAP in order to achieve a more accurate model of MAF. From observing how much the error had reduced by changing of the throttle discharge coefficients and considering that the experimental data obtained from the EcoCAR team had known issues of repeatability in their Throttle Position Sensor (TPS), the need for development of a throttle controller in the simulation environment emerged.

3.4) Development of a Controller for the Model

Alongside offering the means for simulation of many different engine and vehicle configurations, GT–Power also has the option of implementing controllers in its models in order for calibrating and validating with a certain set of experimental data.

There are various types of controller objects which are available in GT–Power, such as an ‘Optimization Controller’, ‘Injector Controller’, ‘Throttle Controller’, ‘EGR Valve Controller’, ‘Mass Flow Controller’, ‘PID Controller’ and some for Turbochargers. For our purpose, we could use the Throttle controller or a PID controller to control the MAP and in turn get more accurate flow rates.

The throttle controller template calculates the desired throttle angle or orifice diameter based on several physical quantities of the engine including upstream density, pressure and engine speed. The feed-forward portion of the controller is open-loop. This means that it estimates the throttle opening based on modeling fundamentals (i.e. RPM, engine size, throttle size, plenum volume, etc.) without taking into the "error" between the specified “Target Signal” and the input signal. The feedback controller uses an adjustable-gain PI technique to minimize the remaining error [21].
However, it was found that GT–Power template had problems in trying to adapt the proportional and integral gain values for a variety of cases. As the controlled signal neared the target value, the actuator would start oscillating the throttle between 2 limits and lead to the oscillation of the controlled signal as well (shown in figure).

Figure 30: Controller Performance for "Adaptive" Gains

This lead to the decision to implement a simple PID controller with values of gains scheduled by means of a ‘Controls.xls’ file provided by GT-Power which schedules the gains for any operating point for which the controller needs to be tuned for. The Excel file assumes that the plant behaves like a linear first order system and then characterizes
the plant on some time constant, input and output changes. A brief look into the ideology is illustrated.

And the equation used in GT-Power for the first order approximation is given below,

\[ Y = Y_{\text{initial}} + (K)(\Delta X)(1-e^{-\frac{t}{\tau}}) \]  

(3.3)

Where \( K = \Delta Y / \Delta X \), also known as the output ratio

\( t = \) time

\( \tau = \) time constant of the plant
The values generated by the Excel file were then used on the model and all part load conditions were run on it. The values of Proportional and Integral gains were

$$K_p = 13.78$$

$$K_i = 289.1$$

These values turned out to work well for all part load conditions, thus an entire exercise of gain scheduling wasn’t necessary across various operating ranges.

A snapshot of the engine model with a controller is shown below.

Figure 32: Engine Model with Controller
For the case shown in Figure 30, we get the following graph with the PI Controller (Shown in snapshot above).

![Graph showing manifold pressure, target MAP, and throttle angle over time.](image)

Figure 33: Performance Summary of PI Controller

### 3.5) Validation of Controller with E–85 Data

The proportional and integral gains were changed to 20 and 100 for better WOT performance (faster convergence) and the model was run for all cases (part–load and WOT). The controller performance is plotted in the following graphs.
We can see in the figure above that the value of error stays within ±0.5 kPa for most conditions and only increases to -2 kPa at WOT. A similar trend is observed while plotting the %Error for MAP.
We can now have a look at the MAF values and compare it with the desired experimental MAFs. The absolute value of the errors are quite small but as the desired values in low throttle operating points are quite low, the %Error in MAF is slightly high in this region.
Figure 36: Accuracy of Airflow Model (Error Values)

Figure 37: MAF Accuracy After New PI Controller
An airflow comparison between experimental and simulated MAFs gives a better picture of how well the controller is working. From the figure below and comparing it to Figure 21, we can see a vast improvement in the low throttle response of the system.

