Modeling and Analysis of Crankshaft Energy Harvesting for Vehicle Fuel Economy Improvement

THESIS

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By

Benjamin M. Grimm

Graduate Program in Mechanical Engineering

The Ohio State University

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Master's Examination Committee:

Professor Marcello Canova, Advisor

Professor Giorgio Rizzoni

Dr. Shawn Midlam-Mohler

Dr. Fabio Chiara
Abstract

Vehicle fuel economy improvement is currently a widespread theme in the automotive industry. To this effect, the U.S. Department of Energy has initiated a large-scale research project aimed specifically at the topic. In collaboration with Chrysler LLC, the Center for Automotive Research (CAR) at The Ohio State University has been commissioned to develop a Vehicle Energy Manager (VEM) designed to optimize energy flows within a vehicle. A comprehensive, forward-looking Vehicle Energy Simulator (VES) was developed as part of this project to serve as a control development platform for the VEM. The work presented herein relates to crankshaft energy harvesting as it pertains to conventional vehicles that are not equipped with the required hardware for typical regenerative braking. Rather, detailed management of engine-level ancillary loads coupled with driveline control strategies optimized for energy harvesting have been explored. Moreover, investigation of the potentials for kinetic energy recovery by displacing the use of friction brakes has been performed. It was found that, by reducing pumping losses during deceleration fuel shutoff (DFSO) and displacing 20% of the braking requirement during a FTP, a 1.8 MPG improvement could be realized for the vehicle studied. The energy recovery component corresponding to PMEP reduction through manipulation of throttle has been optimized by identifying instantaneous engine and powertrain states that maximize harvesting opportunities. Additional gains would accrue as the brake displacement percentage increases to near-hybrid levels.
Acknowledgements

I would like to thank all of the members of this project for their support during my time as a graduate student at The Ohio State University. Specifically, I would like to thank my advisor, Professor Marcello Canova, for giving me the opportunity to be a part of such a detailed and demanding endeavor. Dr. Shawn Midlam-Mohler helped acclimate me to the environment that is "CAR," and Dr. Fabio Chiara offered invaluable assistance during the long and tedious process of writing a thesis. I also owe a debt of gratitude to my counterpart in this project, Neeraj Agarwal, who on a daily basis unwaveringly answered a never-ending stream of questions on topics ranging from programming commands, to the bus schedule. All in all, I had a very rewarding experience as a graduate student. CAR is a special place where extreme technical expertise exists within a sense of community. I won't forget.
To my parents. Always supportive.
Vita

2006 .......................................................... Dalton High School

2011 .......................................................... B.S. Mechanical Engineering,

The Ohio State University

2010 to present ......................................... Graduate Research Associate,

Department of Mechanical and Aerospace Engineering, Center for Automotive Research, The Ohio State University

Fields of Study

Major Field: Mechanical Engineering
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Chapter 1: Introduction and Scope of Work

Automakers worldwide are under extreme regulatory and consumer scrutiny to increase the fuel economy of their vehicles. Fuel economy (FE) improvements are not only beneficial for the environment, but reduction in fuel consumption by the transportation sector has significant economic advantages as well. To this effect, the automotive industry is investing heavily in fuel-saving technologies. Traditionally accepted methods of FE improvement such as engine downsizing + turbocharging are being pursued with vigor, but more subtle, often incremental improvements are sometimes overlooked.

Section 1.1 Scope of Work

As part of a large research project funded by Chrysler and initiated by the U.S. Department of Energy, The Ohio State University Center for Automotive Research (CAR) has been commissioned to develop a supervisory Vehicle Energy Management (VEM) strategy to improve vehicle fuel economy. The VEM aims to reduce fuel consumption by 4-8% through the optimal usage of engine-level auxiliary loads and advanced thermal system architectures. Before work can begin on the VEM, a Vehicle Energy Simulator (VES) had to be developed around which controls could eventually be designed. At this stage, the VES has been validated and preliminary analyses have begun that will ultimately be useful for the VEM development. The project can broadly be
divided into two sections: Ancillary Load Reduction (ALR) and Thermal Management Systems (TMS). The goal of the TMS component of the project is to reduce warm-up time of the vehicle's working fluids. In doing so, viscosity, which proportional to temperature and inversely proportional to powertrain efficiency, can be reduced to steady-state levels as soon as possible. Details of the TMS modeling and analysis can be found in [24]. This document focuses on the ALR component of the project, specifically on the crankshaft energy harvesting elements. The overarching goal of the work presented herein is to explain the modeling and validation procedure required to develop the VES, and to lay a foundation for determining the crankshaft energy available for recovery. The energy recovery analyses will be presented in a logical progression from unconstrained cycle analyses to detailed harvesting scenarios that take into account realistic vehicle constraints. The tools developed for the Vehicle Energy Analysis chapter could be extended to other vehicle platforms where different, or increased opportunities for crankshaft energy harvesting exist.

**Section 1.2 Document Layout**

Following the introduction, this document contains four distinct chapters that sequentially build upon one another.

**Chapter 2: State of the Art** includes a general summary of the technologies and strategies developed specifically in response to the "energy consumption problem." The chapter is divided amongst propulsion-related, non-propulsion-related and energy saving technologies.
Chapter 3: Model Development, Calibration and Validation focuses on the structure and development of the VES with particular emphasis on the powertrain and Chrysler control strategy models. VES-level, as well as component-level calibration and validation results are included where applicable.

Chapter 4: Vehicle Energy Analysis contains the bulk of the original work presented in this document. A systematic approach to identify and maximize recoverable crankshaft energy has been devised for implementation in both post-processing and simulation environments. Normalized energy results are included for comparison between scenarios.

Chapter 5: Conclusions and Future Work summarizes the most important findings from Chapter 4 and list some future work that should be undertaken to extend this study. Since the VES is an ever-evolving model, a detailed list of suggested model-based improvements has been included.
Chapter 2: State of the Art

Section 2.1 Overview on Energy Use in Transportation

As energy concerns mount worldwide, the efficiency of automobiles is becoming of paramount importance. The reasons for concern are numerous and include environmental issues associated with the burning of fossil fuels, the increasing consumption of finite resources and in the case of the United States, the economic burden incurred from an enormous oil production deficit. Figure 1 depicts this deficit in a dramatic fashion.

![Figure 1: Geographic and resource constraints for oil [1]](image-url)
Transportation currently consumes a significant percentage of the fuel refined worldwide which is a trend that will only worsen. From 2008 to 2035, the transportation sector is estimated to account for 82% of the increase in liquid fuel use. The massive influence of transportation on energy consumption can be seen in Figure 2.

![Figure 2: World liquid fuel consumption by sector (million barrels per day)](image)

Given the innumerable motivations to reduce energy consumption, specifically in the transportation sector, the automotive industry has been under extreme scrutiny to produce more and more efficient vehicles. At present, the aggregate fuel conversion efficiency of the transportation industry approaches a mere 20%. The conversion efficiency of the electric power industry is not much better, standing around 30% [3]. Governmental, environmental and consumer pressures have ushered in the dawn of a new era in terms of fuel economy requirements and expectations. Figure 3 outlines the current, proposed and estimated fuel economy mandates in four major world markets.
As Figure 3 evidences, complying with the progressively stringent regulations on corporate average fuel economy (CAFE) represents one of the main challenges that is currently being faced by the automotive industry.

**Section 2.2 Proposed Solutions**

Auto makers have long sought to increase fuel efficiency in their vehicles through several means such as alternative fuels, design improvements in vehicle chassis, body and powertrain and, more recently, the introduction of electrified and hybrid powertrains. Though advances in technology are sure to benefit future generations of vehicles, many options exist today that do not require significant design changes. For example, some low-cost alternatives to full hybrid vehicle architectures are the so called “mild-hybrids”
or “micro-hybrids”, which allow for regenerative braking, engine downsizing with boosting, fuel shut off and engine start-stop with a significantly simplified powertrain electrification. This section will give a general overview of the technologies and methods proposed to reduce fuel consumption across the transportation industry.

2.2.1 Sources of Inefficiency

To understand how to fix a problem, one must first understand the problem. In this case, the problem is the inefficiency involved with converting an energy source (i.e. fuel) into motive force in a vehicle. The sources of inefficiency are interconnected, compounding and numerous. Figure 4 breaks down the energy losses over two drive cycles for a mid-sized sedan. As one can see, only 20-30% of the fuel energy is actually available to move the vehicle.
Figure 4: Energy utilization in a mid-sized sedan [1]

Given the energy usage breakdown depicted in Figure 4, efficiency improvements can be garnered through any combination of the following three ways:

1. Improve the efficiency of the energy conversion device itself as to maximize the available brake energy. Generally these would be considered propulsion related improvements and could range from friction reduction to thermodynamic cycle optimization.

2. Reduce powertrain and vehicle parasitic losses. Grouped as non-propulsion related technologies, these improvements increase the overall efficiency of a vehicle by maximizing the conversion of brake energy to motive force.
3. Recover energy generated through inherent inefficiencies that would otherwise be lost. In some cases, a more appropriate alternative to improving the efficiency of a system is to capture the energy created by said inefficiency (i.e. exhaust gas heat recovery).

2.2.2 Propulsion Related Improvements

The subject of propulsion-related technologies that include improvements in fuel economy is a massive one. Control development such as valve timing to fundamental design changes such downsizing should all included. To this end, an effort will be made to select some of the more promising ideas and technologies rather than flood the document with information. More detail will be included in the Energy Saving and Recover section because those topics relate more directly to the work presented in Chapter 4.

*Spark-Ignition Engines*

A significant percentage of the vehicles on the road today are powered by spark-ignition (SI) engines fueled with gasoline. This section will focus on technologies that do not adversely affect existing drivability or performance metrics, and can theoretically be implemented in the near term.

*Cylinder Deactivation*

Cylinder deactivation may be used during partial load operation to mitigate throttling and thermal losses. When some cylinders are deactivated, the remaining cylinders incur a higher load to overcome the friction and pumping losses of the inactive cylinders and...
maintain the desired power output. This higher load requires a larger throttle opening and hence lower throttling losses. Typically, cylinder deactivation is achieved through valve deactivation which effectively seals the cylinders until called upon. This causes the deactivated cylinders to behave as air springs, compressing and expanding the trapped charge and, ultimately, marginally increase the resistive load at the engine crankshaft. Due to NVH concerns, cylinder deactivation has primarily been implemented on engines with at least six cylinders [4]. Tests performed by FEV show that FC reduction of 7% is possible for a V8 on the New European Drive Cycle (NEDC) [5].

**Gasoline Direct Injection (GDI)**

GDI, also known as direct injection spark ignition (DISI), is a process in which fuel is vaporized and injected directly into the combustion chamber rather than into the intake port above the intake valve. More complete fuel vaporization reduces knock tendency and hence increases the knock-limited compression ratio. This higher compression ratio directly improves cycle efficiency [6], as can be qualitatively shown by looking at the thermal efficiency for an ideal constant-volume thermodynamic cycle (Otto cycle):

$$\eta = 1 - \frac{1}{\gamma - 1}$$

(2.1)

GDI engines can be designed to increase compression ratio by 1.0 to 1.5, which results in FC reductions ranging from 1.5 to 3 percent [4]. Stoichiometric direct injection also affords the use of more effective and low-cost three-way catalyst with standard closed-loop emissions control systems. GDI engines typically function un-throttled which further improves their efficiency.
**Turbocharging and Downsizing**

Turbocharging and downsizing reduces engine mass and pumping losses [7]. It is well known that smaller engine displacement leads to lower friction losses (FMEP). On the other hand, if the number of cylinders is reduced (i.e. V6 to I4), countermeasures may be necessary to combat NVH issues. In addition, turbocharging or supercharging is required to restore the maximum performance of the engine, avoiding penalty during accelerations. NESCCAF claims a 6-8% reduction in fuel consumption through the use of downsizing and turbocharging [8]. Most estimates of fuel consumption benefits assume that the vehicle is equipped with some sort of GDI which is typically required to reduce knock in downsized engines.

**Valve-Event Modulation (VEM)**

VEM refers to a broad range of specific technologies that affect the volumetric efficiency of an SI-engine intake system by way of altering valve timing. Examples are variable value timing, variable valve lift, cam phasing and intake value throttling. VEM technologies reduce pumping losses and consequently increase brake torque which allows for engine downsizing given a constant torque requirement. FC reduction can range from 1% using just intake cam phasing to 11% using variable valve timing and lift [4].
Compression-Ignition Engines

Compression-ignition (CI) engines are fundamentally more efficient than their SI counterparts for three reasons:

1. Lean fuel-air mixtures
2. No throttling of intake charge
3. Higher compression ratios

Widespread proliferation of light-duty diesel vehicles has not occurred in the United States primarily for emissions reasons, NOx and particulate matter specifically [4]. However, recent improvements in diesel exhaust gas after treatment may have ushered in the dawn of a new era in terms of CI market share in light-duty vehicles. At the 2008 Detroit Auto Show, 12 OEMs introduced 13 new diesel-powered vehicles for the US market [9].

Many of the technologies and concepts being explored to improve the efficiency of SI engines also apply to CI engines (except GDI of course). These improvements fall into the following four broad categories.

1. Downsizing the engine without compromising power
2. Improving thermodynamic cycle efficiency
3. Reducing engine friction
4. Reducing ancillary load losses

Specific CI technologies will not be covered in this document because the most tangible and immediate influence of the CI-engine is simply the displacement of the SI-engine.
Ricardo Inc. at the request of the EPA carried out a full system simulation (FSS) in 2008 to evaluate the FC reduction potential of CI-engines. In the study, SI-engine powertrains were replaced by downsized CI-engine powertrains with similar performance and FC results were compared to the SI-engine baseline vehicles. The findings are summarized in Table 1 for three vehicle classes.
Table 1: Estimated FC reduction for three EPA vehicle classes [10]

<table>
<thead>
<tr>
<th>Vehicle</th>
<th>Major Features</th>
<th>SI to CI Downsize Ratio</th>
<th>Combined Fuel Consumption Reduction</th>
</tr>
</thead>
<tbody>
<tr>
<td>Full-size car</td>
<td>3.5L VG SI gasoline, AT5</td>
<td>80%</td>
<td>Baseline</td>
</tr>
<tr>
<td></td>
<td>2.8L I4 diesel, DCT6, EACC, HEA, EPS</td>
<td></td>
<td>33.2%</td>
</tr>
<tr>
<td>Small MPV</td>
<td>2.4L I4 gasoline SI, DCP, EPS, AT4</td>
<td>79%</td>
<td>Baseline</td>
</tr>
<tr>
<td></td>
<td>1.9L I4 diesel, DCT6, EACC, HEA, EPS</td>
<td></td>
<td>31.9%</td>
</tr>
<tr>
<td>Truck</td>
<td>5.4L V8 gasoline SI, CCP, AT4</td>
<td>89%</td>
<td>Baseline</td>
</tr>
<tr>
<td></td>
<td>4.8L V8 diesel, DCT6, EACC, HEA, EPS</td>
<td></td>
<td>34.1%</td>
</tr>
</tbody>
</table>

Notes: AT5 - automatic 5 speed lockup transmission, AT4 - automatic 4 speed lockup transmission, CCP - coordinated cam phasing, DCP - dual cam phasing, DCT6 - dual clutch six speed automated manual transmission, EACC electric accessories, EPS - electric power steering, HEA - high efficiency alternator

The fuel consumption gains listed in Table 1 are due to more than just supplanting SI-engines with CI-engines; the gains are the result of an improved powertrain which includes downsizing and an advanced transmission.

**Hybrid Powertrains**

Emissions and fuel economy standards have ushered in a new era in powertrain configurations - hybrids. The term "hybrid" refers to the vehicles that use multiple power sources to either directly or indirectly provide propulsion. Although hybrid-electric is the most common, other secondary power sources can be used such as hydraulic hybrids, and fuel cell hybrids. Variations in hybrid architecture are vast, but most still include some form of internal combustion. The relative impact of a hybrid's ICE determines the degree of hybridization which ranges from mild hybrids, such as those using E-assist, to full EVs like the Nissan Leaf.
Hybrid powertrains can increase fuel efficiency through four following ways [11]:

1. **Regenerative Braking** allows some of the braking energy that is typically dissipated through friction brakes as heat to be recovered and used to offset some of the power required later. This concept is heavily employed in industry and represents one of the primary reasons to consider hybrid architectures.

2. **Idling Reduction** involves shutting off the ICE at low speeds so long as the secondary power source is sufficient. Engine idling when a vehicle is not moving is technically a zero-efficiency condition and is therefore an unnecessary waste of energy.

3. **Engine Efficiency Improvement** is possible by allowing a secondary energy source to displace some of the power required from the ICE, so that the engine can continually operate in a more efficient region. This idea typically applies at very high or very low engine speeds.

4. **Downsizing** of the ICE is possible due to the existence of a secondary power source that can provide motive force.

With the addition of a second power source, a variety of system architectures have been developed effectually utilize the added degrees of freedom in terms of power production and recuperation. The three main architecture categories are shown pictorially in Figure 5.
The fuel consumption benefits of hybrid powertrains can be realized by comparing hybrid versions a vehicle to its conventional counterpart. In Table 2, metrics for the hybrid are divided by the same metric for a comparable conventional vehicle [4].

**Table 2: Comparison of fuel economy and performance for hybrid vs. conventional vehicles [4]**

<table>
<thead>
<tr>
<th>Architecture</th>
<th>EPA FE [mpg]</th>
<th>FC [gal/100 miles]</th>
<th>0 to 60 mph Acceleration</th>
</tr>
</thead>
<tbody>
<tr>
<td>Prius/Camry</td>
<td>2.00</td>
<td>0.50</td>
<td>1.10</td>
</tr>
<tr>
<td>Civic Hybrid/Civic SI</td>
<td>1.51</td>
<td>0.66</td>
<td>1.16</td>
</tr>
<tr>
<td>Tahoe Hybrid/Tahoe SI</td>
<td>1.53</td>
<td>0.65</td>
<td>1.07</td>
</tr>
</tbody>
</table>
2.2.3 Non-Propulsion Related Improvements

Although many technologies are available today to improve the efficiency of vehicle energy conversion devices, some of the most promising improvements are not related to the propulsion system at all. Five broad areas for improvement are aerodynamics, mass reduction, tires, HVAC and transmissions. These will be covered below.

Aerodynamics

The drag force experienced by a vehicle scales with the square of velocity and is defined as

\[ F_{\text{aero}} = \frac{1}{2} C_d A V^2 \]  \hspace{1cm} (2.2)

where \( C_d \) is the drag coefficient and \( A \) is frontal area. Typically \( C_d \) ranges from 0.25 to 0.38 on production vehicles and depends on numerous factors ranging from external mirrors to spoilers [4]. Achieving \( C_d \) values at or below 0.25 for most passenger cars generally requires modifications that are impractical or too costly for production vehicles. That being said however, vehicles with higher \( C_d \) values (0.30 or greater) could reduce \( C_d \) by up to 10% without significant revision or cost. The following are some vehicle design changes that lower \( C_d \) and therefore decrease aerodynamic losses.
1. General vehicle shape
2. Elimination of external rear-view mirrors
3. Improved air inlets
4. Retractable rear spoilers
5. Wheel well covers
6. Enclosure of vehicle underbody

Calculations done by Argonne National Lab suggest that a 10% reduction in aerodynamic drag would result in a 0.25% decrease in FC on an urban cycle and a 2.15% decrease in FC on a highway cycle due to the higher speeds as cited in [4]. Reduction in drag force could come from an improved $C_d$, smaller frontal area or a combination of both.

**Mass Reduction**

The primary benefit of mass reduction stems directly from Newton's second law; less energy is required to change the linear momentum of a vehicle if it weighs less. In the automotive industry, it is assumed that mass reduction is only possible if noise/vibration/harshness (NVH), crashworthiness and comfort are not adversely affected. The two main methods of mass reduction in vehicles are changing the design to require less material, or substituting lighter materials in the place of heavier ones [4]. Some examples of the later would be replacing mild steel with high-strength steel so a smaller gauge can be used, or trading steel for aluminum. It is important to note that material changes often require component redesign and manufacturing alterations the costs of which may be prohibitive. In recent years, vehicle body structure changes have allowed the introduction of lighter materials into the manufacturing process. The four primary body design methodologies are described below.
1. **Body-on-frame** designs have an independent body structure that sits on top of a structural frame. Both the body and frame have structural integrity and therefore this design offers the most strength and stiffness. For these reasons, heavy trucks and SUVs still use this design despite weight.

2. **Unibody** designs use the internal vehicle components as the load-bearing structures while the external panels do not provide any structural advantage. This method replaced the body-on-frame design for most vehicle types due to its decreased weight; a separate load-bearing structure is not required. Unibody vehicles are still steel-based and are designed for high volume production.

3. **Space frame** designs have been recently developed to accommodate more aluminum which cannot be formed or joined in the same ways as steel. Space frames consist of extruded aluminum connected at the ends in what are called nodes. Non-structural exterior panels "hang on" to the nodes [4].

4. **Monocoque** designs rely on the exterior surface of the vehicle for structural support.

This method typically uses composites and currently more common in aircraft and ships. Though space frame and monocoque vehicles offer the advantage of lighter materials, the higher material costs have rendered steel unibody by far the most prevalent design. With this in mind, reducing the weight in steel-based vehicles is very important. Some ways are:

1. Substitute higher-strength steel for lower-strength steel;

2. Use sandwich metal panels instead of pure steel;
3. Introduce new manufacturing processes that allow for more design optimization. Although most of a vehicle's mass is associated with the structural and propulsion aspects, the non-body components such as glass and interiors (i.e. electronics and seats) also offer mass-reduction possibilities.

In addition to the direct benefit of structural mass reduction, a secondary benefit is the lower weight of the powertrain required to move a fundamentally lighter vehicle. A study done by the Department of Energy provides a general relationship between mass reduction and fuel consumption; for every 10% reduction in mass (primary and secondary included), fuel consumption is reduced by 6-8% [12]. The effect of mass reduction is more pronounced in city driving due to the higher frequency of acceleration/braking events. Table 3 summarizes average fuel consumption findings from a study done by Ricardo Inc. taking into account different vehicle types and driving cycles.

<table>
<thead>
<tr>
<th>Vehicle Mass Reduction from Baseline</th>
<th>5%</th>
<th>10%</th>
<th>20%</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass reduction only</td>
<td>1-2%</td>
<td>3-4%</td>
<td>6-8%</td>
</tr>
<tr>
<td>Mass reduction and resized engine</td>
<td>3-3.5%</td>
<td>6-7%</td>
<td>11-13%</td>
</tr>
</tbody>
</table>

The FC gains shown in Table 3 make mass reduction the single most effective way to improve fuel economy.
**Tires**

Rolling resistance force is generated from the continuous deformation of rotating tires. The amount of retarding force produced depends on tire design and inflation. Underinflated tires create more rolling resistance which in turn increases fuel consumption. Tire manufacturers are constantly trying to decrease the rolling resistance of their tires without compromising performance. A report by NRC in 2006 suggests that a 10% reduction in rolling resistance will reduce fuel consumption by 1-2% [13].

**HVAC**

Heating, ventilation and air conditioning (HVAC) represents one of the largest mechanical ancillary loads on an engine. Many technologies exist that could improve the efficiency of the AC system including larger heat exchangers, variable displacement compressors and pulse-width modulated blower controllers [4]. The load on the AC system may be further decreased through the use of reflective paint and cabin ventilation while parked. A variable-stroke compressor coupled with improved humidity controls has the potential to decrease fuel consumption by 3-4%.
Transmissions

Improved transmissions can reduce fuel consumption in two ways:

1. Allow the engine to operate in a more efficient region of the engine map. Higher torque at lower speeds is generally the most efficient operating region for an ICE (see Figure 6) because the relative effect of engine friction losses is reduced.

![Sample engine efficiency map](image)

*Figure 6: Sample engine efficiency map (2 L, 82 mm stroke) [14]*

To reside in this region more often, additional gears are being added to transmissions and the final drive ratio is being reduced.

2. Improvement of the mechanical efficiency of the transmission itself.

Of these two goals, the first holds the most promise and is being pursued by a number of OEMs. In addition to automatic transmissions with more gears, dual-clutch automated
manual transmissions (DCTs) are being introduced as high efficiency manual alternatives. The lack of a torque converter in DCTs gives rise to their high efficiencies since traditional clutches slip less as compared to torque converters. Both next generation automatics and DCTs offer a finite number of gear ratios. To that end, continuously variable transmissions (CVTs) feature a theoretically infinite number of gear ratios in a given range. The concept of a CVT allows for significant optimization of the engine operating points to reduce fuel consumption, but high costs, torque limits and low mechanical efficiencies have slowed market penetration [12]. Table 3 lists estimated FC improvements for various transmission technologies when compared to a naturally-aspirated SI-engine vehicle with a 4-speed automatic of similar performance characteristics.

Table 4: Transmission technologies and estimated FC reduction

<table>
<thead>
<tr>
<th>Technology</th>
<th>FC Reduction [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>5-speed automatic</td>
<td>2-3</td>
</tr>
<tr>
<td>6-speed automatic</td>
<td>3-5</td>
</tr>
<tr>
<td>7-speed automatic</td>
<td>5-7</td>
</tr>
<tr>
<td>8-speed automatic</td>
<td>6-8</td>
</tr>
<tr>
<td>6-speed DCT</td>
<td>6-9</td>
</tr>
<tr>
<td>CVTs</td>
<td>1-7</td>
</tr>
</tbody>
</table>

2.2.4 Energy Saving, Recovery and Storage

Although many technologies are being developed to improve the efficiency of vehicles from a power-production stand point, some of the most promising improvements lie in the realm of recovering energy that is typically wasted. Energy recovery is the general theme of this work.
Engine Stop/Start

Stop-start, sometimes called idle-stop, involves turning off the engine under idle conditions and is therefore most useful in city driving situations. Eliminating idle fuel consumption can generate significant savings, especially in extreme stop-and-go traffic conditions. On a Federal Test Procedure (FTP) cycle where the idle events are well defined, the fuel savings range between 3 and 5% but for congested city driving, this reduction can approach 10% [4]. It should be noted that stop-start is more effective in SI-engine vehicles because idle fuel consumption in SI-engines is higher than that of CI-engines because of throttling losses. A study of stop/start benefits based on simulation fuel consumption results is included in Chapter 4.

Ancillary Load Reduction

The main function of a vehicle's prime mover is obviously to provide motive force to the wheels. However, a significant burden is placed on the engine by ancillary loads that are required by the numerous systems and conveniences extraneous to vehicle propulsion. A study conducted in [15] suggests that up to 8% of fuel energy is used to power these auxiliary loads. The A/C system and engine cooling system represent the largest energy sinks as reported by [16][17][18][19], and can account for power consumptions of up to 4 kW and 1.5 kW respectively [20][21][22]. Some other ancillary loads include power steering and the various intermittent electrical loads imposed by the headlights, heated seat, infotainment system, etc. Since most of the ancillary loads in a conventional vehicle are coupled directly to the engine crank shaft via a serpentine belt, they inherit engine
speed as an operating point which in many cases is not ideal. Advances in electrification of loads and implementation of loads that can be declutched have can reduce some of these issues.

Exhaust Gas Waste Heat Recovery

With reference to Figure 4, a significant percentage of the energy released through combustion is wasted as heat in the exhaust stream. A way to utilize this otherwise wasted energy is to heat up the working fluids of a vehicle upon cold starts. An exhaust heat recovery system (EHRS) can be used to heat up engine coolant and oil, as well as transmission fluid, to minimize the time it takes for those fluids to reach their steady state temperatures [23]. Since oils and coolants have viscosities that decrease with increases in temperature, rapid warm-up can mitigate some of the powertrain losses due to friction. This goal of decreasing warm-up time is precisely the Master Thesis developed by Neeraj Agarwal [24]. Results reported by [25] suggest that fuel consumption can be up to 10% higher over a NEDC for a cold start case (20C) as compared to a warmed-up case (90C).

Regenerative Braking and Crankshaft Energy Harvesting

Regenerative braking refers to the capture of energy during braking that would otherwise be lost as heat to the environment in a conventional vehicle. In hybrid-electric vehicles, this energy is converted into electricity though use of an electric machine (EM) working as a generator. The energy is stored in a battery system and can then be used to provide
motive force by using the EM as a motor. A consideration that must be made when calculating the benefits of regenerative braking is that the systems required for full-scale regenerative braking (i.e. EM, battery, power electronics) can easily add to vehicle's mass, up to 5% in some cases [26]. Crankshaft energy harvesting employs essentially the same concept as regenerative braking, but the energy is extracted at the engine-level and may be used immediately for reasons other than secondary energy storage. These could include battery charging, A/C system pressure increases or additional engine cooling via the radiator fan. Crankshaft energy harvesting is especially applicable to conventional powertrains because no dedicated systems exist for chassis-level regenerative braking.

**Energy Storage**

The main technological hurdle preventing the total proliferation of hybrids is energy storage. Presently, no energy storage method can even begin to rival the energy density of liquid fuels. Despite the gross thermodynamic inefficiency of the internal combustion engine, its ability to utilize liquid fuels has rendered it, by far, the primary energy conversion device used in vehicles throughout the world. That being said, significant efforts are underway to improve the characteristics of alternative energy storage methods with the hope that one day they will be able to compete with fossil fuels. Table 5 summarizes some of the available technologies that are, or could be used to store energy in a vehicle platform.
This section will focus on three: flywheels, batteries and hydraulic accumulators.

Batteries were chosen because they currently represent the primary alternative to fuels.

Flywheels and hydraulic accumulators were selected because of their fundamental dissimilarity to batteries, as well as the fact that they have begun to appear recently in the hybrid vehicle market.

**Battery Storage**

Batteries are electrochemical energy storage devices that are comprised of individual cells that are connected together. Within these cells is a conducting medium called the electrolyte. The two main types of batteries are primary and secondary. Primary batteries cannot be recharged while secondary batteries can. Hybrid powertrains therefore contain secondary batteries. Within a battery, either primary or secondary, multiple cells are connected in series to obtain what are known as strings. These strings sum the voltages of the cells they contain and are connected in parallel so that the battery current is divided amongst them. The overall pack voltage and pack capacity (energy) depend on the length of the strings, and the total number/size of the individual cells. In addition to the voltage and current ratings, secondary batteries are further divided into

---

### Table 5: List of some energy storage technologies

<table>
<thead>
<tr>
<th>Energy Storage Type</th>
<th>Energy Storage Technology</th>
</tr>
</thead>
<tbody>
<tr>
<td>Electrochemical</td>
<td>Batteries</td>
</tr>
<tr>
<td></td>
<td>Fuel Cells (Hydrogen)</td>
</tr>
<tr>
<td>Electric Field</td>
<td>Supercapacitors</td>
</tr>
<tr>
<td>Magnetic Field</td>
<td>Inductors</td>
</tr>
<tr>
<td>Kinetic</td>
<td>Flywheels</td>
</tr>
<tr>
<td>Chemical</td>
<td>Liquid Fuels</td>
</tr>
<tr>
<td>Hydraulic</td>
<td>Accumulators</td>
</tr>
</tbody>
</table>
ambient temperature and high temperature categories based on the operating temperature of the electrolyte [27].

Many different battery types exist, each with their own specific advantages and disadvantages. In addition to energy and power density, practical and economic factors influence the use, or lack of use, of any given battery type. Table 6 summarizes the general pros and cons of the four most widely-used battery technologies. Nickel-Metal Hydride and Lithium Ion are the types typically used in hybrid vehicles.

Table 6: Summary of Secondary Battery Types [27]

<table>
<thead>
<tr>
<th>Battery Type</th>
<th>Advantages</th>
<th>Disadvantages</th>
</tr>
</thead>
<tbody>
<tr>
<td>Lead Acid</td>
<td>Can be designed for high power</td>
<td>Poor Cold temperature Performance</td>
</tr>
<tr>
<td></td>
<td>Inexpensive</td>
<td>Short Calendar and Cycle Life</td>
</tr>
<tr>
<td></td>
<td>Safe</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Reliable</td>
<td></td>
</tr>
<tr>
<td></td>
<td>OLD ESTABLISHED TECHNOLOGY as starter battery</td>
<td></td>
</tr>
<tr>
<td>Nickel-Cadmium</td>
<td>High Specific Energy</td>
<td>Does not deliver sufficient power</td>
</tr>
<tr>
<td></td>
<td>Good Cycle life compared with lead acid</td>
<td></td>
</tr>
<tr>
<td>Nickel-Metal Hydride</td>
<td>Reasonable Specific Energy</td>
<td>High Cost</td>
</tr>
<tr>
<td></td>
<td>Reasonable Specific Power</td>
<td>Heat Generation at High Temperatures</td>
</tr>
<tr>
<td></td>
<td>Much longer cycle life than lead acid</td>
<td>Low cell efficiency</td>
</tr>
<tr>
<td></td>
<td>Safe</td>
<td>Need to control Hydrogen Losses</td>
</tr>
<tr>
<td></td>
<td>Abuse-tolerant</td>
<td></td>
</tr>
<tr>
<td>Lithium Ion</td>
<td>High Specific Energy</td>
<td>Needs Improvement in:</td>
</tr>
<tr>
<td></td>
<td>High Specific Power</td>
<td>Calendar and Cycle life</td>
</tr>
<tr>
<td></td>
<td>High Energy Efficiency</td>
<td>Abuse Tolerance</td>
</tr>
<tr>
<td></td>
<td>Good High temperature performance</td>
<td>Acceptable Cost</td>
</tr>
<tr>
<td></td>
<td>Low Self-Discharge</td>
<td>Higher degree of Battery safety</td>
</tr>
</tbody>
</table>

*Flywheel Storage*

Flywheels store kinetic energy and can be used to either provide a continuous flow of energy when the energy source is intermittent in nature, or provide bursts of power by "clutching in" an already spinning flywheel. Flywheel systems for use in hybrid drive
applications typically utilize a CVT to connect the flywheel shaft to a drive axle. An example from Land Rover is shown in Figure 7.

![Component Details](image)

*Figure 7: Flywheel hybrid system with CVT [28]*

The energy stored in a flywheel is proportional to the square of its rotational speed.

\[
E = \frac{1}{2} I \omega^2
\]  

(2.3)

The torque required to increase the speed can be come from either the engine, or from the vehicle during a braking event. Once up to speed, the flywheel can transmit energy back to the powertrain when requested.
Hydraulic Storage

Hydraulic accumulators store energy by compressing an inert gas (usually nitrogen) which effectively acts as a compressed spring. Hydraulic fluid is pumped into one side of a chamber that is separated into sections by way of an impervious barrier; gas on one side, fluid on the other. This barrier is typically a bladder or a piston. The amount of energy stored is proportional to the pressure and volume of the compressible fluid, and power is related to pressure and volumetric flow rate.

\[
E = \Delta pV = p_2V_2 - p_1V_1
\]
\[
P = pQ = p \frac{dV}{dt}
\]  
(2.4)

The stored energy can be released by allowing the hydraulic fluid to leave the accumulator and flow back into a pump/motor. Parker Hannifin has found that their hydraulic accumulator system is capable of recovering and reusing up to 70% of braking energy [29].

Comparison of Energy Storage Methods

All energy storage methods have advantages and drawbacks. The two key characteristics that define a technology's use are its specific energy and specific power. In terms of hybrid vehicles, specific energy is directly related to range while specific power is directly related to performance. It is important to note that no perfect solution exists. Flywheels and hydraulic accumulators possess significant specific power potential but lack the necessary energy storage capability. Batteries have the opposite problem. Though they cannot compete with conventional liquid fuels, batteries have appreciable
specific energy but suffer with specific power. A Ragone plot in Figure 8 displays a number of different energy storage devices as a function of specific energy and power.

![Ragone plot showing energy storage devices](image)

**Figure 8: Specific energy vs. specific power of energy storage technologies [30]**

Range requirements have dictated the widespread use of batteries over other energy storage methods. In all actuality, the most effective solution would be a combination of technologies. A promising example would be the marriage of batteries and a flywheel. Batteries could provide range while the flywheel would enhance acceleration performance and reduce the burden on the prime mover. An example of this concept in practice is the use of two different types of batteries, one with high specific energy, and one with high specific power. Power electronics and feedback control are then used to
select the energy source based on the driver demands. Though a combination of energy storage methods is probably the best way to alleviate the shortcomings of the individual technologies, too much complexity could be counterproductive if reliability issues and cost outweigh the benefits. Table 7 summarizes from a very high level the principal advantages and disadvantages, as well as appropriate applications of the three energy storage methods in question. Costs have been ignored.

Table 7: High-level summary of energy storage methods

<table>
<thead>
<tr>
<th>Pros</th>
<th>Flywheels</th>
<th>Hydraulic Accumulators</th>
<th>Batteries</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Power density</td>
<td>Power density</td>
<td>Energy density</td>
</tr>
<tr>
<td></td>
<td>Size and weight</td>
<td>Reliable</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Dependable</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Conversion efficiency</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Environmentally friendly</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Cons</td>
<td>Energy density</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Apps</td>
<td>High performance vehicles</td>
<td>Large, heavy vehicles</td>
<td>Passenger vehicles</td>
</tr>
<tr>
<td></td>
<td>Urban stop and go driving</td>
<td>Urban stop and go driving</td>
<td>Engine off situations</td>
</tr>
<tr>
<td></td>
<td>Vehicle launch</td>
<td></td>
<td>Range extension</td>
</tr>
</tbody>
</table>
Chapter 3: Model Development, Calibration and Validation

As part of a large research project conducted between The Ohio State University and Chrysler, LLC, a comprehensive Vehicle Energy Simulator (VES) was developed and validated on experimental data, to provide a tool for energy analysis and control design purposes. The eventual goal of the project is to use the VES as a platform for developing a Vehicle Energy Management (VEM) strategy with the overarching goal of reducing vehicle fuel consumption. The simulator is a forward-looking model that captures low frequency energy and power dynamics with the primary objective of fuel economy prediction. The VES compiles models developed by numerous parties at Ohio State and represents a collective effort of the Chrysler project team. This section will focus primarily on the development of the powertrain models, selected thermal models, the implementation of production engine and transmission control algorithms (i.e. shift schedule) in the VES, and the development of a driver model. A brief description of the remaining models will be given with reference to more complete works. An overview of the experimental setup is included as well as calibration details where applicable.

Section 3.1 Experimental Setup

An engine dynamometer and a chassis dynamometer at the Center for Automotive Research (CAR) are the primary methods of data collection for the validation of the VES
model. In addition, tests were performed in the vehicle on the road by way of a data acquisition (DAQ) system powered by the vehicle's battery. A van fundamentally identical to the one allocated to OSU was retrofitted with a slightly different thermal system architecture at Chrysler to provide non-baseline configuration thermal data.

3.1.1 Engine Dynamometer

A double-ended 300 HP AC dynamometer (dyno) was available at the Ohio State University Center for Automotive Research to conduct engine testing in a controlled laboratory environment. The test cell is equipped with a Horiba Mexa 7500 emission analyzer and a 128-channel data acquisition system. An image of the dynamometer room is shown below in Figure 9 and a schematic of the DAQ system follows in Figure 10.

Figure 9: Engine dynamometer room
For this project a V6, 3.6l naturally aspirated spark-ignition engine was used to generate experimental data for model validation. The engine, whose specifications are listed in Table 8, is a production unit that is installed on the 2012 Chrysler Town & Country Minivan.

**Table 8: Engine specifications**

<table>
<thead>
<tr>
<th>Specification</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Displacement [L]</td>
<td>3.6</td>
</tr>
<tr>
<td>No. of Cylinders</td>
<td>6</td>
</tr>
<tr>
<td>Bore [mm]</td>
<td>96</td>
</tr>
<tr>
<td>Stroke [mm]</td>
<td>83</td>
</tr>
<tr>
<td>Compression Ratio</td>
<td>10.2</td>
</tr>
<tr>
<td>Max Torque [Nm @ rpm]</td>
<td>353 @ 4400</td>
</tr>
<tr>
<td>Max Power [kW @ rpm]</td>
<td>216 @ 6350</td>
</tr>
</tbody>
</table>
The experimental data acquired on the engine dynamometer have been used to calibrate and validate models of the main engine components and subsystems. A list of important tests is included in Table 9, together with the principal motivations and expected outcome of each test.

One of the main experiments conducted on the test engine was a collection of steady-state points that fills completely the engine operating range, in terms of speed and torque. This test is a standard procedure accepted in industry to provide a preliminary performance characterization of an engine in its entire operating domain. At Chrysler, this testing procedure is named “Big Grid”, and from this point forward, this term will be used to denote the data set generated according to this testing procedure.

Figure 11: Big Grid test points
The Big Grid data set available for this work contains several engine operating variables (pressures, temperatures, flow rates, etc...) at 247 different engine speed and torque combinations.

**Table 9: Engine dyno testing summary**

<table>
<thead>
<tr>
<th>Test</th>
<th>Motivation</th>
</tr>
</thead>
</table>
| Constant speed and load warm-up | 1. Validate engine warm-up curves  
                              | 2. Verify correct engine calibration                                     |
| Control valve characterization | Validate external cooling circuit                                           |
| Big Grid data points      | 1. Validate the steady state data received from Chrysler  
                              | 2. Add points not covered by the Big Grid                                |
| FMEP characterization  | Characterize FMEP based on engine speed, torque and oil temperatures         |
| Catalyst testing        | Calibration of the exothermic heat release submodel                        |

3.1.2 Chassis Dynamometer

The chassis dyno at CAR is fitted with two 24" rolls and is designed for vehicle testing up to 150 HP. A driver's aid station running a LabVIEW GUI allows a driver to follow both standard and user-defined drive cycles. A schematic and image of the in-vehicle DAQ system are displayed in Figure 12.
The laptop used in the DAQ system runs INCA, which is an interface for communication with vehicle ECUs. Signals can be added and removed from experiments based on naming conventions defined by Chrysler. The recorded data is exported from INCA in a .dat file which is later converted into a .mat file using a post processing script. Once in the .mat format and loaded into MATLAB, the signals can be used directly.
The test vehicle used to generate experimental data is a 2012 Chrysler Town & Country Minivan. Specifications of the test vehicle are included in Table 10.