![Graph](image)

Figure 38: Comparison of Airflow for Model and Experimental Data

Additional points which can be noted are the amount by which the throttle response of the simulations differ from the measured value of throttle angle by the TPS in the EcoCAR team’s data.
Figure 39: Error Analysis on Throttle Values

It can be seen above that the experimental TPS was predicting more throttle opening values than the model in the beginning and reverses the trend as the throttle angle increases with the average lying about -0.8 degrees, meaning the model predicted lesser values on average. The trend for %Error in throttle is given in the following plot.
So it can be inferred that the PI controller developed is able to perform aptly to give a reasonable prediction of MAF values. The above plots presented in this chapter show that by proper control of MAP, we can make good predictions on MAF values since airflow is dependent on manifold pressure as shown in equation (3.1). In conclusion, the controller is able to reduce the error in MAF for the part load simulations as shown in Figure 38. Thus, we can use this PI Controller for the part load conditions of the CNG engine by simply switching the fuel in the model to methane. The process of development of different CNG engine configurations is explained in the next chapter.
CHAPTER 4:

DEVELOPMENT OF CNG MODELS IN GT-POWER

After properly tuning the PI controller properly, the next course of action was to modify the baseline E-85 model to run on CNG fuel and run simulations for part load and WOT cases. The main challenges faced in this stage of the project was the validation of the model with experimental data. Due to several delays, availability of equipment and time constraints, only a few data points were successfully gathered and validated against.

The CNG engine was run on the car in a chassis dynamometer testing setup, instead of an engine dynamometer, which would have given better ideas regarding the load torques against which the engine is operating. More in–depth details of these challenges on the experimental side are explained in another fellow MS student’s thesis working on this CNG engine, David Hillstrom. The main goals of this chapter are to modify the baseline E-85 model into 4 separate CNG engine models as shown in the figure below.
4.1) **Switching to CNG Fuel and Integration with Experimental Data**

The advantage of modeling in a software is truly shown in this section, where we are able to switch conveniently between various engine technologies by making some minor adjustments or modifications to the baseline model. Switching of fuels from E-85 to CNG is just a matter of selecting the fuel type in the following figure and changing other parameters for it which might need to be addressed.

![Figure 41: Goals of CNG Engine Modeling](image)
The modification is simply ignoring the vaporized fluid fraction and changing of the AFR to 16.9 for pure methane [1] from 9.87 for E-85. Since the composition of CNG differs, for sake of uniformity and idealness, pure methane has been chosen as the fuel of choice to replicate CNG behaviors. Compared to the large range of operating conditions of E-85 data that was available from the engine dyno, only a small fraction of it could be extracted from the chassis dyno testing for CNG.

The data was obtained and recorded in real time from the OBD port. During post-processing of the data, regions where MAP, engine speeds, etc. appeared to be in steady state were averaged out and then sorted. Thus, from a few hours of testing only 15 steady-state data points were extracted. Also, unlike an engine dyno, where we can give a constant engine speed or torque as an input, here the pedal position was the only way the vehicle could be run at various speeds as is the case in real world driving. This gave us many restrictions on the range of engine speeds or throttle openings which could be
achieved. However, with the limited data points to corroborate against, the modified model performed well to suggest a good set of gains (3. 2).

The results as shown in the next chapter will follow the trend as explained in section 2.2). The cases are setup for the 15 operating points is done the usual manner in GT-Power. However, if we want to compare across fuels or engine technologies, we need to compare them over the same operating conditions and then over some normalized common quantity. For this particular reason, WOT was chosen as the data points of comparison across fuels and engines and for all the naturally aspirated engines, it showed the expected trends. The WOT operating conditions were the same as ones considered in beginning of Chapter 3.

4.2) Exploration of other engine technologies for CNG

As discussed in Chapter 2, by having a baseline model, we can make modifications to the structure of the model and obtain a simulation environment for different engine technologies. The types of CNG engine models obtained and studied in the thesis are:

4.2.1 Port Fuel Injection

As explained before, the PFI model is made by just switching of the fuel from E-85 to methane and also modifications of other parameters.

4.2.2 Direct Injection

A direct injection model is made in a GT–Power model by changing the injector template and using a simple Diesel injector but by changing the fuel object in it to
‘methane – vap’ as we are assuming CNG to be pure methane for simplicity. The template is then directly attached to the engine cylinder.