Table 10: Vehicle specifications

<table>
<thead>
<tr>
<th>Make, Model and Year</th>
<th>2011 Chrysler Town &amp; Country</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass</td>
<td>2154 kg</td>
</tr>
<tr>
<td>Frontal Area</td>
<td>2.42 m²</td>
</tr>
<tr>
<td>Aerodynamic Drag Coefficient (Cₐ)</td>
<td>0.33</td>
</tr>
<tr>
<td>Gear Ratios (1-6)</td>
<td>4.127 - 2.842 - 2.284 - 1.452 - 1.000 - 0.690</td>
</tr>
<tr>
<td>Final Drive</td>
<td>3.16</td>
</tr>
<tr>
<td>Tire Radius</td>
<td>0.3514 m</td>
</tr>
<tr>
<td>Engine</td>
<td>3.6L V6 SI (see engine spec table)</td>
</tr>
<tr>
<td>Transmission</td>
<td>62TE 6-Speed Automatic Transmission</td>
</tr>
</tbody>
</table>
Road Load Calibration

The purpose of a chassis dynamometer is to simulate, in a controlled environment, the loads experienced by a vehicle on the road. In order for tests to accurately represent actual driving conditions (on the road), the load felt by the vehicle on the dyno at a given velocity must reflect the real vehicle usage conditions. To this end, internal friction of the dyno motors, bearings etc. plus tire deformation forces must be accounted for as additional braking forces that are not present on the road. Moreover, the inertia of the dyno rolls will change the shape of the coast down curves. Calibration of the road load profile imposed by the dyno is therefore required. The first step of this process is to determine the road load characteristics of the vehicle itself (see vehicle model description for more details) which results in an equation of the form [31]:

\[
F = a_0 + a_1 V + a_2 V^2 + 0.5 \rho_{air} C_d A_{frontal} V^2 + mg \sin(\theta)
\]  

(3.1)

By grouping related coefficients in (3.1), a simplified form can be derived:

\[
F = \left( a_2 + 0.5 \rho_{air} C_d A_{frontal} \right) V^2 + \left( a_1 + mg \sin(\theta) \right)
\]

\[
F = C^* V^2 + B^* V + A^*
\]

(3.2)

English system units:

F{lbf}, V{mph}, A*[lbf], B*[lbf/mph], C*[lbf/mph^2]

where F represents an aggregate friction force that is purely a function of velocity. In the United States, it has become common practice to represent (3.2) in an alternative form that can be used to intuitively compare vehicles.

\[
P = C + B + A
\]

(3.3)
In (3.3), each term has units of horsepower at 50 mph, or "HP@50" shorthand. The $A$, $B$, $C$ coefficients must be in these units to be compatible with the dyno computer and are obtained through multiplying (3.2) by velocity to attain power, plugging in 50 mph for velocity and then converting from [lbf · mph] to horsepower. These relations can be simplified to:

1. $C^* \text{[lbf/mph]}^2 = 0.003 \times C \text{[HP@50]}$
2. $B^* \text{[lbf/mph]} = 0.15 \times B \text{[HP@50]}$
3. $A^* \text{[lbf]} = 7.5 \times A \text{[HP@50]}$

The summation of $A+B+C$ gives the total horsepower required to overcome all road loads at 50 mph. The **second step** in the calibration process is to characterize the internal dyno friction as a function of velocity. This information is obtained by first assuming zero values for the three dyno coefficients and running a coast-down test with the vehicle on the dyno from the same maximum velocity used to capture the original vehicle road load equation (typically around 60 mph). It is important to warm-up the dyno before performing the coast-down test in order to avoid transient temperature-dependent effects. Once the "zero" coefficients are established, they are subtracted from the appropriate coefficients so that the following equation holds:

---

Figure 14: Chassis dynamometer computer
This subtraction forces the dyno coefficients to be slightly lower than the original ABC coefficients to account for the quadratic dyno friction + tire deformation losses vs. velocity profile. The **third step** is to enter the *dyno* coefficients and re-run a coast down test. The dyno computer will calculated force as a function of velocity (see vehicle model calibration) and perform a second order curve fit. If the resulting coefficients expressed in units of HP@50 are sufficiently close to $A$, $B$, and $C$, the dyno friction was accurately captured and the calibration would be over. This rarely occurs. In reality, the error between the ideal coefficients and the coefficients from the fit must be calculated and re-applied to the dyno coefficients until the observed coast-down force profile is acceptable (typically requires one to two iterations). A sample calculation of all the steps described is included for reference in Table 11. It is important to note that the final coefficients apply only to the dynamometer located at CAR since it is a non-standard two-roll chassis dyno.

**Table 11: Calculation of dynamometer coefficients** [32]

<table>
<thead>
<tr>
<th></th>
<th>C [HP@50]</th>
<th>B [HP@50]</th>
<th>A [HP@50]</th>
<th>Σ [HP@50]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Highway (H):</td>
<td>8.426</td>
<td>1.300</td>
<td>6.137</td>
<td>15.863</td>
</tr>
<tr>
<td>Zero (Z):</td>
<td>1.150</td>
<td>-0.440</td>
<td>6.430</td>
<td>7.140</td>
</tr>
<tr>
<td>Dyno (D1) = H - Z:</td>
<td>7.276</td>
<td>1.740</td>
<td>-0.293</td>
<td>-</td>
</tr>
<tr>
<td>Measured (M1):</td>
<td>9.190</td>
<td>0.380</td>
<td>6.240</td>
<td>15.810</td>
</tr>
<tr>
<td>Error (e) = M1 - H:</td>
<td>0.764</td>
<td>-0.920</td>
<td>0.103</td>
<td>-</td>
</tr>
<tr>
<td>D2 = D1 + e:</td>
<td>6.512</td>
<td>2.660</td>
<td>-0.396</td>
<td>-</td>
</tr>
<tr>
<td>M2:</td>
<td>8.150</td>
<td>1.610</td>
<td>6.000</td>
<td>15.76</td>
</tr>
</tbody>
</table>

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Section 3.2 VES Structure

The VES is a comprehensive vehicle energy-based model implemented in the MATLAB/Simulink environment, and based on the engine and powertrain system of 2011 Chrysler Town & Country minivan. A top-level view of the Simulink model is shown in Figure 15. The signal directions should be noted as the order of subsystems is important. To ease the organization of signals, a bus structure has been adopted. Starting from the lowest level, each subsystem outputs a bus which is nested within the bus of the next higher level and so on. This methodology orders the signals in a certain hierarchy that allows users unfamiliar with the inner workings of the VES to locate variables.

Figure 15: Top-Level VES Simulink Structure
The six major VES-level subsystems seen in Figure 15 are:

1. **Driver** - contains the logic to mimic a human driver and follow a desired vehicle speed profile. The driver model also contains idle speed controllers and any necessary feed forward pedal commands;

2. **Vehicle Plant** - contains all of the models that describe the performance and energy usage of the main mechanical, thermal, and electrical components on the vehicle. The vehicle plant contains the vast majority of the VES models and can be recalibrated to model other vehicle platforms.

3. **Chrysler Control** - location of models of the production control algorithms (developed by Chrysler). This subsystem contains the essential control logic for engine torque management and transmission torque management, and are largely based on Chrysler proprietary data. The models of the engine and transmission control strategies are essential to properly run the models implemented for the engine, transmission and torque converter.

4. **OSU Control** - location of the control strategies for the Vehicle Energy Management (VEM) system, which will be developed by OSU. This subsystem, which is an empty block for the model validation, will host the control algorithms developed for crankshaft energy harvesting, as discussed in the following Chapter.

5. **Sensors** – this subsystem holds any conversion routines between vehicle-level signals to the units and the parameters required by the vehicle control system (in this case, the Chrysler Control block);
6. **Actuators** – this block contains conversion factors that enable the Chrysler control block to provide commands to the different actuators present in the engine and transmission models.

Of the six top-level subsystems, the only two not covered by this document are sensors and actuators. Within the vehicle plant, all of the models have been organized into four major systems: air conditioning, electrical, mechanical and thermal. Signals are passed between systems via feedback loops where needed. Figure 16 summarizes the first three levels of the VES.
Each major subsystem in the VES has its own initialization file which contains all of the necessary parameters, maps and offline calculations required by that model. Global variables were defined to organize the MATLAB workspace into structures that are directly called by the associated models (e.g. the torque converter model calls \textit{TC.variable\_name}). The structure format was also chosen to eliminate overwrite issues when multiple initialization files contain variables of the same name. To ease future control development, all of the variables throughout the VES that could be considered tunable quantities were located into an OSU control variables initialization file. This file will eventually be coupled with the OSU control block as the energy management deliverable. A separate master script runs all of the individual initialization files before the VES is used.

**Section 3.3 Individual Model Details**

In the following sections, the individual models of the four primary systems will be described in detail. Certain models have been developed by other members of the OSU team and will be briefly covered in this document for completeness. An inputs/outputs (I/O) table will be included for the models explained in more depth.

3.3.1 Air Conditioning System

The standard automotive air conditioning (AC) system contains seven primary components: compressor, discharge hose, condenser, receiver, thermal expansion valve (TXV), evaporator and a suction hose [33]. The air entering the cabin is at the
temperature of the air exiting the evaporator. Figure 17 is a schematic of a typical AC system.

Figure 17: Schematic of air conditioning system (adapted from [33])

The TXV and compressor have very fast response times and have therefore been modeled as static components. The overall dynamics of the AC system originate from heat transfer in the condenser and evaporator, which exchange heat with the ambient air and cabin air respectively, and pressure dynamics coming from the variation of density and flow rate in the system. A moving boundary modeling method was adopted to characterize the dynamics of the condenser and evaporator pressures, as well as the heat transfer processes. This modeling approach was chosen in place of a finite volume method because it provides nearly the same accuracy with a lower simulation burden [33].
3.3.2 Electrical System

The electrical system contains models for the alternator, the battery and the various electrical loads (resistive and inductive). Figure 18 displays the overall inputs and outputs of the electrical system model. In particular, the alternator torque output is the most pertinent to the powertrain model as it directly contributes to the engine dynamics equation as a disturbance term.

![Figure 18: Electrical system](image-url)
**Alternator**

A static alternator model relating output current, power and magnetic torque to field current, engine speed and battery voltage has been developed and coupled to the other two models that comprise the electrical system.

![Diagram of the alternator model](image1)

**Figure 19: Schematic of the alternator model [34]**

**Battery**

A Thevenin equivalent circuit model (see Figure 20) was adopted to model the lead-acid battery contained in the vehicle [35].

![Diagram of the battery model](image2)

**Figure 20: Scheme of the battery model using the equivalent circuit analogy**
Equations defining battery voltage, state of charge and temperature as functions of the alternator current and electrical load current have been implemented in the electrical system of the VES. Proper characterization of the battery parameters is especially important because the battery represents one of the primary energy sinks for crankshaft energy harvesting.

_Electrical Loads_

The minivan's electrical loads were characterized experimentally. Electrical shunts were installed to measure both the battery and the alternator current. Simple tests were carried out to identify the electrical draw induced by the various loads in the vehicle such as heated seats, headlights and the infotainment system. The summation of the electrical load current is then subtracted from the alternator current before entering as an input to the battery model.

3.3.3 Mechanical System

The mechanical system is divided into two major parts: powertrain and vehicle dynamics. The powertrain contains the traditional driveline components as well as a tractive force model, while the vehicle dynamics section includes the road loads, inertia transfer and longitudinal dynamics equations. Within the powertrain subsystem, torque is propagated from the engine model to the wheels and speed is fed back from the vehicle level, passing from model to model until reaching the engine crank. Figure 21 summarizes the torque, speed and control signal pathways/directions in the mechanical system of the VES.
Manipulation of the mechanical system to characterize and maximize recoverable energy will be the topic of Chapter 4.

*Engine Torque Production*

A complex mean value engine model (MVEM) has been developed and calibrated to predict the torque production at the shaft and the exhaust gas temperature and flow rate of a Chrysler 3.6L Phoenix V6 naturally-aspirated gasoline engine. The model contains eight interconnected subsystems that are arranged in the following order based air flow path:
1. Ambient Conditions
2. Throttle Body
3. Intake Manifold
4. Volumetric Efficiency
5. Fueling Dynamics
6. Engine Torque Production
7. Exhaust Temperature
8. Exhaust Manifold

Detailed mathematical formulations can be found in the master's thesis of Neeraj Agarwal [24]. Figure 22 summarizes the interconnections between the major subsystems of the MVEM.

![Mean value engine model diagram](image)

**Figure 22: Mean value engine model**

Initial calibration of the engine model used the Big Grid steady-state data which were provided by Chrysler as part of the standard engine calibration process. One of the challenges specific to the engine model was using steady state data to calibrate dynamic models. Once a preliminary model was obtained, the engine dyno at OSU was used to both recalibrate the engine model, and, indirectly, confirm the experimental data set initially obtained from Chrysler.
ICE Dynamics and Torque Converter

The torque converter is located, both in a vehicle and in the VES, between the engine flywheel and the transmission input. It consists of three distinct parts: a pump, which is attached to the engine, a turbine, which is attached to the transmission, and a stator that directs fluid flow within the torque converter. An unlocked torque converter operates in one of two modes, both of which are beneficial to vehicle operation. Torque amplification mode occurs at low speed ratios (defined below) which is useful in vehicle drive-away situation. Fluid coupling mode exists at speed ratios close to unity and allows for smooth power transfer with the added benefit of engine/road pulsation filtering [36]. As is the case with the VES, many torque converters poses the ability to "lock up" via an internal clutch that physically connects the pump and turbine. This functionality is ideal from an efficiency standpoint when torque multiplication and damping are not necessary. In the VES, enabled subsystems and switching logic have been used to model the two different states of the torque converter (TC), locked and unlocked. The engine speed is passed between subsystems via integral state ports so that when switching between locked and unlocked, the last engine speed value becomes the initial condition of the newly-active subsystem. In the unlocked state, low-frequency engine crankshaft dynamics have been captured using the following equations:
The industry-standard K-factor ($K$) and torque ratio ($TR$) approach has been utilized to directly model the steady-state outputs of the torque converter, both in normal and overrun operation [36]. This method was chosen due to the availability of data from Chrysler and is based on measureable characteristics of the torque converter. The popular quasi-static approach developed by Kotwicki [37] was considered, but was eventually ruled out due to the complexity of calibrating the coefficients. Instead, two lookup tables with speed ratio ($SR$) as the input have been implemented to calculate the necessary parameters. The definitions for the torque ratio, speed ratio and the K-factor are reported in Equation 3.5.

Plots of the torque ratio and K-factor, each as a function of the speed ratio, are shown in Figure 24. Data has been provided for speed ratios up to 1.5 which accounts for some degree of overrun operations.

\[
\sum T = J \dot{\omega} = T_{\text{engine}} - T_{\text{pump}} - T_{\text{ancillary}} = (J_{\text{engine}} + J_{\text{pump}} + J_{\text{TC cover}}) \dot{\omega}_{\text{engine}} \tag{3.5}
\]
Torque ratio directly relates turbine torque to pump torque while the K-factor, also known as the capacity factor, can be used to determine the pump torque from pump speed. Figure 25 is a simple schematic of the mechanical portion of the unlocked torque converter model that explains how torque ratio and K-factor are used to calculate engine speed and turbine torque.

\[
SR = \frac{\omega}{\omega_p} \quad TR = \frac{T_r}{T_p} = f(SR) \quad K = \frac{\omega_p}{\sqrt{T_p}} = f(SR)
\]  

(3.6)

It is important to note at this juncture the definition of pump torque. Pump torque is the resistance torque applied to the engine crankshaft that is caused by the fluid coupling between the pump and the turbine. Pump torque can be greater, less than or equal to engine torque.
Just as with the transmission mechanical model, the difference between the input and output power is defined as energy lost. This term is assumed to be entirely converter into heat which is rejected to the Automatic Transmission Fluid (ATF). This quantity, required as an input to the transmission thermal model, is calculated as follows:

\[ \dot{Q}_{gen} = T_{in} \omega_{in} - T_{out} \omega_{out} \]  

(3.7)

In principle, during a locked state, engine speed should be equal to turbine speed. In the model implementation, however, a low pass filter has been included to mitigate instantaneous changes in engine speed when switching between the locked and unlocked states. This is also required to have an integral state port available so speed states can be passed between enabled subsystems. The time constant of the first order system (low pass filter) described below was selected to reflect the actual time required to transition from unlocked to locked, and vice versa.

\[ \tau \frac{d \omega_e}{dt} + \omega_e = \omega_i \]  

(3.8)

In the locked state, the turbine torque equals engine torque, and therefore no dissipation is considered. Also, the engine-side inertias are no longer included in the engine dynamics equation; rather they are translated to the vehicle level during lockup where they are included in the vehicle dynamics equations. More details on inertia transfer are available in the vehicle dynamics section.

**Torque Converter Efficiency**

Though torque converter efficiency is not directly used in the VES, it is required for the analysis models discussed in Chapter 4. Opposed to the transmission, TC efficiency
calculation is relatively straightforward and is based purely on an energy balance. That being said, complexity is added during overrun operation and very specific problems have been observed during shifts and lock/unlock events.

\[
\begin{align*}
1. \eta_{TC_{\text{normal}}} &= \frac{P_{\text{turb-side}}}{P_{\text{eng-side}}} = \frac{T_{\text{turb}}\omega_{\text{turb}}}{T_{\text{eng}}\omega_{\text{eng}}} \\
2. \eta_{TC_{\text{overrun}}} &= \frac{P_{\text{eng-side}}}{P_{\text{turb-side}}} = \frac{T_{\text{eng, effective}}\omega_{\text{eng}}}{T_{\text{turb, effective}}\omega_{\text{turb}}} = \frac{(1/TR)T_{\text{turb, effective}}\omega_{\text{eng}}}{T_{\text{turb, effective}}\omega_{\text{turb}}} 
\end{align*}
\]

The "effective" turbine torque in the equation for overrun efficiency is calculated as part of the transmission overrun efficiency model (discussed in the next subsection) and represents a virtual wheel torque translated to the transmission input. The effective engine torque is related to this "virtual" turbine torque through the TC torque ratio. At the vehicle level, the term "overrun" refers to situations when wheel power is greater than engine power \((|P_{\text{whl}}| > |P_{\text{eng}}|)\). These situations are of particular importance to the transmission efficiency calculation, but in the case of the torque converter, overrun is defined differently. TC overrun occurs when the turbine speed is larger than the pump speed which can happen when the vehicle itself is not in overrun. Therefore, the VES TC efficiency model switches between the equations in (3.9) based on a SR condition.

During shifts, IMEP is decreased to roughly model the effect of spark retard. This near instantaneous decrease in IMEP cuts engine torque for a brief moment, but non-instantaneous engine dynamics keep the engine speed higher than the turbine speed for the duration of the shift. Because of the SR condition on efficiency calculation, the VES assumes normal operation and TC efficiencies greater than 1 occur due to the low engine torque in the denominator of (3.9). As a fix, a constant efficiency is imposed during shifts.
Another problem takes place when transitioning from locked to unlocked. The two enabled subsystems in the TC model contain integrators that can pass engine speed states between one another. This allows for smooth transitions between lockup states. As of now, a similar setup does not exist for turbine torque which results in instantaneous torque jumps. In the future, first order systems similar to the one designed for locked engine speed will be introduced to allow for the matching of turbine torques when switching between enabled subsystems.

**Automatic Transmission Model**

Automatic transmissions are largely used in the automotive industry to transmit power from the engine to the wheels. Automatic transmissions typically contain planetary gear sets, different parts of which are held stationary to yield multiple gear ratios. In addition to allowing the engine and wheels to operate at different speeds, transmissions also decouple the engine from a stopped vehicle by including neutral and park functions. The VES automatic transmission is modeled after the Chrysler 62TE six-speed transmission, found in the 2011 Town & Country Minivan. A simple, quasi-static model has been implemented,
where the input speed is related to the output speed through a selectable gear ratio. One limitation of “discrete” gear shifting models when implemented in low-frequency vehicle longitudinal dynamics models is that they may lead to discontinuities during shift events. More detailed transmission models also include high frequency dynamics that ultimately contribute in determining the correct dynamic response [38]. For simplicity reasons, a “zero-order” (no dynamics) model was chosen to be implemented in the VES, however a simple way to compensate for discontinuities was devised by Chrysler and OSU. Logic has been included to linearly interpolate (blend) between gear ratios during a shift only to capture the fact that shifts are not instantaneous events. The output torque is found by using the conservation of energy equation, where an efficiency term is defined as a function of input speed, input torque and transmission fluid temperature (which affects the viscosity of the fluid). The parameters in the efficiency equation are determined empirically from a curve fitting analysis, and are different for each gear selection. In (3.10), input speed and torque are provided by the torque converter turbine model, as normally occurs during normal vehicle operations (traction phase) [40].
1. $N_{in} = G N_{out}$
2. $N_{out} T_{out} = \eta N_{in} T_{in}$
3. $\eta = C_1 - \frac{C_2 + C_3 \nu + C_4 N_{in}}{T_{in}}$  \hspace{1cm} (3.10)

where

1. $\nu = \frac{\mu}{\rho}$
2. $\mu = \exp(a_1 + \frac{b_1}{T} + c_1 T + d_1 T^2)$ \hspace{1cm} (3.11)
3. $\rho = \frac{1}{a_2 + b_2 T + c_2 T^2}$

The equations for ATF viscosity and density are pure empirical relations developed by Dr. Scott [40]. While experimental data for transmission efficiency were only available at two temperatures, which is relatively standard in industry, the relationship to temperature is not linear, as shown in [40]. It was therefore necessary to determine the nonlinear equation linking efficiency to temperature which was required to couple the mechanical and thermal states of the transmission. Data provided at $44^0\text{C}$ and $93^0\text{C}$ were used in a curve-fitting spreadsheet to find the four coefficients in the transmission efficiency equation for each gear. Figure 27 is a sample plot of the fit for a range of torques at 2000 rpm in fourth gear.
Saturations were required to control extrapolation issues at low speeds and torques. Figure 28 displays two efficiency surfaces (2\textsuperscript{nd} and 5\textsuperscript{th} gears) to show dependency on speed and torque. As could be inferred from inspection of (3.10), efficiency drops as turbine torque decreases. For gears 1-3, the coefficient C4 is negative implying that greater efficiency is achievable at higher speeds which is precisely the case for the gear 2 surface below. For gears 4-6, C4 is positive but very small in magnitude which results in the relatively flat dependence on speed depicted by the gear 4 surface.

Figure 27: Calibration of transmission efficiency
The heat generation caused by friction in the transmission internal components is modeled simply as the difference between the input and output power:

\[ \dot{Q}_{\text{gen}} = T_n \omega_m (1 - \eta) \]  

(3.12)

Similar to the torque converter, the heat generation from the transmission internals is an input to the transmission thermal model.

**Overrun Efficiency Calculation**

The equation for efficiency in (3.10) was specifically designed to calculate transmission efficiency during traction operations, for which the transmission input torque and speed
are defined by the torque converter turbine. On the other hand, the method described above cannot be directly applied during overrun operations (e.g., when the vehicle is coasting or braking) because the transmission "inputs" switch when the tractive force is coming from the vehicle wheels, directed to the engine. In overrun conditions, both speed and torque signals must be switched, as shown in Figure 29.

![Normal Operation and Overrun Operation](image)

**Figure 29: Torque and speed propagation during normal and overrun modes**

Overrun conditions are typically occurring when the aggregate torque at the wheel is negative, taking into account the transmission output torque, the braking torque and the road load torque.

\[
T_{\text{whl}} = \pm T_{\text{trans}} - T_{\text{Brakes}} - T_{\text{RL}}  \quad (3.13)
\]

As an additional condition, engine torque must be negative to ensure that the driver is not commanding any positive torque. Three methods were devised to calculate transmission efficiency during overrun, all three of which required two assumptions. An assumption common to all methods is that the power loss through the transmission is the same regardless of power flow direction.
Method 1: Engine-Side Inputs

The primary assumption made in method 1 is that calculation of the transmission efficiency should respect the original intent of (3.10) and use engine-side speed and torque. In normal operation, engine-side conditions are those of the turbine, but in overrun, turbine torque, as calculated by the VES, is directly related to engine torque which is meaningless given the reversed torque flow. It was therefore required to calculate an effective turbine-side torque that would allow for calculation of the efficiency. Using (3.10) and (3.14), a single equation with one unknown (turbine torque) can be derived. Here the subscripts $t$ and $w$ represent the effective turbine and wheel respectively. Notice that the wheel power is now considered the input.

$$\omega_t T_t = \eta \omega_w T_w$$ \hspace{1cm} (3.14)

Combining (3.10) and (3.14) and rearranging yields

$$\omega_t T_t = \omega_w T_w \left( C_1 - \frac{C_2 + C_3 \nu + C_4 N_t}{T_t} \right)$$

$$\omega_t T_t = \omega_w T_w C_1 - \omega_w T_w \left( \frac{C_2 + C_3 \nu + C_4 N_t}{T_t} \right)$$ \hspace{1cm} (3.15)

By moving all terms to one side, the quadratic equation can be solved for the effective turbine torque. This can then be plugged back into (3.14) to determine efficiency.
\[ \omega_i T_i^2 - \omega_w T_w C_1 T_i + \omega_w T_w \left( C_2 + C_3 \nu + C_4 N_i \right) = 0 \]

\[ T_i = \frac{-b \pm \sqrt{b^2 - 4ac}}{2a} \quad (3.16) \]

\[ \eta = \frac{\omega_w T_w}{\omega_i T_i} \]

The equations in (3.16) were implemented in Simulink to dynamically calculate efficiency during overrun. Only the ",-b+..." equation is utilized because, given the sign convention used, negative turbine torques are not possible during overrun. Although this method outputs reasonable efficiency values and has been left in the VES as an optional subsystem, Methods 2 and 3 were selected as being more formally correct.

**Method 2: Wheel-Side Inputs**

Method 2 is based on the consideration that the transmission efficiency should always be calculated based on the *input* transmission conditions, regardless of the vehicle operating mode. In overrun conditions, this means that the transmission efficiency should be determined based on the torque and speed from the vehicle side (wheels):

\[ \eta_{\text{overrun}} = C_1 - \frac{C_2 + C_3 \nu + C_4 N_{\text{wheel}}}{T_{\text{wheel}}} \quad (3.17) \]

**Method 3: Wheel-side inputs translated to the engine-side**

Method 3 takes into consideration the fact that the normal transmission efficiency equation was developed for a specific range of turbine torques and speeds. The wheel-side of the transmission operates at higher torques and lower speeds compared to the turbine-side, and therefore using the wheel-side conditions directly to find efficiency
(Method 2), often results in non-realistic values because the inputs are outside the calibrated range. For this reason, wheel-side inputs are translated to the turbine-side by using the gear ratio and final drive ratio. The overrun efficiency can then be calculated using reasonable inputs based upon wheel-side conditions. Method 3 is currently active in the VES. As one may gather, calculation of overrun efficiency has been a particular challenge given the normal flow of torque in the VES.

\[
\eta_{\text{overrun}} = C_i - \frac{C_2 + C_3 \nu + C_4 (GR) N_{\text{wheel}}}{(1/GR) T_{\text{wheel}}}
\] (3.18)

Regardless of the method chosen to calculate transmission overrun efficiency, Equation (3.19) is used to find the effective virtual turbine torque required for the TC efficiency model. The TC efficiency is then used in conjunction with transmission efficiency to determine an overall powertrain efficiency that is used in the torque transfer subsystem of the disturbance torque analysis model (see Chapter 4).

\[
T_{\text{turb,eff}} = \frac{T_{\text{whl}} \omega_{\text{whl}} \eta_{\text{overrun}}}{\omega_{\text{turb}}}
\] (3.19)

For illustration purposes, Figure 30 shows a plot of the transmission efficiency during the first 80 seconds of an FTP cycle. In the VES, the transmission efficiency is calculated using subsystems that are enabled/disabled based on wheel torque criteria. A downstream switch selects between the two and imposes zero efficiency during zero cycle velocity.
As depicted in Figure 30, transmission efficiency tends to rapidly decrease anytime the vehicle transitions from normal to overrun operations. Although partially due to gear selection and lockup status, the primary reason for the low efficiency dips is the changeover from positive to negative power flow through the transmission. With reference to equation (3.10), efficiency drops when input torque decreases. As a vehicle in normal operation follows a drive cycle, the torque required will decrease (due to the driver reducing the accelerator pedal position) as negative power becomes necessary. This decrease in torque required translates directly to lower turbine torque, and consequently to lower transmission efficiency as a negative power event approaches. Once the wheel-level torque becomes negative, the transmission inputs switch and the
efficiency calculation commences with small negative torques that steadily increase in magnitude as the negative power event continues. This cycle repeats itself, in reverse, when positive power is requested by the driver.

**Tractive Force Model**

The tractive force model converts the transmission output torque after the final drive into a force at the wheel. It is also where the brake command $\beta$ is taken into account. The force at the front wheels is given by:

$$F_{\text{front}} = \frac{T - \beta m_{\text{veh}} g \mu r_{\text{wheel}} (P_{\text{front}})}{r_{\text{wheel}}}$$

(3.20)

and the force at the rear wheel is:

$$F_{\text{rear}} = \frac{\beta m_{\text{veh}} g \mu r_{\text{wheel}} (P_{\text{rear}})}{r_{\text{wheel}}}$$

(3.21)

The variable $P$ is the brake proportioning between the front and rear axles. The total motive force available is then:

$$F_{\text{total}} = F_{\text{front}} + F_{\text{rear}}$$

(3.22)

which is passed on to the vehicle dynamic model.
Vehicle Longitudinal Dynamics Model

The VES vehicle model is strictly a 1-D model that captures the longitudinal dynamics of the vehicle. The rotational inertias of the driveline components (transmission gears, torque converter, etc.) seen from the transmission input have been translated to the vehicle level through effective speed ratios and converted into equivalent masses [31]:

\[
m_{\text{eff}} = \frac{J_{\text{driveline}} G_{\text{trans}} G_{\text{finaldrive}}}{r_{\text{wheel}}^2}
\]

These equivalent masses are then added to the overall vehicle mass to affect acceleration performance. During a shift, the transmission inertias corresponding to adjacent gears are linearly-blended for a predetermined shift time. The engine-side inertias (engine, TC pump, TC cover) are added only during lockup. Otherwise, they are included in the engine dynamics equation in the torque converter model. At the vehicle level, the wheel inertias are also included.

Figure 32: Vehicle Model I/O
The three standard road loads taken into account are as follows:

**Aerodynamic drag**

\[ F_{aero} = 0.5 \rho_{air} C_d A_{frontal} V^2 \]  

(3.24)

where \( C_d \) is the coefficient of aerodynamic drag and \( A_{frontal} \) is the frontal area of the vehicle. The information on vehicle drag were provided by Chrysler, as shown in Table 10. The density of air is assumed constant and calculated using the ideal gas law:

\[ P_V = P \frac{1}{\rho} = RT \]  

(3.25)

\[ \rho_{air} = \frac{P_{amb}}{R_{air} T_{amb}} \]

**Tire rolling resistance**

\[ F_{roll} = a_0 + a_1 V + a_2 V^2 \]  

(3.26)

At slow speeds, the rolling resistance force is calculated along a line that intersects the above equation and passes through zero. This was done to avoid a positive rolling resistance road load at zero vehicle velocity. The constant term in the rolling resistance function exist to capture the fact that friction is present the instant an object attempts to move. The fact that no friction exists until that moment makes friction modeling near zero velocity difficult.

**Grade**

\[ F_{grade} = mg \sin(\theta) \]  

(3.27)
It should be noted the model was developed using the aforementioned road loads but was fundamentally altered once the validation effort began. Details of this process can be found in the following road load calibration section. Vehicle velocity is calculated as:

\[ F_{\text{roadload}} = F_{\text{aero}} + F_{\text{roll}} + F_{\text{grade}} \]

\[ V = \int \frac{F_{\text{tractive}} - F_{\text{roadload}}}{m_{\text{vehicle}} + m_{\text{effective}}} \, dt \]  \hspace{1cm} (3.28)

Through the bus system, vehicle velocity is then fed back to the driver model where corrections to throttle and braking pedal positions are made.

**Road Load Calibration**

Validation of the vehicle dynamics model primarily consisted of confirming the road load equations. This proved to be more difficult than anticipated. A chronological list of the methods attempted will be included here as a guide to those endeavoring to derive a given vehicle's road load characteristics. The overarching goal of each method was to either obtain an equation relating road load force to velocity that could be implemented directly in the VES, or to validate the existing individual equations.

**Method 1: Experimental Coastdowns**

The first method attempt involved fitting curves to experimental coastdown data so that the resulting analytical relationship between velocity and time could be used to derive force as a function of velocity. The vehicle was taken to a flat, straight road during a calm afternoon and two coastdowns were performed in neutral from roughly 65 mph - one going either direction. In such a test, the only forces acting on the vehicle are the road load forces because the powertrain is completely disengaged from the wheels. In
post processing, the two data sets were synced such that the initial velocity was 60 mph for both tests. The signals logged from the ECU were of a sample-and-hold nature so logic was written to select the first point of each "hold" for use in curve fitting routines. Figure 33 shows the experimental data for the first coast down test.

![Figure 33: Vehicle coast down test #1](image)

Initially, the coastdown trajectories were assumed to be quadratic in nature, so the data was "cut" at some threshold velocity, below which decidedly linear behavior was observed. The two traces were then averaged to reduce experimental errors, and a second order polynomial was fit to the result with the constant coefficient forced to 60 mph using MATLAB's "cftool." Once the vehicle's experimental coastdown characteristics were
documented, a "coastdown version" of the VES was created to compare the simulator, which contained the three individual road load equations, to the experimental data. This altered VES was essentially the standalone vehicle model with the acceleration integrator set to an initial condition of 26.82 m/s (60 mph). The results of both experimental coastdown tests as well as the simulated test using the original Chrysler road load information are displayed in Figure 34.

![Neutral Coast-Down Testing](image)

**Figure 34: Experimental and simulated coast down tests**

At first glance, the simulated coastdown seems to agree quite well with the averaged experimental trace, but further manipulation revealed a fundamental flaw in the general fitting approach. The traces in Figure 34 represent vehicle velocity as a function of time
and can therefore be used to find instantaneous acceleration, and consequently force. The analytical derivative was taken of both the averaged experimental and simulated quadratic fits to obtain acceleration, which was then multiplied by vehicle mass to yield road load force. When the road load force and power are plotted as functions of the vehicle velocity, the following relationships are revealed.

![Figure 35: Road load force as a function of velocity (Method 1)](image-url)
A quadratic fit was then applied to the experimental force vs. velocity trace and the resulting coefficients were converted to HP@50 for use in the chassis dyno. A quadratic fit was specifically chosen to model the physical relationship between velocity and force because total road load is the summation of two second order, and one first order polynomials in velocity. Not until calibration of the chassis dyno was well underway was the fundamental flaw in Method 1 discovered. In the very early stages of the study, the velocity vs. time coastdowns were assumed to be quadratic.

\[ V = at^2 + bt + c \]

\[ \Rightarrow F \propto \frac{dV}{dt} = 2at + b \]  

(3.29)
With this assumption:

\[ at^2 + bt + c - V = 0 \]

\[ t = \frac{-b \pm \sqrt{b^2 + 4a(c-V)}}{2a} \]  \hspace{1cm} (3.30)

Plugging the equation for time into the second part of (3.29) yields:

\[ F \propto \frac{dV}{dt} = 2\alpha \left[ \frac{-\lambda \pm \sqrt{b^2 + 4a(c-V)}}{2\alpha} \right] + \lambda \]  \hspace{1cm} (3.31)

\[ F \propto \sqrt{b^2 + 4a(c-V)} \propto \sqrt{V} \]

The initial assumption therefore results in a road load force that is proportional to the square root of velocity which is precisely the trend witnessed in Figure 35. In reality, the total road load force should be proportional to the square of velocity meaning that the results of Method 1 are invalid.

**Method 2: Updated Fits**

Method 2 followed the same procedural logic as Method 1, but began with a curve fitting procedure for the experimental vehicle velocity data \( v(t) \) that is more based on first principles. At each point in time during the coastdown test, the following relations hold:

\[ F = m \frac{dV}{dt} \text{ and } F = CV^2 + BV + A \]  \hspace{1cm} (3.32)

By observing that the second equation results in a negative force for positive velocities (summation of all road load equations), a nonlinear ordinary differential equation with velocity as the dependent variable can be derived.
\[ m \frac{dV}{dt} + CV^2 + BV = -A \]  

(3.33)

One possible analytical solution of (3.33), valid in the interval \( t \in [0,200] \), can be found as follows:

\[
V(t) = \frac{\sqrt{4AC - B^2} \tan \left( \frac{1}{2} \left( c_1 \sqrt{4AC - B^2} - \frac{t\sqrt{4AC - B^2}}{m} \right) \right) - B}{2C}
\]  

(3.34)

A verification of the solution to (3.13) can be found in Appendix A. In (3.34), the constant \( c_1 \) is determined based on the initial vehicle velocity. The above form was then used to curve fit the vehicle velocity profile during the coastdown test, to determine the unknown parameters. Starting values for \( A, B \) and \( C \) were back-calculated from HP@50 coefficients stored in the OSU chassis dyno computer database for a vehicle similar to the Chrysler Minivan. These values served as a reference point around which the curve fitting optimization routine could be initialized. The result of the fit represents a considerable improvement from Method 1, and is shown plotted with the experimental coast down data in Figure 37.
As in Method 1, the analytical derivative of the $v(t)$ fit was taken, yielding the vehicle acceleration. The acceleration trace was then multiplied by the vehicle mass to acquire $F(t)$, which when plotted against $v(t)$ gives the trace shown in Figure 38. It can be observed that the relationship between road load force and velocity is quadratic as it should be. A second order polynomial was then fit to the data and the coefficients were converted to HP@50 for use in the dyno.
Method 3: Dynamometer Method

When performing a coast down test on the dyno to calibrate coefficients (see dyno calibration section), the dyno computer determines the $F(v)$ traces using a "brute force" approach that does not involve fitting a curve to $v(t)$ data. To serve as a sanity check, this method was applied to the experimental data. In this case, acceleration and force are calculated purely based on the change in velocity over some time period. $F(t)$ is plotted against the average velocity for each time window in Figure 39.

$$V_{i,avg} = \frac{V_{i+1} + V_i}{2} \quad \text{and} \quad a_i = \frac{V_{i+1} - V_i}{t_{i+1} - t_i}$$ (3.35)
As the various methods of determining the road load equation were attempted, it became clear that the dyno should not be made to match the model, but that the dyno should match experimental data and the model should follow. To this end, the VES road load model became simply a second order equation that calculates the resistant force at the wheel as a function of the vehicle velocity that could be easily interchanged depending on the chosen method.
Method 4: EPA Coefficients

Near the end of the work involving Method 3, it became known to the team that the EPA has a compiled list of pre-dyno road load coefficients for a number of makes and models. Using the available data, the \( C, B \) and \( A \) coefficients for a 2011 Town & Country were included and instantly became the benchmark used to compare Methods 1-3. As stated in the section describing Method 3, the VES was fitted with various selectable road load equations and the one involving the EPA coefficients was set as default. Figure 40 compares the force vs. velocity relations gleaned from the three successful methods explored. Below the figure is a table of coefficients and their corresponding HP@50 values. The specific sources of the discrepancy between the experimental curves and the EPA curve are unknown, but more than likely originate from the test procedure itself. The OSU tests were performed on the open road and the speed data was acquired from the vehicle ECU. The EPA, on the other hand, carries out its tests with more accurate speed characterization on a controlled track.
Figure 40: Comparison of road load relationships (Methods 2, 3, 4)

Table 12: Summary of road load coefficients for Methods 2, 3 and 4

<table>
<thead>
<tr>
<th></th>
<th>Method 2</th>
<th>Method 3</th>
<th>Method 4</th>
</tr>
</thead>
<tbody>
<tr>
<td>C [lbf/mph²]</td>
<td>0.0160</td>
<td>0.0088</td>
<td>0.0253</td>
</tr>
<tr>
<td>B [lbf/mph]</td>
<td>1.1412</td>
<td>1.4957</td>
<td>0.1950</td>
</tr>
<tr>
<td>A [lbf]</td>
<td>43.1528</td>
<td>40.1449</td>
<td>46.0300</td>
</tr>
<tr>
<td>C [HP@50]</td>
<td>5.333</td>
<td>2.933</td>
<td>8.426</td>
</tr>
<tr>
<td>B [HP@50]</td>
<td>7.608</td>
<td>9.971</td>
<td>1.300</td>
</tr>
<tr>
<td>A [HP@50]</td>
<td>5.754</td>
<td>5.353</td>
<td>6.137</td>
</tr>
<tr>
<td>Total HP@50</td>
<td>18.695</td>
<td>18.257</td>
<td>15.860</td>
</tr>
</tbody>
</table>

Note: The highlighted values were used in the first step of dynamometer calibration
3.3.4 Thermal System

The VES thermal system is the most complex of the four major systems. In addition to supporting the mechanical system during pseudo-steady-state temperature operation, the thermal system was specifically designed to capture the warm-up phase. This phase is important because the efficiency of the transmission is especially low during this period and any improvements directly affect vehicle fuel consumption. Development of a thermal management strategy specifically designed for rapid warm-up of vehicle working fluids represents the thermal counterpart to the work presented in this document [24].

Coolant-Oil Loop

The coolant-oil loop model is a high level structure that contains a large number of submodels that are interconnected based on exhaust, coolant and oil flows. Figure 41 depicts the coolant flow path and most of the models involved.
Figure 41: Flow diagram of the engine coolant path (red=exhaust gases, blue=coolant)

More details and a complete list of submodels can be found in [24].

*Transmission and Torque Converter Thermal Model*

The thermal model of the transmission and torque converter is based on the *lumped thermal mass* approach [46], where in particular the transmission internals, the torque converter and the transmission sump/case are considered as the three masses of the thermal system. The inputs and outputs of the transmission thermal model are detailed in Figure 42 and a diagram of the transmission oil flow path is shown in Figure 43.
Figure 42: Input/output representation of the transmission thermal model

Figure 43: Block diagram of the transmission oil flow path
The governing equation that describes the temperature dynamics of the transmission internals (gears, clutches, bearings, etc.) is given by:

\[
C_{ti} \frac{dT_{ti}}{dt} = \dot{Q}_{gears} + c_p \dot{m}_{ti} (T_{TO,\text{return}} - T_{ti}) \tag{3.36}
\]

where \(C_{ti}\) is the heat capacity of the transmission internals (calibration parameter, kJ/K), \(\dot{Q}_{gears}\) is the rate of heat generation (kW) produced by the friction losses in the transmission gears (based on Equation (3.12)), \(\dot{m}_{ti}\) is the mass flow rate of transmission fluid entering the system control volume, and \(T_{TO,\text{return}}\) is the inlet ATF temperature.