The only point of concern was specifying the injection timing and duration of the injectors as the injection process must occur between the events of IVC and start of ignition (SOI). After observing the SOI values in CAD from GT–Post for the PFI model, it was a simple choice to have 40 deg bTDC as the value for injection timing for the CNG. The template of the injector used is shown below.

Figure 43: Direct Injector Template With Values Used in Model.

The model generated was seen to follow the expected trends of MAF, volumetric efficiency, APC etc. when compared with a PFI engine. The detailed plots are shown in Chapter 5.
4.2.3 PFI Engine with Turbocharger

By using a turbocharger, we are increasing the density of the inlet air so that we get better volumetric efficiencies, higher MAFs, MAPs, APCs etc. With regards to GT–Power, a guided procedure is laid out in the help manuals regarding how a turbocharger can be matched to an engine. However, this procedure needs experimental data from the engine at peak torque speeds in order to match properly.

As said before, a turbocharger is matched with an engine by means of creating a compressor and turbine map with proper scaling which makes the model run within the surging and choking limits of the compressor by verifying the operating point locations.

The model is created by first adding a compressor map from a commercial turbocharger in Garrett – Honeywell catalogue and then adding a turbine to the end and using the turbine map for the corresponding compressor. Both the compressor and turbine objects are then connected via a Turbocharger Shaft template in GT–Power and the initial speed was specified as an estimate as it changes to the proper steady state value after the model is allowed to run for some time. Since the turbocharged model only operates at WOT, the throttle controller can be removed.

In addition to the compressor and turbine objects, a ‘cooler’ must also be attached at the outlet of the compressor to reduce the temperature of the compressor air and thus, further increase its density of the inlet air. The structure of the model is shown in the figure below.
The results of the simulations and the effects on the airflow quantities are observed in the following chapter.

4.2.4 DI Engine with Turbocharger

The DI – boosted model is made by following the same process of adding a turbocharger in the exhaust and intake system as explained in the PFI – boosted case but by starting with the DI naturally aspirated model as the base line and adding the compressor and turbine as specified above. The architecture of the model as seen in GT–Power is shown below.
Both the PFI and the DI–Boosted models use the GT1544 turbocharger as specified in [16]. This turbocharger was chosen as the power rating for this product is 100–160 hp, which fits in with our vehicle’s specification of 110 hp. The compressor and turbine maps of the chosen turbocharger are given in the figures below.
Figure 46: Compressor Map for GT1544

Figure 47: Turbine Map for GT1544
As can be seen from Figure 46, the operating points from the simulations are well within the compressor map and away from the surging and choking lines. This trend is seen for most cases of WOT operations. The detailed results of impact of turbocharging in airflow parameters and engine performance is shown in Chapter 5.

4.3) Bypassing of Combustion Model

As we had explained before, GT-Power does a good job in simulation of 1-D airflow paths but fails to provide a reliable combustion model as combustion is a highly 3-D process occurring in multiple phases. Although, GT-Power does have the capability of modeling combustion, it needs a lot of empirical data which can be obtained by complete experimental characterization of the engine and obtained in-cylinder engine performance parameters.

Since, parameters such as in-cylinder pressures and temperatures were not available this lead to the incapability to obtain quantities such as heat release rates, bmep, bsfc etc. Thus, an accurate engine map could not be constructed. However, as we have seen in Figure 11, the brake torque has a direct correlation with APC, creating a reliable air model for an engine provides a fair idea of the engine performance without knowing its combustion characteristics.

Therefore, by the above modeling processes followed, this study’s aim was to create a reliable airflow model, validated for all available experimental data points to give us reasonable engine breathing and temperature characteristics for intake and exhaust sides. This process has thus given us the capacity to model various turbocharged configurations which were shown in this chapter.
CHAPTER 5:

SIMULATION RESULTS AND COMPARISONS

The simulations were run for all part load cases for the naturally aspirated engine configurations and only for the WOT for the turbocharged cases. A comparison of the naturally aspirated cases and turbocharged cases with each other as well as with the baseline engine case of E–85 fuel is illustrated in this chapter and we can see the effect of changing fuels as well as application of different engine technologies after switching over to gaseous fuel. The process followed this far for the validation process is shown below.