For the torque converter, a similar equation can be defined:

\[
C_{TC} \frac{dT_{TC}}{dt} = \dot{Q}_{TC} + c_p \dot{m}_{TC} (T_{TO,\text{return}} - T_{TC}) \tag{3.37}
\]

where in this case \(\dot{Q}_{TC}\) is determined by the torque converter model (Equation (3.7)).

Finally, the thermal dynamics of the transmission sump/case is given by:

\[
C_{sc} \frac{dT_{sc}}{dt} = c_p \dot{m}_{tc} (T_{TC} - T_{sc}) + c_p \dot{m}_{ti} (T_{ti} - T_{sc}) - UA_{amb} (T_{sc} - T_{amb}) \tag{3.38}
\]

where \(UA\) is the product between the overall heat transfer coefficient and the area of the sump, and accounts for the rate of heat rejection of the sump/case to ambient.

The three aforementioned models are implemented in the VES so that the temperatures of the transmission internals, torque converter and sump/case are the outputs. The specific heat capacity \(c_p\) of the transmission fluid is calculated dynamically as a function of temperature:

\[
c_p = c_0 + c_1 T \tag{3.39}
\]
where the constants have been determined experimentally [40]. The overall mass flow rate of transmission fluid is found using two 2D lookup tables. The first determines line pressure as a function of gear and turbine torque, while the second determines volumetric flow rate from line pressure and engine speed. The individual mass flow rates, \( \dot{m}_{TC} \) and \( \dot{m}_T \), sum to equal the total flow rate to/from the transmission oil heater/cooler and are related through the calibration parameter \( \alpha \) such that \( \dot{m}_T = (1 - \alpha)\dot{m}_{total} \) and \( \dot{m}_{TC} = \alpha \dot{m}_{total} \).

The calibration parameter \( \alpha \) was nominally set at 1/8, as suggested by Chrysler.

The primary input to Equations (3.36) and (3.37) is the heat generated in the transmission and torque converter respectively. This warms up the ATF as it flows through the transmission into the sump.

![Figure 44: Heat generation in the transmission and torque converter](image-url)
Comparison of simulated transmission oil sump temperature with experimental warm-up data serves as a partial validation until more data can be gathered. Figure 45 shows the simulated and experimental cold-start warm-up trajectories of the ATF.

![Figure 45: ATF warm-up validation [24]](image)

The OSU test vehicle was fitted with thermocouples to measure the temperature of the oil leaving and returning to the transmission. Along with the sump temperature logged by the vehicle ECU, a full calibration and validation will be performed. The "C" values in Equations (3.36) through (3.38) are well established which renders the flow split parameter $\alpha$ the primary calibration term. The internal torque converter and transmission temperatures cannot be easily measured, so $\alpha$ will be manipulated until the sump temperature can be calculated accurately both during warm-up and in steady-state.
Catalytic Converter

If an SI-engine operates primarily at stoichiometric air/fuel ratios, a three-way catalyst (TWC) can be used to control harmful emissions through the oxidation of NO, and the reduction of CO and hydro carbons [6]. TWCs are located directly downstream from the exhaust manifold and are located as close to the engine as possible to maximize pollutant conversion efficiency, which is a strong function of temperature. Since the chemical reactions that take place in a TWC are exothermic in nature, and since the catalyst represents a large thermal sink, an accurate thermal model is required. The temperature of the exhaust leaving the catalytic converter is especially important if exhaust gas heat recovery systems are being used. Similar to the transmission thermal model, a lumped mass model with three sections is used to model the axial temperature distribution within the catalyst. The first and third sections represent the bricks in a three way catalyst (TWC) and the second section represents an air gap between those bricks.

Heat generation from the exothermic catalytic reactions has been included. The first and third sections contain three distinct mass lumps: exhaust gas (g), substrate (s), and can (c) while the second only contains gas and can. Figure 47 displays the energy flows within the catalyst for which the governing equations were derived.

Figure 46: Catalytic Converter I/O
The total heat transfer equation for the exhaust gas control volumes is

\[
\frac{dU_g}{dt} = m_g c_{p,g} \frac{dT_g}{dt} = (\hat{H}_{g,\text{in}} - \hat{H}_{g,\text{out}}) - \dot{Q}_{\text{conv,g,s}} + (1 - \alpha)\dot{Q}_{\text{gen,chem}} \tag{3.40}
\]

where the enthalpies are defined as

\[
\hat{H}_{g,\text{in}} = \dot{m}_g c_{p,g} T_{g,\text{in}} \quad \text{and} \quad \hat{H}_{g,\text{out}} = \dot{m}_g c_{p,g} T_{g,\text{out}} \tag{3.41}
\]

and the convective heat transfer within the elements is

\[
\dot{Q}_{\text{conv,g,s}} = h_{gs} A_{gs} (T_g - T_s) \tag{3.42}
\]
The parameter $\alpha$ is simply a way of dividing the exothermic heat release between the substrate and the gas masses. The total heat transfer equation for the substrate control volumes is then

$$\frac{dU_s}{dt} = \dot{Q}_{\text{conv},s} - \dot{Q}_{\text{cond},s} + \alpha \dot{Q}_{\text{gen},\text{chem}} \quad (3.43)$$

where

$$\dot{Q}_{\text{cond},s} = \frac{1}{\frac{2\pi kL}{\ln \frac{r_2}{r_1}}} \left( T_s - T_c \right) = K_{sc} (T_s - T_c) \quad (3.45)$$

The total heat transfer equation for the can control volumes is

$$\frac{dU_c}{dt} = \dot{Q}_{\text{cond},c} + \dot{Q}_{\text{cond},c,up} - \dot{Q}_{\text{cond},c,down} - \dot{Q}_{c,\text{amb}} \quad (3.46)$$

where

$$\frac{dU_c}{dt} = m_{c,p,c} \frac{dT_c}{dt} \quad (3.47)$$

and

$$\dot{Q}_{\text{cond},c,up} = \frac{A_k}{L_c} (T_{c,in} - T_c) \quad (3.48)$$

$$\dot{Q}_{\text{cond},c,down} = \frac{A_k}{L_c} (T_c - T_{c,out}) \quad (3.49)$$

$$\dot{Q}_{c,\text{amb}} = h_{c,\text{amb}} A_{c,\text{amb}} (T_c - T_{\text{amb}}) \quad (3.50)$$
Much of the catalytic converter model was developed as part of [41]. The fundamental ideas and governing equations have remained mostly unchanged, but certain alterations have spawned a very different catalyst model. The major changes are as follows.

1. Reduction from 6 to 3 lumps, one of which is modeled as an air gap rather than a substrate section.

2. Removal of upstream electric heater model

3. Conversion from static map inputs to dynamic inputs routed from the rest of the VES

4. Addition of exothermic heat release

**Exothermic Heat Release Submodel**

Pollutants (THC, CO, NOx) in the exhaust gas stream are removed in the catalytic converter by means of an exothermic chemical reaction. This heat production was included in the catalyst model because it directly affects the exiting gas temperature. Three dimensional maps are used to determine the conversion efficiency in each brick of the catalyst. This conversion efficiency is then multiplied by the mass flow rates of the respective pollutants and then by the enthalpies of formation pertaining to the reactions [42]. The heat generated in each lump is then added as an input to the gas and substrate lumps in each section. For one pollutant in one section, the heat generation is determined by

\[
\dot{Q}_{\text{exotherm}} = \eta_{\text{conversion}} \times \dot{m}_{\text{pollutant}} \times \Delta_f H^o
\]

(3.51)

where \( \eta \) is a function of temperature, mass flow rate and air fuel equivalency ratio. In addition, \( \dot{m} \) is a function of engine torque and engine speed while \( \lambda \) is scheduled on
engine speed and throttle position. Due to the finite surface area of each section, and to the finite rate of each exothermic reaction, a maximum conversion of each species has been included in the model to capture situations of extremely high exhaust mass flow. To develop the three-dimensional efficiency maps, a coated catalyst was fitted with thermocouples and emissions analyzer lines. An image of the instrumented catalyst is included as Figure 48.

Validation of the catalyst mode has not been performed yet but two testing methodologies have been devised to provide the data necessary for the exothermic heat release submodel. First, a somewhat expansive DOE could be carried out to populate the conversion efficiency maps described earlier. Though time intensive, the resulting maps would give some insight into the way the heat was being generated which could prove
useful. The second approach essentially assumes the catalyst to be a "black box" and tests would be run to characterize the outlet versus inlet temperature for a variety of conditions. Either testing method will provide enough temperature data to calibrate the entire catalyst model, not just the exothermic heat release submodel.

3.3.5 Models for the Chrysler Control Algorithms

The Chrysler control subsystem of the VES contains the basic control algorithms that are necessary to operate the engine and transmission systems during steady-state and transient conditions. In particular, the six outputs of the Chrysler control block are:

1. Gear command
2. Lockup command
3. Desired engine idle speed
4. Alternator field current
5. Torque reduction command
6. Fuel Shutoff command

The desired idle speed is simply a constant and generally does not change throughout a drive cycle. It was initially thought to be a function of coolant and/or engine oil temperature but further analysis revealed that transient ancillary loads (i.e. starter) on the engine are what actually dictate non steady state idle speeds. The torque reduction command becomes active for a specified duration during gear shifts. A gain is directly applied to the engine IMEP in the torque production model to mimic a spark retard that is imposed to the engine during gear shift events. This is a gross simplification but an average gain was found to suffice for this application.
**Transmission Shift Model**

The current shift model selects transmission gear based on vehicle speed and throttle position. The throttle position $\alpha$ is passed to a set of maps that output threshold velocity, one for every possible up and down shift. The two threshold velocities for each gear are multiplexed together and a multi-port switch selects the threshold velocity set corresponding to the current gear. An “if” block compares the vehicle velocity to the threshold velocities for an up or down shift, and then uses action subsystems to add 1, 0 or -1 to the gear counter accordingly. In addition to the threshold velocities, some conditions were added to ensure that minimum and maximum engine speeds were being considered, as well as deceleration rate. Figure 50 summarizes the logic decisions.

![Figure 50: Shift logic](image-url)
The updated gear selection is fed back to the multi-port switch and a new threshold velocity set is passed to the “if” block. An example of two shift lines for a given gear is shown in Figure 51.

![Map of gear shifting schedule](image)

**Figure 51: Map of gear shifting schedule**

As described later in the driver model section, the shift model has an inherent flaw that is unavoidable for the time being. Shift conditions are meant to be a function of pedal position, but the VES driver model can currently output only throttle position. Since throttle position is a significantly more active signal as compared to pedal, the shift model was observed to be shifting far too often, especially during decelerations. Two courses of action were taken to mitigate this problem. First, a minimum deceleration condition was added to the down/up/stay logic to ensure that down shifts would not occur...
until a certain deceleration rate was observed. Second, and most importantly, the shift maps themselves were translated and stretched based on experimental data. Vehicle velocity and throttle position were categorized based on the current gear for each data point, and then the unaltered shift lines were superimposed on top of the scattered points so they could be adjusted to fit the boundaries. Though the shift model is still not perfect, simulated gear position does agree quite well with experimental data, especially in the lower gears. Figure 52 serves as validation of the shift model.

![Figure 52: Shift model validation](image-url)
**Torque Converter Lockup Model**

The model of the torque converter lockup control logic is very similar to the gear shift control model in its structural implementation.

Depending on the gear, an unlock speed and a lock speed based on the throttle position are continuously fed to a comparison block. The current speed (in this case transmission output speed) is compared to the unlock and lock thresholds and the appropriate action is taken. If the speed is between the two critical values, the lockup command holds at its current position. When the shift model changes gear, the set of threshold speeds are instantaneously updated. Logic is also included to unlock the TC during shifts for a specified amount of time. Figure 54 displays example lockup lines for a given gear.
Figure 54: Map of torque converter lockup schedule
Deceleration Fuel Shutoff (DFSO)

Deceleration fuel shutoff is used to save fuel without negatively affecting the vehicle drivability. This generally occurs when a driver is either coasting or braking lightly. Since drivability is inherently subjective, the rules developed by Chrysler are heuristic in nature and have been implemented as such in the VES. When all of the requirements are met, a positive fuel cut command is sent to the remaining segments of the DFSO control model where counters are used to determine what phase of fuel cut the DFSO event is in. The phase number is passed through the bus structure to separate fuel flow and IMEP submodels that calculate the respective trajectories based upon both predetermined and dynamically calculated slopes. The general methodology of the DFSO implementation is shown graphically in Figure 56.

Figure 55: DFSO model I/O
Details of the three primary DFSO-related models are included below.

**DFSO Control Model**

The DFSO Control Model contains two distinct sections: conditional logic and counters & resets. The **conditional logic** section determines, based on specified conditions, when it is permissible to enter the DFSO mode. This segment outputs a Boolean signal with 0 corresponding to a "DFSO not allowed" case. While refraining from using specific numbers, the following six conditions must be met in order to enable the DFSO mode.

1. The transmission input speed must be greater than a minimum value (rpm). Once in fuel cut mode, break when a different minimum speed is reached.
2. The coolant temperature must be greater than a minimum threshold (°C).
3. The transmission oil temperature (approximated as the sump temperature) must be greater than a minimum threshold (°C).
4. The gear must be higher than a minimum value.
5. If the fuel cut mode is enabled above, for instance, gear N, then break on the N to N-1 shift and do not reenter fuel shutoff until gear N-1 is abandoned.

6. Most importantly, the engine *throttle* must always be at its minimum value during deceleration, implying that the accelerator pedal must be completely released. Note that engine throttle and accelerator pedal are not necessarily the same thing. In reality, the accelerator pedal position is the ultimate condition required to enter DFSO, which eventually necessitates the addition of a feed-forward component.

When one of the conditions is no longer true, the conditional logic section outputs a 0 and in doing so, begins a time counter that prevents the model from reentering DFSO too soon. The delay time was set to 1 s as per Chrysler's recommendation. Other dependencies (i.e. braking intensity and velocity) were explored but none showed strong correlation to the DFSO activity. A negative acceleration requirement was specifically excluded to ensure the capturing of down-grade situations that may be present in other cycles. The turbine speed requirement inherently includes decelerations of the vehicle and hence a direct braking level condition was not necessary.

To calibrate the conditions for alpha and turbine speed, "fuel shutoff pattern number" was plotted as the dependent variable versus each. This pattern number corresponds to a cylinder deactivation strategy used by Chrysler, the specific implementation of which was out of scope for this project. A simplified method was adopted and is described with the fuel flow submodel. Regardless, a non-zero value of the pattern number variable implies some level of DFSO, which made it ideal for reverse-engineering condition
limits. Figure 57 includes plots of velocity, turbine speed and alpha vs. pattern number for experimental FTP data from Chrysler and OSU tests.

Immediately apparent are the well-bounded ranges for each condition. The velocity plot represents an instance where the variable in question does not appear to have any direct correlation to DFSO; fuel shutoff pattern numbers greater than one are observed for effectively the entire FTP velocity range. For the variables that did show meaningful correlation, the upper and lower limits were extracted and included in the conditional logic. The throttle limits proved to be problematic and constituted the primary

![Figure 57: Calibration of DFSO condition limits for velocity, turbine speed and alpha](image-url)
motivation for a feed forward pedal command. More details on this issue are included later in this chapter.

The second section of the DFSO control model, **counters & resets**, takes the Boolean signal from the conditional logic subsystem and converts it into a specific DFSO fuel cut phase:

0 = no part of DFSO active
1 = down-slope and then hold phase
2 = up-slope and then break phase

Figure 58 is a good graphical example of what occurs during DFSO events. The example taken for this figure is a portion of the FTP driving cycle. The DFSO phase command is superimposed over the altered and unaltered fuel flow rate traces. The blue and red traces are the outputs of simulation; comparisons with real DFSO event data will be included later.
Figure 58: Overview of DFSO phase command and related fuel flow trace, compared to a simulated case in absence of DFSO (data from FTP cycle)

The phase number described above is calculated as the difference between a conditional counter and a reset counter (see Figure 59).

Figure 59: Counters & reset DFSO control submodel
As the input Boolean signal goes from 0 to 1, signifying that DFSO can be entered, the "either edge" counter equals 1 until the input signal reverts back to 0, at which point the counter increases to 2. The conditional counter is effectively reset to 0 (by subtraction of 2) upon the activation of any of three reset signals. The "< 3" conditional block seen in Figure 59 accounts for situations when the conditional logic allows reentry into DFSO before phase 2 has completely finished. In this situation the counters & resets output is set to 1 (down-slope and then hold phase) until DFSO in naturally exited. The overall output of the DFSO control model describes the necessary DFSO phase and is passed to the fuel flow and IMEP calculators.

Fuel Flow Submodel

The VES fuel flow submodel is located in the engine torque production model and calculates fuel flow as a function of DFSO phase. A multiport switch with data ports corresponding to DFSO commands of 0, 1 or 2 selects the appropriate fuel calculation. As covered in the previous section, DFSO=0 implies that no alteration of the fuel flow may be made, so the rate calculated upstream is passed directly through the fuel flow submodel untouched. When DFSO=1, an integrator is initialized with the previous value of the fuel flow (when DFSO just equaled 0) and a constant, calibrated negative slope is integrated until either fuel is totally cut, or DFSO must be exited. A sample of fuel shutoff events was sampled and an initial representative slope \( \Delta h_{\text{fuel}} / \Delta t \) was determined that applied to both on-off, and off-on situations. A model-in-the-loop script was then written to iteratively improve the slope. In reality, a specific cylinder fuel cut pattern is followed, but the slope method was nonetheless adopted because of its
simplicity. When DFSO eventually equals 2, the last switch port becomes active and a similar method used in the down-slope phase is applied to gradually increase fuel flow back to unaltered levels. There are a few differences however. First, the initial condition of the integrator must be set to the previous time-step level of the altered fuel flow and must be reset to zero when phase two is not active to prevent incorrect initializations caused by memory blocks. Secondly, logic is included to break phase 2 when fuel flow reaches its unaltered level multiplied by a gain. A gain slightly smaller than one is required as a predictive measure because the break logic is one time step behind the simulation to avoid algebraic loop problems. Without the gain, overshoot problems are abundant. A "DONE" command is then sent to the counters & reset model and the DFSO phase transfers from 2 back to 0. At this point the DFSO event is over and the unaltered fuel flow calculated upstream is passed. A first order transfer function (tau=0.1s) filters any instantaneous changes that may occur when switching between states.

**Correction to Engine IMEP**

A correction to the engine IMEP submodel was applied in order to successfully implement DFSO. The model implemented is slightly more complex than the fuel flow submodel because the slopes involved are not constants, but are rather determined anew for each DFSO event. IMEP is a function of the total mass flow into the engine which includes both air and fuel. Since in a port-fuel injected stoichiometric SI engine the fuel represents a significant fraction of the total mass flow, the IMEP model perceives a complete cut in fuel only as a slight decrease in the total mass flow, and hence may not fully eliminate IMEP due to errors and approximations in the IMEP model (which is
solely based on fired conditions). Note that, in this work, IMEP is considered as the *gross* IMEP, hence deprived of the pumping losses (PMEP). In this sense, the lower limit of the engine IMEP is assumed zero and occurs when no fuel is injected.

The IMEP submodel therefore employs the same multiport switch format as described for the fuel flow submodel. A similar "done" check is performed during phase two and a "DONE2" command is sent to the counters & resets model when IMEP reaches its unaltered level. Either "done" command, fuel flow or IMEP, will terminate phase 2. The primary difference between the IMEP and fuel submodels is the calculation of slope. The decision was made to model the shutoff and turn-on of the fuel with the integration of a constant slope. Since IMEP is approximately proportional to the fuel flow, both IMEP and fuel must reach their terminal values at roughly the same time. This means that IMEP slopes must be continually recalculated to take into account the varying initial conditions of each DFSO event. Figure 60 outlines the IMEP slope calculator.
Figure 60: DFSO IMEP slope calculator

The IMEP slope calculator uses integrators with triggered external resets to capture instantaneous values of both IMEP and fuel flow by integrating zero and only varying the initial condition. A snapshot of the Simulink diagram was included to show the intricacies of the integrator reset conditions. For use in phase 1, quotient 1 calculates the time required by the fuel to reach zero by rearranging (3.52). Quotient 2 then finds the required IMEP slope given the $\Delta t$ for fuel.
Fuel Slope = \frac{\Delta \dot{n}_{\text{fuel}}}{\Delta t} \\
IMEP Slope = \frac{\Delta \text{IMEP}}{\Delta t} \tag{3.52}

The same methodology is used for the IMEP up-slope but the initial conditions are taken as the unaltered fuel flow and IMEP at the beginning of phase 2. **Quotient 3** calculates the estimated time required for fuel to reach its unaltered value if constrained to the calibrated slope, and **quotient 4** finds the corresponding IMEP slope to match that time. The IMEP up slope is an estimate, but the relative brevity of phase 2 compared to the down slope of phase 1 reduces the accuracy requirement. Figure 61 superimposes the simulated fuel flow rate with the simulated engine IMEP (after implementing the aforementioned correction), for two DFSO events. It can be seen that the two traces are fundamentally related and behave as intended during fuel cut. Unfortunately, plots like Figure 61 serve as one of the few ways to validate the IMEP submodel because experimental data for IMEP during DFSO are not available.
As mentioned earlier, the limits extracted from Figure 57 for the throttle command $\alpha$ resulted in an ill-conditioned logic section of the DFSO control model. DFSO was being entered far too often and oscillations in fuel flow occurred when the throttle position was near the upper limit condition on alpha. The fundamental reason for the problem is that DFSO is actually a function of pedal, rather than throttle, which is a quandary because the VES driver model outputs throttle position. More details are included in the driver model section, but a significant effort was made to relate throttle to pedal before feed
forward was considered. Unfortunately, the "throttle-by-wire" system used by Chrysler made it impossible to reverse engineer the relationship without significant support, which limited OSU’s options to open-loop commands. A feed forward on pedal position was implemented first with the condition that pedal must equal zero before DFSO can be entered. This simple change drastically improved the behavior of the DFSO control model, but regardless, DFSO mode was still being entered 2-3 times more often than shown by experimental data. This fact implied that other DFSO entry conditions existed that were unknown to OSU. To keep the project on schedule, the DFSO command for a FTP cycle was introduced as a direct feed forward while the additional entry conditions could be established. The feed-forward map used in the VES was extracted from experimental data using the fuel shutoff pattern number variable described earlier. First, the index values for pattern number = 10 (fuel fully off) were found and the start/end points of each set were identified. Multiple DFSO events were sampled, and for events during which the highest pattern number attained was 10, roughly one second was required to fully cut fuel. The DFSO start time was therefore offset 1 second earlier than the first observed pattern number of 10 for each event. The value of 1 second was set as a calibration parameter in the map creation script and could be changed in the event of a different cylinder deactivation strategy. Figure 62 shows an example of start and stop times for two DFSO events.
A DFSO command versus time map was generated by assigning a non-zero value to every green and red point in Figure 62. In conjunction with a "HitCross" block, an impulse-like signal was created for use with the rising-edge counter in Figure 59. Figure 63 shows a validation of simulated fuel flow for a representative cross-section of DFSO events. Figure 64 is a close-up of a single event that clearly displays the accuracy of the calibrated phase 1 and phase 2 slopes. The only improvements that could be made to the FF map would be to calibrate the start time offset for each event, and to include DFSO events during which full fuel cut is not achieved. These events are not currently included because they are few in number and extremely short in duration.
Figure 63: Validation of fuel flow during DFSO (4 events shown)

Figure 64: DFSO validation, close-up of a single event
3.3.6 Driver Model

The driver model began as a standard PID control on the error between the desired velocity profile (determined by the specific input driving cycle) and the actual vehicle velocity [43]. Over time, complexity was added to ensure a smoother response on the accelerator and pedal positions, and to simulate more realistic characteristics of vehicle longitudinal dynamics under real-world usage conditions.

The main enhancements that have been introduced in the driver model are here illustrated.

1. Separate PID gain sets were used for braking and accelerating to capture the fact that the situations are completely different in terms of the force garnered from a given pedal position. Within the PID blocks, the integrators are reset both when entering and exiting idle condition to avoid windup issues.

2. A low pass filter (implemented as a first order transfer function) was placed after the PID outputs to attenuate high frequency signals that would otherwise result in unrealistic chattering in the pedal commands.

3. Idle speed controllers were added to the accelerator (α) and brake (β) commands to control the vehicle during zero cycle velocity situations. In such events, the brake

Figure 65: Driver model I/O
command, beta, is simply set to a constant to prevent the vehicle from moving due to non-zero turbine torque. The accelerator idle speed controller is a bit more complex. If the cycle velocity is neither decreasing nor zero, the accelerator command determined by the PID block is passed through untouched sans a saturation constraining the command to physical throttle limits. If the cycle velocity does indicate an idle requirement, an embedded PID control maintains engine speed around the desired value until idle is exited.

*Output Accelerator Command (α)*

The final accelerator command value leaving the driver model is the summation of a throttle feed-forward (FF) term and a PID command that slightly modifies the FF value to ensure that the vehicle follows the specified trajectory. During the phases where the cycle velocity is zero, the PID addition is forced to zero such that the FF outputs the idle α trace seen in experiments. The option to bypass the idle speed controller described earlier was one of the motivations for implementing a feed-forward, as the idle speed controller tended to oscillate. A synchronization function is used to line up the FF map and the drive cycle time vectors.

*Output Brake Command (β)*

Beta is calculated using a PID controller coupled with some conditions that allow for coasting. When the α command is less than a specified value and the velocity error is negative, an enabled subsystem containing the braking PID controller becomes active. The "less than" condition on α allows very small throttle angles to occur, as would be
expected in coasting, without the braking system to be enabled. When the driver model is eventually made to output pedal position (see next section), the conditional braking entry logic will be slightly modified to simply require a zero accelerator pedal position opposed to a minimum throttle position. The output of the brake PID to sent to a switch that only passes the signal if the enabled subsystem is active. This effectively disallows $\alpha$ and $\beta$ to be nonzero concurrently. Though no FF is included for $\beta$, the added importance of an accurate $\beta$ signal required by the models described in section 4.3 may warrant it.

A few other additions to the driver model were explored and may be implemented in the future if required. The most interesting of these is the introduction of deadzones scheduled on vehicle velocity. This idea was developed for two reasons: to mimic the fact that a human driver can follow a trace more accurately at lower speeds, and to make the PID block less reactive by allowing a band of acceptable error.

**Throttle vs. Pedal**

As touched upon numerous times throughout the model development chapter, the VES driver model outputs the accelerator command $\alpha$. In reality, this command is directly fed through the throttle position command, rather than being filtered and processed through a torque-based control logic (as typically implemented in modern SI engine torque control systems). While this approximation does not cause inaccuracies in the model when simulating driving cycles and predicting fuel economy values, it may represent an issue
when more advanced control strategies will be implemented and tested in the VES. For this reason, a summary of the various states and submodels that are directly dependent on the driver command $\alpha$ is reported in Table 13. This should serve as a reference for future extensions and modifications of the VES.

Table 13: Current and future states of models dependent on alpha

<table>
<thead>
<tr>
<th>Model</th>
<th>Current Status</th>
<th>Future Status</th>
</tr>
</thead>
<tbody>
<tr>
<td>Throttle Body</td>
<td>Throttle from driver</td>
<td>Pedal from driver</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Subsystem relating pedal to throttle position</td>
</tr>
<tr>
<td>DFSO</td>
<td>Pedal FF (standalone)</td>
<td>Pedal from driver</td>
</tr>
<tr>
<td>Shift Schedule</td>
<td>Throttle from driver</td>
<td>Pedal from driver</td>
</tr>
<tr>
<td></td>
<td>Modified shift maps</td>
<td>Original shift maps</td>
</tr>
<tr>
<td>Lockup Logic</td>
<td>Throttle from driver</td>
<td>Pedal from driver</td>
</tr>
<tr>
<td></td>
<td>Modified lockup maps</td>
<td>Original lockup maps</td>
</tr>
</tbody>
</table>

The decision to output throttle position rather than pedal position was made for primarily one reason: throttle-by-wire. Pedal and throttle, though fundamentally related, do not behave in a way that is fully understandable from a third party viewpoint. As one can see in Figure 66, throttle position is a higher frequency trace compared to pedal and is often operating in a way not implied by the pedal position. One very important thing to note is that values for both pedal and throttle opening go beyond 100% for the experimental data. The reasons for this issue stem from proprietary Chrysler calibration procedures which are unknown to OSU. Each vector was normalized by its maximum value to yield usable signals when needed.
In the test vehicle available at OSU, no physical link exists between the pedal position and the throttle, making it very difficult to reverse engineer a relationship that is no doubt dependent on the operating conditions of the engine, etc. An attempt was made to relate the two by dividing the cycle into acceleration, deceleration and idling regions to observe the traces during similar vehicle operation. This method holds promise, but a decision was made to use a throttle FF until added support from Chrysler's controls group could be recruited. Figure 67 displays throttle position versus pedal position which further reinforces the nonlinear, erratic relationship.
Throttle position FF was chosen over pedal position due to its greater complexity and less flexible end use. The models that do require pedal position (i.e. shift and lockup) were temporarily altered to accommodate throttle position with the exception of DFSO. Correct entry into DFSO mode is so precisely dependent on zero pedal position that no substitute could be found with respect to throttle. In fact, it was determined that at the beginning of zero-pedal events during the FTP (points of interest to DFSO), throttle was found to vary between roughly 8 and 24 percent. This range made it impossible to set an upper throttle limit for DFSO entry that wouldn't induce early, late or excessive DFSO events.
Section 3.4 VES Validation

Validation of the VES was performed using a ground-up approach: functions, then submodels, then component models, then systems, and so on. Much of the validation effort for the mechanical systems was performed well before the entire VES model could even run as a whole. Once all the major sub assemblies had been connected, VES-level validation was carried out on the EPA Federal Test Procedure (FTP) as depicted in Figure 68. Only the first 1350 seconds of the FTP were considered so that simulation results could be compared to experimental data provided by Chrysler. The cold start phase and the transient phase of the FTP represent the Urban Dynamometer Driving Schedule (UDDS). From this points forward, "FTP" implies the shortened version equivalent to the UDDS.

![EPA Federal Test Procedure](image)

Figure 68: FTP drive cycle [44]
The following is a series of validation plots that capture the most important mechanical and fuel consumption variables.

![Figure 69: VES velocity validation](image)

![Figure 70: VES engine speed and throttle validation](image)
Figure 71: VES engine speed, fuel flow rate and fuel consumption validation

Table 14: RMS error, mean error and standard deviation for major validation variables

<table>
<thead>
<tr>
<th></th>
<th>RMS Error</th>
<th>Mean Error</th>
<th>Standard Deviation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vehicle Velocity [m/s]</td>
<td>0.24</td>
<td>-0.53</td>
<td>1.23</td>
</tr>
<tr>
<td>Engine Speed [rpm]</td>
<td>272.9</td>
<td>-89</td>
<td>258</td>
</tr>
<tr>
<td>Engine Torque [N]</td>
<td>23.2</td>
<td>6.1</td>
<td>22.3</td>
</tr>
<tr>
<td>Fuel Flow Rate [g/s]</td>
<td>2.3E-4</td>
<td>-5.2E-5</td>
<td>2.2E-4</td>
</tr>
</tbody>
</table>

The cumulative fuel consumption error at the end of 1350 seconds of the FTP is 0.02 kg which represent roughly a 2% error.
Chapter 4: Vehicle Energy Analysis

This chapter illustrates the application of the VES to a feasibility study oriented to estimate the opportunity for fuel consumption reduction through vehicle brake energy recovery on a conventional (non-hybridized) vehicle. First, a simple vehicle road-load analysis is conducted on a set of regulatory driving cycles to determine the brake energy recovery opportunities in “ideal” conditions (i.e., not constrained by the powertrain efficiency and by the energy storage capacity and power limits).

Section 4.1 Unconstrained (Ideal) Cycle Analysis

An analysis of the energy and power requirements at the wheel was performed on two drive cycles to establish statistics that would later assist in establishing the energy and power availability for engine crankshaft harvesting opportunities. The study was done on a FTP cycle as well as on an Artemis Urban cycle as shown in Figure 72 and Figure 73 respectively. Only the first 1350 seconds of the FTP were considered so that simulation results could be compared to experimental data. A human-driven case was not analyzed because a well driven pre-defined drive cycle is nearly identical to the ideal trace.
Figure 72: Vehicle velocity profile over the FTP Drive Cycle [44]

Figure 73: Vehicle velocity profile over the Artemis Urban drive cycle [45]
Some general characteristics of the two aforementioned cycles are listed in Table 15.

<table>
<thead>
<tr>
<th></th>
<th>FTP</th>
<th>Artemis Urban</th>
</tr>
</thead>
<tbody>
<tr>
<td>Duration [s]</td>
<td>1350</td>
<td>993</td>
</tr>
<tr>
<td>Distance [km]</td>
<td>11.8</td>
<td>4.9</td>
</tr>
<tr>
<td>Average Speed [km/hr]</td>
<td>31.6</td>
<td>17.7</td>
</tr>
<tr>
<td>Top Speed [km/hr]</td>
<td>91.2</td>
<td>57.7</td>
</tr>
</tbody>
</table>

From the velocity traces alone, a required power profile can be generated and then analyzed to reveal the statistics explored later in this section. The classic vehicle dynamics equation is

\[ m \frac{dV}{dt} = F_i(t) - F_{RL}(t) \]  

(4.1)

where \( F_i(t) \) is the tractive force at the wheel and \( F_{RL}(t) \) is the road load force [31].

Rearranging (4.1) and converting to power yields

\[ P_i(t) = F_i(t)V(t) = \left( m \frac{dV}{dt} + F_{RL}(t) \right) V(t) \]  

(4.2)

As concluded in the road load section of the vehicle dynamics model description, an aggregate road load equation of the form

\[ F_{RL} = CV^2 + BV + A \]  

(4.3)

was adopted. In this case, the three coefficients that characterize the grade, tire rolling resistance and aerodynamic drag for the test vehicle considered in this study were taken directly from the EPA website, which lists recommended values for the dyno coefficients for a variety of production vehicles. This equation could easily be replaced using the individual terms representing aerodynamic, grade and rolling resistance forces. By
combining (4.2) and (4.3), one will notice that power is solely a function of velocity and time and hence a power profile can be generated starting solely from the cycle velocity profile. The power required at the wheel for the FTP and Artemis Urban cycles is shown in Figure 74 and Figure 75 respectively.

Figure 74: Vehicle power profile at the wheel for the FTP cycle
Four distinct power regions can be defined based on the instantaneous sign of the instantaneous power at the wheel and the vehicle velocity:

1. \( P > 0 \) \( \Rightarrow \) Traction Phase
2. \( P < 0 \) \( \Rightarrow \) Braking Phase
3. \( P = 0 \) and \( V \neq 0 \) \( \Rightarrow \) Coasting Phase
4. \( P = 0 \) and \( V = 0 \) \( \Rightarrow \) Stopped Phase

The braking phase is of interest for this part of the analysis since it represents the only region in which energy recovery is possible through crankshaft energy harvesting.

Energy (fuel) saving would be the goal in the coasting and stopped phases. To isolate the
braking events, the beginning and ending times had to first be identified which were found by observing the zero crossings of the power profile. Once the start point and duration of each braking event were determined, the power at the wheel, energy and velocity matrices where populated based on event number. Basic MATLAB commands were then used to establish statistics about the drive cycles, the most important of which are listed in Table 16. Since the analysis from this point forward deals mainly with braking events which require negative power/energy, absolute values will be used for simplicity.

<table>
<thead>
<tr>
<th>No. of braking events</th>
<th>FTP</th>
<th>Artemis Urban</th>
</tr>
</thead>
<tbody>
<tr>
<td>Average braking time [s]</td>
<td>9.3</td>
<td>8.1</td>
</tr>
<tr>
<td>Total positive energy [kJ]</td>
<td>7.6E3</td>
<td>4.2E3</td>
</tr>
<tr>
<td>Total braking energy [kJ]</td>
<td>3.2E3</td>
<td>2.8E3</td>
</tr>
<tr>
<td>Max braking energy [kJ/event]</td>
<td>261.5</td>
<td>177.5</td>
</tr>
<tr>
<td>Average braking energy [kJ/event]</td>
<td>59.6</td>
<td>42.8</td>
</tr>
<tr>
<td>Max braking power [kW/event]</td>
<td>37.5</td>
<td>64.3</td>
</tr>
<tr>
<td>Average max braking power [kW/event]</td>
<td>11.9</td>
<td>14.4</td>
</tr>
<tr>
<td>Average mean braking power [kW/event]</td>
<td>4.4</td>
<td>6.2</td>
</tr>
<tr>
<td>Average max braking velocity [m/s]</td>
<td>12.1</td>
<td>8.0</td>
</tr>
<tr>
<td>Average mean braking velocity [m/s]</td>
<td>9.3</td>
<td>5.6</td>
</tr>
</tbody>
</table>

The green highlighted value in Table 16 is of particular importance because it will serve as the normalizing factor for all the remaining energy recovery analyses. This value represents the theoretical maximum amount of energy that can be recovered over the first 1350 seconds of the FTP. Such a baseline will help give perspective to the remaining results. The only exceptions are the engine stop/start and DFSO fuel saving results.
These values have been normalized by the total energy required calculated directly from fuel usage.

Two of the most important reasons for performing road-load driving cycle analysis are to determine the frequency of various maximum power levels, and the cumulated energy associated with said maximum power levels. This information is immediately useful to understand the potential of any existing energy reservoirs, and would also be necessary for future energy storage sizing problems. To categorize braking events based on their maximum negative power level, power "bins" were generated with constant spacing from zero to the maximum braking power observed. An identical approach was taken to create energy bins. A script was written to place the energy value of each braking event into the power bin corresponding to the maximum braking power of that event. Then, the energies in each power bin were sorted into their respective energy bins such that an $m$-by-$n$ matrix was defined where $m =$ number of power bins and $n =$ number of energy bins. The values populating the matrix are the number of braking events observed at the intersection of each power and energy bin. The actual matrix is very difficult to visualize in 3D space due to the large number of bins without any residing events, so additional logic was included in the analysis script to cluster adjacent bins into larger bins for better visualization. Figure 76 is a pictorial representation of the master power x energy matrix $g_o$ from which later plots are derived.
The frequency of each power level can simply be defined as the number of events at that power level \((N_i)\) divided by the total number of events \((N)\).

\[
f_i = \frac{N_i}{N}
\]  

(4.4)

The frequency of each maximum power level over the cycles analyzed is plotted in Figure 77 and Figure 78.
The energy accumulated at each power level can be found by first multiplying the number of events in each energy bin by that energy bin's average value, and then summing the resulting energies. In mathematical notation:

$$CE_i = \sum_{j=1}^{n} N_j E_{avg_j}$$

(4.5)

The energy accumulated at each power level divided by the total energy for the whole cycle is plotted for the FTP and Artemis Urban cycles in Figure 79 and Figure 80.
Figure 79: Energy cumulated at different power levels (FTP)

Figure 80: Energy cumulated at different power levels (Artemis Urban)
Section 4.2 Constrained Cycle Analysis (FTP)

The analysis done in section 4.1 lays the foundation for a more realistic constrained analysis which introduces the following five considerations:

1. Power limit of the energy storage system
2. Energy limit of the energy storage system
3. Power transmission losses (efficiency)
4. Engine-level power trace
5. DFSO events only

The first three points capture the fact that any energy storage system has some physical characteristics which will drastically affect the amount of energy recoverable from a given drive cycle. The fourth point stems from the fact that in a conventional vehicle the only opportunity for energy harvesting exists at the engine crankshaft, rather than at the vehicle wheels. This is an important concern because the efficiencies of the drivetrain components (torque converter, transmission, differential) will considerably reduce the amount of energy available at the crankshaft during overrun operations. Constraint 5 is not considered in this section, but it is taken into account when determining the effectiveness of the scenarios proposed in Section 4.3. Figure 81 displays two representative negative power events taken directly from the power profile calculated for the road-load driving cycle analysis. The colored areas represent energy, the red being a positive energy requirement where no recovery is possible. The green area signifies recoverable energy which is subject to both power and energy constraints. The orange
area corresponds to energy that is unrecoverable due to possible limitations in the maximum power that can be absorbed during the recovery phase (here assumed to be 15 kW). Similarly, the blue area represents energy that cannot be recuperated because the capacity of the storage medium has been reached. Again, the actual limit was chosen purely for visualization purposes. In most cases, the limiting factor is power rather than energy because braking events are relatively short. This is the case for event 1 in Figure 81.

Figure 81: Qualitative representation of power and energy limits (Red: positive (traction) power, Green: negative power (recoverable), Orange: power that cannot be recovered due to power limits, Blue: power that cannot be recovered due to energy limits)
To determine the power profile available at the crank, an FTP cycle (see Figure 72) had to first be simulated to establish the powertrain efficiency as a function of time. This efficiency trace, which is representative of the VES powertrain under FTP conditions, was then used to translate the idealized power profile at the wheel to the engine crankshaft. Calculation of transmission and torque converter efficiency is described in detail in the appropriate modeling sections. Figure 82 displays the ideal wheel-level power profile, the powertrain efficiency trace, and the resulting engine-level power which was then used for the constrained analysis.

\[ \text{Figure 82: Power profiles at the wheel and at the crankshaft during the first 70s of a FTP cycle (top), and related drivetrain efficiency (bottom)} \]
Notice that in Figure 82, the engine crankshaft power profile is greater in magnitude than the wheel power profile during normal (i.e., powered) operations and the opposite is true during overrun. This is due to the inversion of the power flows on the vehicle, which causes the efficiency to switch to its reciprocal when wheel torque transitions from positive to negative. Once the engine-level power profile was established, a Simulink model was developed to impose various real-world constraints that ultimately reduce the amount of recoverable energy. Figure 83 displays the model and the seven primary sections which are explained thereafter. A Simulink diagram is provided directly to show the intimate details of the switches and integrator reset conditions.