![Validation Process Flow Diagram](image)

Figure 48: Validation Process Flow Diagram
The comparison of the PFI model with available experimental data is also illustrated in this chapter alongside some discussions of the observed trends in the results.

5.1) Validation of Model with Experimental CNG Data.

As mentioned before (section 4.1), the data from the OBD port was obtained by running the vehicle on the dyno and averaging out portions of data with constant MAPs and using that averaged value as a data point for validation. A look into the controller performance for the CNG fuel PFI model can be shown below for the data points obtained from the dyno.

![Controller Performance for Dyno Data Points - CNG Fuel](image)

Figure 49: Controller Performance for Dyno Data Points - CNG Fuel
As we can see that the throttle controller is able to converge the MAP to the desired tolerance of ± 0.5 kPa for all the cases except for the case of 800 RPM, where the error is 0.96 kPa. The average error in MAP is observed to be around 0.23 kPa. The MAF error can also be seen to be closer the diagonal line when comparisons are drawn between simulated and experimental data.

![Figure 50: MAF Comparison Plot for Chassis Dyno Test](image)

The error could be attributed to the fact that the experimental data has been averaged for points showing constant MAP and RPM values. A much more accurate representation of the comparison would have been possible if engine dyno data was available for the CNG engine instead of relying on ECU data from a chassis dyno run.
5.2) **Comparison of WOT Results of 4 CNG Engine Configurations**

The volumetric efficiency comparison shows an expected trend of the airflow with the CNG engines having lesser volumetric efficiency than the liquid fuels for naturally aspirated cases. The volumetric efficiency for the boosted engine cases gives higher values as expected. It was seen that the CNG DI-boosted case has a maximum volumetric efficiency of 5%, 34% and 49% more than the maximum volumetric efficiencies of CNG PFI-boosted, CNG DI and CNG PFI engines respectively. The values have been tabulated in Appendix A.

![Figure 51: Atmospheric Referenced Volumetric Efficiency Comparison at WOT](image)

The APC curve comparisons at WOT also showed expected patterns. All APC values have been normalized with the maximum APC of all CNG engine configurations,
which is of the CNG – DI configuration. From the simulation results tabulated in Appendix A, we can see that the maximum APC of the CNG DI-boosted engine is 5% more than maximum APC of the CNG PFI-boosted engine, 108% more than maximum APC of the CNG DI engine and 132% more than the maximum APC of the CNG PFI engine. The trend of APC variation for different cases is shown in the figure below.

![APC Comparisons for CNG Engines](image)

Figure 52: APC Comparisons for CNG Engines

Again it can be seen from the figure above that the naturally aspirated cases behave properly with the APC being higher for the DI case as all fuel is injected directly into the cylinder after IVC, giving chance for more air intake. Looking at the boosted cases one sees that the DI – boosted case has higher APC than the PFI – boosted engine.
for most cases but reduces in high engine speeds. This can be due to the turbine not working under maximum efficiency as the enthalpy difference between inlets and outlets might not be huge. As the temperature values can’t be trusted post combustion, it might also be possible to get better efficiency via a wastegate.
CHAPTER 6:

CONCLUSIONS AND FUTURE WORKS

From the results and discussions presented in Chapters 3 through 5, many conclusions can be drawn regarding the procedure followed for the model development, validation of the experimental data and exploration of different CNG engine configurations. This chapter summarizes the results of the entire model building process and suggests the scope for future work which can be done on the model(s).

6.1) Conclusions

The basic structure of the baseline model was inherited and explored for WOT cases to check its accuracy. After being satisfied with the WOT performance in E-85 fuel, the modifications were then carried out for part-load conditions as well.

The following conclusions could be drawn from the results of the development of the proposed 1-D Model.

- A PI-Controller was successfully developed to provide accurate MAF predictions for the E-85 fuelled EcoCAR engine. It was seen that the controller is able to maintain the error in MAP within ±0.5 kPa and consequently the MAF values generated by the model closely matched the experimental data available from the EcoCAR engine dyno tests.
• This controller provided the engine model the flexibility to be operated on different fuels and different engine configurations and under a wide range of operating conditions.