**Figure 83: Block Diagram showing the Simulink model used to conduct the constrained energy**

**Section 1** is a 1D lookup table that feeds the model of the engine-level power profile as a function of time. Figure 82 depicts this trace in detail. **Section 2** consists of a switch that passes only the negative portions of the power profile to capture solely the braking events (zero is passed otherwise). Additionally, a negative one gain switches the sign for easier manipulation downstream. **Section 3** first multiplies the power by an efficiency defined in an initialization file and then compares the instantaneous power to the power limit set in the same initialization file. If the power is at or above the limit, the power limit
constant is passed to the next block. **Section 4** determines the braking event energy by integrating the input power signal. A "HitCross" block is used in conjunction with the integrator's external reset functionality to reset the event energy to zero when the braking event is over. The event energy is then compared to the pre-defined energy limit in **section 5** where a switch passes zero if the energy limit has been reached. Otherwise, the power signal entering the event energy integrator is passed onto the cumulative energy integrator in **section 6** which outputs the constrained recoverable energy to the MATLAB workspace. **Section 7** uses a "HitCross" and a rising edge counter to create a braking event number array that can be synced with the other signals.

Two assumptions were made to simplify the constrained analysis model. First, the efficiency of power transfer between the crankshaft and the components that allow for energy storage was assumed to be constant (90%). This simplification can easily be relaxed by implementing an efficiency term dependent on speed and torque conditions. Second, the energy recovered during a negative power event is taken to somehow be "used up" fully on the following positive power event. This assumption allows the event energy integrator to be reset after each event such that the energy reservoir is "empty" upon entry into the next negative power situation.

The constrained energy model just described was embedded within a "for" loop so model-in-the-loop simulations could be run to characterize the constrained recoverable energy over a wide range of power and energy limits. Figure 84 is the resulting 3D map normalized by the theoretical maximum amount of recoverable energy determined in the
unconstrained analysis section. As one would expect, the most energy can be recovered when the power and energy limits are largest. The power and energy vectors used to create the map both have maximum values based appropriately given the highest event power and energy determined from the ideal cycle analysis in Section 4.1.

Figure 84: Recoverable energy constrained by power and energy limits, normalized by maximum unconstrained recoverable energy

**FTP Start/Stop and DFSO Comparison**

After establishing a baseline for the theoretical amount of energy that can be recovered at the wheel during decelerations, the validated VES was used to estimate the fuel consumption benefits of technologies such as engine start/stop (SS) and deceleration fuel
shutoff (DFSO). With DFSO disabled, the VES was run over the first 1350 seconds of the FTP cycle to establish a baseline condition. The DFSO control logic was then re-enabled and the same test was run.

In order to evaluate the benefits of engine start/stop on fuel consumption, a simple analysis was conducted in post-processing, namely by modifying the baseline engine instantaneous fuel flow profile such that the fuel flow rate equaled identically zero during periods of zero cycle velocity. Note that this procedure is a considerably simplified case, with respect to a real engine stop/start strategy. In a real case, the need to power auxiliary loads and to maintain a minimum catalyst temperature during standstills limit considerably the amount of energy that can be saved through engine stop/start operations. Additionally, drivability concerns and emissions problems upon engine restart are not considered in this simplified case study. Therefore, the following results have to be interpreted as a limit condition, to establish the theoretical maximum benefits achievable through the implementation of this strategy.

In the results that follow, the driving cycle velocity profile was used as opposed to vehicle velocity to ensure that the study results would be less dependent on an accurate driver. The three fuel flow traces (baseline, DFSO and SS) are plotted together in Figure 85. These traces were also integrated with respect to time to yield cumulative fuel flows which are plotted in Figure 86. The combined benefit of DFSO and SS is also included.
Figure 85: Fuel flow rates for different fuel saving techniques (top: FTP, bottom: close-up)
Figure 86: Cumulative fuel flow for different fuel saving techniques

Cycle-length statistics can be calculated based on the final value each cumulative fuel flow vector. In (4.6) FC is fuel consumption and FE is fuel economy.

\[
FC = \max \left( \int \dot{m}_{\text{fuel}} \, dt \right) \quad \text{and} \quad \%\Delta FC = \frac{FC_{\text{base}} - FC_{\text{DFSO,SS}}}{FC_{\text{base}}} \times 100
\]

\[
FE = \left( \frac{1}{\rho_{\text{fuel}}} \right) \times \max \left( \int \dot{m}_{\text{fuel}} \, dt \right) \quad \text{and} \quad \%\Delta FE = \frac{FE_{\text{base}} - FE_{\text{DFSO,SS}}}{FE_{\text{base}}} \times 100
\]

(4.6)
Table 17: Fuel consumption benefits of stop/start and DFSO compared to baseline

<table>
<thead>
<tr>
<th></th>
<th>Decrease in Cumulative Fuel Use [kg]</th>
<th>Decrease in Fuel Consumption [%]</th>
<th>Increase in Fuel Economy [%]</th>
<th>Energy Savings at The Wheel [kJ]</th>
<th>Normalized Energy Savings [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>DFSO Enabled</strong></td>
<td>0.0257</td>
<td>2.55</td>
<td>2.62</td>
<td>84</td>
<td>1.2</td>
</tr>
<tr>
<td><strong>Stop/Start Enabled</strong></td>
<td>0.0803</td>
<td>7.99</td>
<td>8.68</td>
<td>362</td>
<td>5.0</td>
</tr>
<tr>
<td><strong>Both Enabled</strong></td>
<td>0.1059</td>
<td>10.53</td>
<td>11.77</td>
<td>446</td>
<td>6.2</td>
</tr>
</tbody>
</table>

The wheel level energies in Table 17 are calculated using:

\[
E_{whl} = \int P_{whl} dt = \int \eta_{pwrtn} P_{eng} dt = \int \eta_{pwrtn} (\eta_{eng} m_{fuel} Q_{LHV}) dt \quad (4.7)
\]

The total energy required at the wheel for the entirety of the cycle is 7.2E3 kJ as calculated using (4.7) with baseline fuel flow rate. This value has been used to normalize the energy saving results presented above. Based on the results shown in Table 17, both DFSO and SS represent very promising technologies for the reduction in fuel consumption. The relative impact of SS is greater than that of DFSO because of the amount of time spent at idling during the two urban driving cycles. A more aggressive DFSO entry strategy would increase the fuel economy benefits, but would likely negatively affect drivability.
Section 4.3 Preliminary Control Design for Maximization of Crank Shaft Energy Harvesting

The previous two sections aimed to establish both the tools and practices to analyze drive cycles from an energy recuperation perspective. Though the results of those sections are useful from a high-level point of view, the opportunities for brake energy recovery in a conventional vehicle are much more restricted. The constrained and unconstrained analyses set the premises and established an upper bound to the amount of negative energy recapture during vehicle deceleration and braking. On the other hand, brake energy recovery can be made effective only if the vehicle powertrain is specifically equipped with a bi-directional energy converter and a storage system (for instance, an electric motor/generator connected to a high-voltage battery pack).

In conventional vehicles (i.e., not equipped with a secondary energy storage system), regeneration opportunities are considerably limited, and are mostly implemented through a “smart” control of the power consumption of the vehicle ancillary loads (such as the alternator, the A/C compressor, or the coolant pump).

Furthermore, energy harvesting in a conventional vehicle can only occur during very specific situations, not simply every time the wheel-level power requirement is negative. This is an important statement as it reduces considerably the number of recovery events during a driving cycle, and also shortens the duration of each negative power event.
For the above reasons, the constrained engine-level power profiles determined in section 4.2 cannot be used directly because they do not take into account how that profile is obtained. Given these real world constraints, the goal of this section is to characterize and maximize the amount of energy that can be recovered on a conventional vehicle, using control strategies that operate on the primary energy converters (i.e., engine and transmission). The outcome of this work is to formulate a set of guidelines for the design of engine and transmission control strategies that will enable for recovery energy at the crankshaft during vehicle deceleration and braking. Note that the utilization of the recovered energy, which would require a coordinated control of the vehicle ancillary loads, is not treated here, but is considered as part of the future work.

4.3.1 Disturbance Torque Calculation

The first step in characterizing the energy available at the crankshaft is to understand how to apply a load at the engine crankshaft level that will not adversely affect the net power delivered to the wheels during fuel cut-off. This is required to avoid, as much as possible, NVH and drivability issues. There are two ways to accomplish this:

1. Reduce the subtractive components of the engine IMEP, so that an additional load may be applied to the engine without changing net engine brake torque. During DFSO, the engine net IMEP is largely negative, due to the presence of pumping losses. Consequently, a negative BMEP is obtained, also due to the presence of friction losses. This magnitude of the negative BMEP can be reduced (in absolute value) by finding ways to mitigate the FMEP or the PMEP. Since the FMEP is primarily dependent on mechanical friction and temperature-dependent viscosity of the engine oil, this work will
focus only on attempting to reduce the engine PMEP through control of the throttle position during negative acceleration events. This method of introducing a disturbance load is only applicable during fuel cut because it requires large manipulations of throttle.

2. Reduce the percentage of braking force supplied by the friction brakes and offset the reduction with an engine-level load. Brake-by-wire allows the braking torque requested by a driver to be divided amongst numerous systems that can work together to reduce the speed of the vehicle. Theoretically, implementation of this method could occur during every braking event, which is essentially the function of regenerative braking hardware in hybrid vehicles. Since this project focuses on conventional vehicles, brake displacement has only been considered during fuel cut so the results can be compared to method 1. The tools and methodology developed herein could easily be extended to all braking events if such an analysis were deemed to be in scope.

Given the two methods above, three scenarios have been devised to incrementally increase the amount of energy available for harvesting. Models for each scenario have been implemented directly into the VES and can be activated or deactivated depending on the desired configuration. Additionally, a post-processing script has been written to estimate results offline given model outputs calculated in prior simulations. Specifically, the script requires seven data sets: simulation time, PMEP, DFSO, engine speed, gear, position of the brake pedal and efficiency of the transmission. The post-processing method takes into account more assumptions, but allows for rapid and flexible changes in recovery strategies that can be implemented in the VES once validated. The script is a very useful tool to determine approximated results before time-intensive simulations are
undertaken. The following methodology explanations are based on the Simulink models used for the analysis, but due to the variety of scenarios explored, results from the post-processing script are presented.

Scenario 1: PMEP Reduction - Baseline Powertrain Control Strategies

The overarching goal of scenario 1 is to maximize energy recovery without affecting the baseline powertrain control strategies or friction brake command. This has been accomplished by reducing the engine PMEP during the deceleration phases where DFSO is active through manipulation of the throttle position. A negative disturbance torque proportional to the effective decrease in PMEP is added to the engine dynamics node to compensate for the variation in the engine brake torque during deceleration. As partial foreshadowing to scenario 2, PMEP had to first be characterized as a function of throttle position and engine speed. This was done by fitting a surface to engine steady state data that included a wide range of engine speeds and torques. 70% of the Big Grid points were used to establish the initial curve fit, and the remaining 30% were used as validation points. A third order polynomial in both alpha and engine speed was found to fit the data very well. The functional form of the fit is shown in Equation (4.8) and is plotted with the experimental data points in Figure 87.
\[ \text{PMEP [Pa]} = p_{00} + p_{10}\alpha + p_{01}N + p_{20}\alpha^2 + p_{11}\alpha N + \ldots \]
\[ p_{02}N^2 + p_{30}\alpha^3 + p_{21}\alpha^2 N + p_{12}\alpha N^2 + p_{03}N^3 \]
where
\[ p_{00} = 8.659E4 \quad p_{10} = -3449 \quad p_{01} = 3.838 \]
\[ p_{20} = 36.67 \quad p_{11} = 0.1341 \quad p_{02} = 0.002093 \]
\[ p_{30} = -0.1061 \quad p_{21} = -0.002924 \quad p_{12} = 5.47E-5 \]
\[ p_{03} = -4.317E-7 \]

Figure 87: Experimental engine PMEP as a function of engine speed and throttle position, based on steady-state engine operating conditions

Other than at very high engine speeds, the trend observed in Figure 87 is that PMEP decreases as throttle opens, which follows from intuition; less restricted air flow reduces pumping losses. Since scenario 1 only allowed variations in throttle, optimal values for
the throttle opening as a function of engine speed can be extracted from the PMEP fit, to
achieve the minimum PMEP along each engine speed value. Figure 88 and Figure 89
show the optimized throttle position and the corresponding PMEP as a function of engine
speed in a 3D and 2D representation, respectively. Only a speed range typical to DFSO
entry conditions has been shown. As is immediately obvious, the optimal throttle
opening for minimizing PMEP during DFSO is 100% (shown in blue).

![Figure 88: Optimal alpha to minimize PMEP as a function of engine speed during DFSO (3D)](image)

Since the engine under consideration for this project is naturally-aspirated, negative
PMEP values are not possible and only occur in Figure 87 and Figure 88 because of the
fit. These values have been therefore removed to ensure that optimal PMEP is not
positively contributing to the brake torque during DFSO. In addition, it was noted that the negative values occurred only during low speed values. At these conditions, the DFSO is typically not active, but the idle speed control of the engine is activated. Therefore, these conditions are typically not considered for crankshaft energy harvesting.

![Graph](image)

**Figure 89: Optimal alpha to minimize PMEP as a function of engine speed during DFSO (2D)**

Once data displayed in Figure 89 was converted into 1D lookup tables, a baseline FTP was simulated to so that unaltered PMEP could be documented. This "baseline PMEP" is used to calculate the variation in the engine PMEP ($\Delta$PMEP) when optimal throttle positions are imposed. This term represents potential energy that could be harvested at the crankshaft without varying the overall engine brake torque during vehicle
deceleration, therefore remaining “transparent” to the driver. The following steps illustrate the energy harvesting strategy that was implemented in the post-processing script for Scenario 1, and Figure 90 is a schematic of the corresponding model implementation.

**Step 1a:** Load feed forward data for DFSO phase, engine speed, PMEP, velocity, alpha-throttle, Beta-brakes, turbine speed, engine torque, gear position, lockup status and transmission efficiency.

**Step 1b:** Extract data from each map during DFSO only.

**Step 2:** Interpolate in the 1D Optimal PMEP vs. Engine Speed map to determine the lowest possible PMEP given engine speed. This step assumes that the throttle is moved to its optimal opening and PMEP will respond according to the surface fit.

**Step 3:** Determine $\Delta$PMEP as the difference between baseline PMEP and optimal PMEP.

**Step 4a:** Convert $\Delta$PMEP into an effective crankshaft disturbance torque.

**Step 4b:** Convert torque into power by multiplying by instantaneous engine speed.

**Assumptions:**

a) The original fit for $\text{PMEP} = f(N_{\text{eng}}, \alpha)$ based on steady-state data is sufficient to determine PMEP during transients.

b) The calculation of disturbance torque is perfect and therefore will not affect the braking rate of the vehicle.
ΔPMEP is converted into an effective torque using the following equation [6].

\[
T (\text{Nm}) = \frac{PMEP (\text{kPa}) V_d (\text{dm}^3)}{2\pi n_R}
\]  

(4.9)

Current PMEP, rather than PMEP estimated from Figure 89, is subtracted from the baseline PMEP to mitigate some of the inaccuracies inherent in the fit. Steady state data was used to characterize PMEP as a function of engine speed and alpha so transient behavior, which is exactly the case during DFSO, isn't fully taken into account. For this same reason, the optimal alpha calculated based on engine speed is not the global optimal that would result if PMEP was defined based on more conditions (i.e. MAP). Regardless, the optimal alpha passed to the throttle body during DFSO is still results in significant decreases in PMEP.

A feedforward (FF) map for baseline PMEP was chosen over a baseline estimate dependent on unaltered throttle in the spirit of affecting deceleration velocity trace as little as possible. Along this same line of thought, a PID controller on vehicle velocity...
error was added to ensure that the target velocity trajectory could be followed. In theory, a PID should not be required for scenario 1 if the relationship between PMEP, alpha and engine speed is fully understood.

Other inputs not shown in Figure 90 are necessary for activation/deactivation of the torque disturbance calculations. The DFSO activation command is used to enable the aforementioned control logic, since brake energy recovery is possible only during fuel shutoff. A scenario command selects the active components of the disturbance torque calculation and enables the various offshoot support models (i.e. throttle position command override). The “baseline” (i.e., from the engine ECU) throttle position is not required by the torque calculation model but is used to estimate the engine PMEP throughout the cycle to help verifying the map used to determine optimal alpha.

Figure 91 displays the disturbance torque and power results when considering only DFSO events. An interesting observation to note is that disturbance power during DFSO is very well bounded by a range of power levels. This would allow for an every recovery system to be sized and optimized for very specific operating conditions.
The results shown in Figure 91 were calculated using the post-processing script which differs slightly from the model based on some assumptions. First and foremost, the torque calculated using the script is assumed to perfectly offset the reduction in PMEP incurred from varying the throttle. No PID is present and optimal PMEP (see Figure 89) is calculated directly from engine speed, opposed to allowing the VES PMEP model to naturally propagate a lower PMEP when optimal alpha is commanded. These results are not meaningless however, as they validate the potential of the PMEP reduction methodology. The most important difference between the post-processing script and
corresponding VES models is that the simulator includes feedback control in order to make the harvesting activity transparent to the driver; the concepts and actual quantitative results are nearly identical.

**Scenario 2: PMEP Reduction - Improved Powertrain Strategies**

Scenario 2 builds upon scenario 1 by manipulating the baseline powertrain strategies to increase the amount of crankshaft energy that can be recovered during negative acceleration events. In addition to controlling the throttle opening to offset the engine PMEP, energy can be harvested by altering the shift and lockup schedules during DFSO events. This control strategy is sometime referred to as IDFSO - integrated deceleration fuel shutoff. In the proposed strategy, the transmission control unit commands the torque converter lockup during all DFSO events, to maximize driveline efficiency and ultimately maximize the amount of power harvest that can be applied at the engine crankshaft. The optimal selection of gear is significantly more complex. As was the case in scenario 1, the only way to increase the harvest power is to decrease the magnitude of the engine PMEP. When Figure 88 is converted to torque, and then to power, Figure 92 is produced.
The trend in Figure 92 implies that, in order to maximize the crankshaft energy recovery during DFSO, the engine speed should be reduced as much as possible so that the actual power loss due to engine PMEP is minimized. If this can be achieved, the amount of disturbance torque allowable at the crank can be increased. As long as the optimal PMEP remains lower than the baseline PMEP, the correct course of action is often to increase engine speed, but not always. Depending on the baseline engine speed and throttle opening conditions, a map similar to Figure 93 can be dynamically generated to reveal the optimal combination of speed and throttle that allows one to maximize the harvest power that can be applied to the engine crankshaft.
Figure 93: Additional PMEP power available given starting conditions (example)

The surface above is calculated using the difference between a single operating point's conditions compared to the rest of the PMEP map. In this case, the starting conditions were 1700 rpm and near-minimum throttle (14%) which are representative of typical DFSO entry conditions. First, a ΔT map is generated as the difference between the operating point's torque, and the rest of the PMEP-based torque map. Each ΔT point is
then multiplied by the appropriate speed to yield a ΔP map. The maximum value of this map represents the maximum beneficial power difference available given the initial condition. Zeros occur in the map when moving to a point would result in a higher PMEP torque. The convexity in Figure 93 is a consequence of low speeds towards the right hand side, and low (or negative) ΔT towards the left/back. In (4.10), the subscript “o” denotes engine torque, speed and throttle conditions as they are before changing gears and optimizing the throttle position α.

\[ \Delta P_{ij} = (T_o - T_y) \omega_i \]

where

\[ T_y = f(\omega_i, \alpha_j) \quad \text{and} \quad T_o = f(\omega_o, \alpha_o) \]

(4.10)

Other than conditions that exist in the very top-left corner of Figure 88, power gains can usually be made by increasing speed because the baseline throttle position is almost always at its minimum \(T_{ij} < T_o\). Baseline alpha typically is bounded in the lower regions during decelerations for engine braking purposes. The peak of Figure 93 represents the largest improvement in disturbance power possible given a typical starting point. Based on maps calculated using (4.10), logic was written to actively select the optimal gear that would allow the engine speed to be as close as possible to optimal. A simplified schematic of said logic is included in Figure 94. The optimal gear is determined based upon the difference between optimal speed and the speed that would result if each gear were imposed. A comparison between powers would also suffice.
As shown in Figure 94, a gear may only be selected if the minimum and maximum engine speeds for that gear are not violated. A simplified Simulink diagram was included to show the intricacies of the switching conditions. Wheel speed to being back-propagated through the powertrain to calculate effective engine speed, opposed to using current engine speed directly, to circumvent the influence of torque converter speed ratio.

In the VES, the model depicted above is located directly downstream of the unaltered shift model and only becomes active during DFSO events. The same setup exists for imposed lockup.

As in scenario 1, the torque disturbance model for scenario 2 includes a PID that compensates for inaccuracies in the fitting. Since scenario 2 allows for shifts and changes in engine speed, the PID also helps the disturbance torque take into account changes in effective rotational mass and transmission efficiency as they affect velocity.
Since most DFSO situations occur in the higher gears (5 or 6 typically), it was observed that a single down shift resulted in appreciable gains without deviating too far from the baseline gear selection. This observation significantly reduced the complexity of implementing Figure 94 in script format. The following are the post-processing steps for Scenario 2, most of which are identical to those for Scenario 1.

**Step 1a:** Load feed forward data for DFSO phase, engine speed, PMEP, velocity, alpha-throttle, engine torque, gear position and transmission efficiency.

**Step 1b:** Extract data from each map during DFSO only.

**Step 2:** Calculate wheel speed based on vehicle velocity.

**Step 3:** Subtract one gear from gear selection for the duration of the DFSO event.

**Step 4:** Calculate the new engine speed by multiplying wheel speed by overall powertrain ratio taking into account the final drive, transfer ratio and a new gear ratio based on the downshift gear.