• Model was compared and validated against experimental data available from the Honda Civic vehicle’s chassis dyno test after switching the fuel to CNG. Although very few data points were obtained, it could be seen that the MAP was well controlled by the achieving purely steady state values of engine operation from a chassis dyno test run. The real time data being averaged might not give an accurate data point for the model validation process.

• Modifications were made to the CNG model in order to explore performance parameters variations. Since the models’ comparisons were carried out in WOT only, the PI – Controller was not included in this model.

• From the figures shown in the previous chapter, we can observe that the naturally aspirated engines follow the expected trend of DI having higher volumetric efficiency and APC than PFI engine. However, the volumetric efficiency of the liquid fuel is still calculated as more than these 2 cases. The reason for E–85 having more volumetric efficiency than DI might be due to charge cooling effect in the intake manifold.

• The turbocharger matching was done by trying out different compressor and turbine maps at the intake and exhaust side and running the model at WOT conditions. The final turbocharger product was chosen on basis of best
performance in terms of operating point being away from choking or surging lines and yet giving a better mass air flow value when compared to its naturally aspirated counterpart.

- The turbocharger maps selected were simulated under fixed geometry conditions without a wastegate. However, in reality the turbocharger product selected has a wastegate as specified in [16]. This might be leading to some issues in solving for MAP trends in DI–boosted engines and thus causing the volumetric efficiency and APC of the DI–boosted engines to drop down a bit at higher RPMs.

- It was also seen that the boosted engine were giving much higher volumetric efficiencies and APCs than naturally aspirated engines of both CNG and E–85 fuel. And as we have seen in Figure 11, this would translate into higher torque.

- If we look closely at Figure 52, we can see that the APC of the turbocharged cases are roughly twice in value than its naturally aspirated configurations giving us a glimpse of the possible downsizing that can be done on an engine. This would essentially mean that a 1.8 L turbocharged DI or PFI engine could produce a brake torque similar to that of a 3.0 L or 3.2L naturally aspirated engine as APC and brake torques have a direct relationship as illustrated in Figure 11.

- A base model framework was thus created calibrated and validated against experimental data for the Honda Civic CNG engine and can serve as a platform for study of exploration of engine behavior at various configurations and operating conditions.
6.2) Future Works on the Model.

The future work which can be done on the modeling aspects are mainly related to the combustion modeling side and turbocharger fine-tuning. As explained before in section 4.3, measurement of in-cylinder pressure and temperature would enable us to estimate various quantities such as Wiebe parameters, heat release rates, engine maps, bmep, bsfc etc. Only after these values are available, can we model the combustion in GT–Power.

The process of obtaining in-cylinder data is already underway in the experimental characterization of the vehicle.

After getting a more reliable, combustion model, the BSFC and power of the engine will be more accurate which can be utilized in the process of turbocharger matching. In addition to this, modeling of a wastegate into the turbine would be a more realistic approach of the enthalpy drop occurring in the exhaust flow.

This model could act as a basic platform to observe engine behavior at various operating conditions. This study and technique of controlling MAP to get MAF could be employed in a more advanced Virtual Engine Development softwares such as QuickSim or StarCD developed by FKFS in Germany.
REFERENCES


[10] Rizzoni, Giorgio, and Krishnaswamy Srinivasan. “Module 1, ME 7236 Powertrain Dynamics.” The Ohio State University, Department of Mechanical Engineering, Fall 2012.


APPENDIX A: Experimental and Simulated data

1) The Data points from the Chassis dyno for the Honda Civic Vehicle are given in the following table.

<table>
<thead>
<tr>
<th>Engine Speed (RPM)</th>
<th>MAF (g/s)</th>
<th>MAP (kPa)</th>
<th>Throttle Position (deg)</th>
<th>Simulated MAP (kPa)</th>
<th>Error in MAP (kPa)</th>
<th>MAF Simulated (g/s)</th>
<th>Error MAF (g/s)</th>
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2) The data points for the CNG PFI Simulations run at WOT for various cases are given in the table below.

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