**Step 5:** Interpolate in the 1D Optimal PMEP vs. Engine Speed map to determine the lowest possible PMEP given engine speed. This step assumes that the throttle is moved to its optimal opening and PMEP will respond according to the surface fit.

**Step 6:** Determine \( \Delta \text{PMEP} \) as the difference between baseline PMEP and optimal PMEP.

**Step 7a:** Convert \( \Delta \text{PMEP} \) into an effective crankshaft disturbance torque.

**Step 7b:** Convert torque into power by multiplying by instantaneous engine speed.

**Assumptions:**

a) The original fit for \( \text{PMEP} = f(N_{\text{eng}}, \alpha) \) based on steady-state data is sufficient to determine PMEP during transients.
b) The calculation of disturbance torque is perfect and therefore will not affect the braking rate of the vehicle.

c) A single downshift will not violate the any maximum engine speed thresholds.

d) The torque converter is locked so the new engine speed can be calculated without knowledge of the TC slip ratio

Figure 95 shows the results of locking the torque converter and down shifting one gear during DFSO. These results are compared with the baseline strategy results in 4.3.2. It is important to note forcing lockup during DFSO contributes to increases in engine speed, because DFSO events occur in overrun operation where turbine speeds are higher than pump speeds if slip is allowed.
Notice that the disturbance torque is not very different between scenarios 1 and 2, but the resulting harvested power is higher for scenario 2. The minimal torque differences are due to the fact that Figure 88 is relatively flat at large throttle openings for the lower half of the speed range. In most cases actually, the torque for scenario 2 is slightly lower compared to scenario 2 because of the higher speeds involved, but those higher speeds are precisely what accounts for the increase in disturbance power. Figure 96 superimposes the FF gear selections and the improved selection during DFSO. The word "improved" is used purposefully to differentiate between the realistic single down shift strategy, and actual optimization of gear.
A fundamental assumption made in scenario 2 is that instantaneous changes in lockup status and commanded gear do not negatively affect NVH. This is of course a simplification, however adjustments can be made to provide a more realistic evaluation of the energy recovery. In order to successfully apply the energy recovery strategies discussed above to a real vehicle, it will be necessary to evaluate driver acceptance.
Scenario 3: Compensatory Load Considering Brake Reduction

Scenario 3 is fundamentally different from scenarios 1 and 2. Of the two methods of introducing an engine-level disturbance torque touched upon at the beginning of Section 4.3, scenario 3 deals with the second; a percentage of required stopping power during braking is transferred from the friction brakes to an ancillary load. This concept is no different from regenerative braking in hybrids where some of the braking energy is absorbed by electric machines. The methodology employed in scenario 3 is straightforward and can be applied for a variety of brake reduction percentages. Figure 97 depicts a simplified version of the total VES disturbance torque model. The only section added specifically for scenario 3 is the torque transfer subsystem which translates an allowable wheel-level brake reduction torque to the engine-level. The PID controller is included for the same reasons as for scenario 2, but it becomes more active due to the greater impact of transmission efficiency on disturbance torque.
The torque transfer subsystem contains an equation relating wheel-level to engine-level torques. $T_{\text{Brake\_align}}$ represents the amount of braking torque that may be removed from the friction brakes and transferred elsewhere. The goal of Equation (4.11) is calculate an engine-level torque that, when transmitted through the powertrain, will equal identically the torque responsibility removed from the brakes. (4.11) stems from a basic energy balance, but the location of the efficiency terms is important. During braking, a vehicle is in overrun operation meaning that power is flowing from the wheels to the engine. Even though power is being applied at the engine-level, the net flow of power is still negative meaning that the efficiencies must multiply the wheel level power. During overrun, the powertrain inefficiency is actually assisting a disturbance load. The transmission and torque converter efficiencies in (4.11) are those calculated during overrun as described in
the modeling chapter. Though torque converter efficiency is included in the equation, forced lockup during DFSO imposes 100%.

\[
T_{\text{transfer}} = \frac{T_{\text{Brake, \allow}} \omega_{\text{veh, \allow}} \eta_{\text{trans, \allow}} \eta_{\text{TC}}}{\omega_{\text{eng}}}
\]

where

\[
T_{\text{Brake, \allow}} = f_{\text{allow}} \left( \beta m_{\text{veh}} g \mu_{\text{frict, \allow}} r_{\text{whl}} \right)
\]

As one would expect, transferred torque is a very strong function of braking command \( \beta \).

In other words, a very accurate driver model is required to glean meaningful results. To ensure well conditioned results, the actual \( \beta \) trace from simulation, which was very oscillatory in nature, was displaced by a reasonable approximation. For the scenario 3 analysis, braking rate was assumed to be directly related to instantaneous acceleration:

\[
\beta = \frac{dV}{dt} \frac{1}{(dV/dt)_{\text{max}}}
\]

In (4.12), instantaneous acceleration is divided by the maximum acceleration observed over an entire FTP to normalize the \( \beta \) vector from 0 to 1, 1 corresponding to maximum negative acceleration. The following are the steps implemented in the post-processing script for Scenario 3.

**Step 1a:** Load feed forward data for DFSO phase, engine speed, velocity, Beta-brakes and transmission efficiency.

**Step 1b:** Extract data from each map during DFSO only.

**Step 2:** Create a normalized Beta vector by dividing instantaneous acceleration by the maximum negative acceleration observed for the FTP.
Step 3: Calculate wheel speed based on vehicle velocity.

Step 4: Calculated allowable brake torque reduction

Step 5: Transfer the brake torque reduction from the wheel-level to the engine-level by using an energy balance coupled with overrun efficiency.

Step 6: Convert engine-level disturbance torque into power by multiplying by instantaneous engine speed.

Assumptions:

a) The calculation of disturbance torque is perfect and therefore will not affect the braking rate of the vehicle.

b) The torque converter is locked so powertrain efficiency is purely a function of transmission efficiency

c) Brake torque transfer from the wheels to the engine is independent from Scenarios 1 and 2 so the results may be summed to obtain higher disturbance torque and power.

Coupling (4.11) with (4.12) yields Figure 98 for four different brake displacement percentages. The main purpose of Figure 98 is to show the variations in disturbance torque and power magnitudes over the entire FTP. A close-up of two DFSO events is shown after in Figure 99.
Figure 98: Scenario 3 disturbance torque and power

Figure 99: Scenario 3 disturbance torque and power (close-up of 2 DFSO events)
Brake displacement was only considered during DFSO events so that the results could be compared with scenarios 1 and 2. In reality, this concept could be expanded to all braking situations which would vastly increase the amount of energy that could be recovered.

The engine speeds used to calculate transfer torque, and subsequently disturbance power, are those found using the improved powertrain strategies from scenario 2. The engine speed profile from scenario 1 should yield the same power results however, since scenario 3 is based purely on an energy balance across the transmission. The only difference will originate from the transmission efficiency's functional dependence on engine speed.

4.3.2 Analysis and Comparison of DFSO Disturbance Torque Results

The three previous discussions detailed the disturbance torque and power calculations for each of the scenarios explored. This section will first compare some of the results graphically, and then the cycle analysis tool developed for the unconstrained FTP and Artemis analyses will be used to reveal energy related statistics. Figure 100 is a close-up of two DFSO events for which all six disturbance torques calculated are shown. Notice that for the first DFSO event, scenario 3 disturbance torque is zero for roughly the first three seconds. This is because DFSO was entered based on its most important entry condition, accelerator pedal, before the driver felt the need to actuate the brakes. For those first three seconds, the vehicle was coasting and the road loads were sufficient to slow down the vehicle.
As touched upon in the scenario 2 discussion, the benefit of downshifting is not to increase disturbance torque, but rather to decrease disturbance torque slightly to gain a net benefit in disturbance power. This is evident when comparing the scenario 1 and 2 traces in Figure 100 and Figure 101. One will also notice that scenario 3 disturbance torque is much more volatile compared to the others because of its directly relation to Beta. The saw tooth behavior displayed for the first two scenarios in Figure 101 is due a natural downshift corresponding to long fuel cut situations.
Figure 101: Comparison of disturbance power results

Table 18 and helps quantitatively understand the impact of the three scenarios and how said impact compares to more idealized energy recovery analyses. The numbers below have been calculated for the first 1350 seconds of the FTP. The last column in the Table 18 sums the results of the two most extreme realistic scenarios explored.
List of table inclusions:

1. Unconstrained analysis
2a. Constrained analysis \(P_{\text{lim}}=30\ kW, E_{\text{lim}}=300\ kJ\)
2b. 2a considering DFSO events only
3. Scenario 1 Base: Total positive energy required (7.6 MJ)
4. Scenario 2
5. Scenario 3 (20%)
6. Scenario 3 (80%)

Table 18: Overall comparison of energy recovery analyses (first 1350s of FTP, wheel-level)

<table>
<thead>
<tr>
<th>Analysis Number</th>
<th>Statistics</th>
<th>1</th>
<th>2a</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
<th>4&amp;6</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Number of Events</td>
<td>55</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Avg. Event Time [s]</td>
<td>9.3</td>
<td>4.8</td>
<td>4.1</td>
<td>4.8</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Total Braking Energy [kJ]</td>
<td>3.2E3</td>
<td>2.3E3</td>
<td>105</td>
<td>137</td>
<td>124</td>
<td>497</td>
<td>634</td>
</tr>
<tr>
<td></td>
<td>Total Braking Energy [Ah]</td>
<td>74</td>
<td>53</td>
<td>2.4</td>
<td>3.2</td>
<td>2.9</td>
<td>11.3</td>
<td>14.5</td>
</tr>
<tr>
<td></td>
<td>Normalized Total Braking Energy [%]</td>
<td>-</td>
<td>71.9</td>
<td>3.3</td>
<td>4.3</td>
<td>3.9</td>
<td>15.5</td>
<td>19.8</td>
</tr>
<tr>
<td></td>
<td>Max Braking Energy [kJ/event]</td>
<td>262</td>
<td>188</td>
<td>43.5</td>
<td>51.2</td>
<td>71.2</td>
<td>285</td>
<td>336</td>
</tr>
<tr>
<td></td>
<td>Avg. Braking Energy [kJ/event]</td>
<td>60</td>
<td>43</td>
<td>9.4</td>
<td>10.5</td>
<td>9.6</td>
<td>38.2</td>
<td>48.7</td>
</tr>
<tr>
<td></td>
<td>Max Braking Power [kW/event]</td>
<td>37</td>
<td>27</td>
<td>2.8</td>
<td>3.1</td>
<td>4.7</td>
<td>18.8</td>
<td>21.9</td>
</tr>
<tr>
<td></td>
<td>Avg. Max Braking Power [kW/event]</td>
<td>11.9</td>
<td>8.6</td>
<td>2.2</td>
<td>2.4</td>
<td>2.4</td>
<td>9.5</td>
<td>11.9</td>
</tr>
</tbody>
</table>

Note: Braking energy is equivalent in meaning to recoverable energy

Since analyses 4 and 6 are not mutually exclusive, they have been added together as a comparison between maximum realistic energy recovery during DFSO, and overall cycle available energy. This number is highlighted in yellow in Table 18. The row surrounded by a thick boarder gives the total braking energy available normalized by the maximum theoretical available which is the number highlighted in green. To more clearly compare the normalized energy recovered for the various scenarios, Figure 102 has been devised. The color of the number beneath each block represents the energy value by which that scenario's energy was normalized. For example, the green 24% below the blue block means that the recovered energy shown in the blue block is 24% of the energy recovered
In the green block. Only the 20% braking displacement for Scenario 3 has been considered because the higher percentages (40% and 80%) were deemed to be too unrealistic when considering a conventional vehicle without regenerative braking hardware. A 20% brake displacement will be assumed henceforth.

The blue 34.5% for Scenario 2+3 represents the effectiveness of the energy recovery methods explored at capturing the constrained energy available during DFSO. Considering that only 20% of the friction brake demand is being displaced to an engine-level ancillary load, a 34.5% effectiveness level is noteworthy. As would be expected, if one extends the results of the 20% brake displacement to a theoretical 100% displacement case, close to 100% of the energy available for recovery during DFSO could be captured.

By using a proportion between ideal positive energy required and simulated fuel consumption, the energy recovery values can be approximated as increases in fuel
economy. The assumption made here is that the energy recovered can be used in the vehicle directly to reduce the amount of energy required to drive the cycle.

\[
\frac{1/\text{MPG}_{\text{sim}}}{E_{\text{ideal}}} = \frac{1/\text{MPG}_{\text{approx}}}{E_{\text{ideal}} - E_{\text{recovered}}}
\]

By combining Scenarios 2 and 3, an approximate increase of 0.7 MPG in fuel economy could be realized. Although not specifically covered in detail within this document, Scenario 3 need not be constrained to DFSO events only. By extended Scenario 3 (20%) to all braking events and adding the results to those from Scenario 2 (DFSO events only), fuel economy could be improved by 1.8 MPG. The analysis presented herein was primarily developed for crankshaft energy harvesting in a conventional vehicle during DFSO, but the tools developed can be used to extend the analysis beyond the scope of this project. For example, considering 80% brake displacement and no PMEP-reduction, which effectively represents regenerative braking in a hybrid-electric vehicle, it has been estimated that 7.4 MPG, or a 37% increase in fuel economy from baseline could be achieved for the Chrysler Minivan. This result comes with many caveats however. 80% friction brake displacement would require significant hardware changes which would add to vehicle weight, decreasing the potential gains. Also, drivability concerns would disallow instantaneous 80% brake displacement during every braking event.

*Additional Observations*

1. The results indicate that analysis number 4 (20% brake transfer) is roughly equivalent to the PMEP-reduction-based disturbance torque case presented as Scenario 2.
2. Scenarios based on braking displacement have shorter average event times since the first part of DFSO events is typically spent in a coasting phase.

3. The power limit imposed for analysis 2 (30 kW) was never reached. A purposefully high energy limit was set, implying that the nearly 30% reduction in recoverable braking energy was due almost solely to transmission inefficiencies.

4. The theoretical maximum amount of recoverable energy for the first 1350 seconds of an FTP roughly corresponds to the total energy capacity of a typical automotive lead acid battery (~70 Ah).

5. The orange value in Table 18 indicates that roughly 20% of the theoretical energy available can be recovered through optimized throttle and powertrain strategies coupled with aggressive brake transfer.
Chapter 5: Conclusions and Future Work

Section 5.1 Conclusions

Regenerative braking in hybrid vehicles is employed to capture some of the kinetic energy typically lost as heat during braking. Crankshaft energy harvesting refers to the same concept as it pertains to conventional vehicles which do not contain the customary hardware necessary for regenerative braking. Instead, calculated management of engine-level ancillary loads coupled with optimal driveline control strategies can be used to capture some of the energy characteristically wasted during deceleration. Specifically, when the wheel-level power required by a vehicle is negative, opportunities for energy recuperation exist. The theoretical maximum amount of recoverable energy can be quantified for a given drive cycle by identifying the braking events and integrating the instantaneous negative power. The result of such an unconstrained analysis is the absolute maximum energy that can be recovered without affecting the vehicle velocity profile for that cycle (3.2 MJ for the FTP). This theoretical maximum serves as a uniform normalizing factor that can be derived for every drive cycle. In the specific cases of the analyses presented in this document, an important normalizing factor can be established by imposing some constraints on the power profile derived from a drive cycle. These constraints include power limits, energy limits, powertrain efficiency, and most importantly, the requirement that DFSO must be enabled. Combined, these
constraints can reduce the amount of recoverable energy significantly. If only DFSO events are considered, the available energy is reduced four fold. By normalizing the cumulated energies found for the various scenarios by this constrained benchmark, the energy capture *effectiveness* of each scenario can be quantified. Scenario 1 explored in this work attempted to add an engine-level disturbance torque (ancillary load) during DFSO based on an effective decrease in PMEP through optimal manipulation of throttle. Purely by opening the throttle during fuel cut situations, 1.4% of the positive cycle energy requirement can be captured. As an extension, Scenario 2 further increased the allowable disturbance load by locking the torque converter and shifting according to the instantaneous optimal engine speed corresponding to maximum harvesting potential. With these improvements, the 1.4% recovery for Scenario 1 was increased to 1.8%. Scenario 3 involved displacement of braking force from the friction brakes to an engine-level load. Scenarios 1&2 and Scenario 3 are not mutually exclusive and can be combined for increased gains. Assuming a 20% allowable brake displacement, 1.6% energy recovery is possible. By coupling optimal throttle and powertrain controls to minimize PMEP with 20% mild brake transfer, 3.4% of the positive cycle energy requirement and 34.5% of the maximum constrained energy could be theoretically recovered. If the recovered energy can be directly used to reduce the positive energy requirement during a drive cycle, 3.4% energy recovery roughly equates to a 0.7 MPG increase in fuel economy. A 1.8 MPG increase could result if brake displacement was not constrained to only DFSO events, but rather was extended to all braking situations. Naturally, the fuel economy benefits increase as brake displacement percentage increases.
Section 5.2 Future Work

1. To ensure that the VES calibration is robust enough to extend the model’s usefulness to drive cycles other than the FTP, validation must be performed on additional cycles (i.e. US06). Some calibration parameters may then need to be updated. An experimental Artemis Urban cycle is planned on the OSU chassis dynamometer so all the currently necessary feedforward maps (e.g. throttle) can be populated for validation on that cycle.

2. Examine feasibility of implementing additional energy storage capacity in the vehicle such as advanced batteries or ultracapacitors. Also consider the benefits of switching to a 48V electrical system with an improved alternator.

3. Continue the analysis started in Chapter 4 by taking the next step to characterize the energy sinks themselves. A detailed study of the current energy and power limitations of the systems present in the test vehicle will be required before a meaningful control strategy can be devised.

4. Begin formally describing optimization problems that are to be solved using a genetic algorithm routine running on the compute cluster located at CAR.
References


[36] Rizzoni, Giorgio. "Fluid Couplings." ME781 Lecture. The Ohio State University, Columbus, OH. Lecture.


Appendix A: Verification of Analytical Coast Down Solution

Mathematica 8.0 was used to perform the following steps. The "Simplify" command was used to simplify the summation of terms to the most reduced possible form.

\[
\frac{m}{dt} dV + CV^2 + BV = -A
\]

\[
\text{Term 1: } m \frac{dV}{dt} = \left( -B + \sqrt{-B^2 + 4AC} \right) \tan \left( \frac{1}{2} \left( -\frac{\sqrt{-B^2 + 4ACt}}{m} + c_1 \sqrt{-B^2 + 4AC} \right) \right)
\]

\[
V(t) = \frac{\left( -B + \sqrt{-B^2 + 4AC} \right) \tan \left( \frac{1}{2} \left( -\frac{\sqrt{-B^2 + 4ACt}}{m} + c_1 \sqrt{-B^2 + 4AC} \right) \right)}{2C}
\]

\[
\frac{dV}{dt} = \frac{(-B^2 + 4AC) \sec \left( \frac{1}{2} \left( -\frac{\sqrt{-B^2 + 4ACt}}{m} + c_1 \sqrt{-B^2 + 4AC} \right) \right)^2}{4Cm}
\]

\[
\text{Term 2: } CV^2 = \frac{(-B^2 + 4AC) \sec \left( \frac{1}{2} \left( -\frac{\sqrt{-B^2 + 4ACt}}{m} + c_1 \sqrt{-B^2 + 4AC} \right) \right)^2}{4C}
\]

\[
\text{Term 3: } BV = \frac{B \left( -B + \sqrt{-B^2 + 4AC} \right) \tan \left( \frac{1}{2} \left( -\frac{\sqrt{-B^2 + 4ACt}}{m} + c_1 \sqrt{-B^2 + 4AC} \right) \right)}{2C}
\]

\[
\text{Term 1 + Term 2 + Term 3} = -A
\]
Appendix B: Future Simulator Improvements

Improvements that need to be made to the VES are best summarized in list format. It is important to stress that the VES functions well as is, and the following changes are only necessary to improve the model.

1. Compile the VES so accelerated modes can be used in conjunction with genetic algorithm-based optimization routines.
2. Integrate "skip shift" logic in the transmission shift model. If necessary, convert the logic currently implemented in Simulink into a state flow diagram.
3. Employ velocity-dependent convection coefficients for the transmission case and catalyst can mass lumps.
4. Add neutral and park logic as optional Chrysler control blocks so experimental data can be better represented by the model. For example, certain drive cycles require a long idle soak period during which the transmission's state (i.e. free-wheeling or stationary) is important in terms of heat generation.
5. Introduce throttle-dependent shift and lockup durations. Currently they are both set as a constant 0.5 s.
6. Reintroduce/add driver subsystems to further improve the driver model. Specifically, add deadzones and better idle-speed controls.
7. Add masks to certain models to allow the flexibility of choosing between different components. For example, the VES will eventually need to be re-calibrated for 2.4L I4 engine, but it would be useful to retain a selectable version of the engine model based on the current 3.6L V6.

8. Model partial torque converter lockup

9. Add more conditions to shift model. This will require reverse-engineering of the TCU maps.

10. Add electrical starter model. This is important if engine stop/start is to be considered.

11. Increase the complexity of torque reduction during shifts. In an actual vehicle, this is accomplished through spark retard which cannot currently be resolved by the VES.

12. Fundamentally alter the driver model to output pedal instead of throttle. Doing so will eliminate the need for feed forward maps, but will require a detailed understanding of Chrysler’s throttle-by-wire system.

13. Explore FMEP dependency on alpha as it pertains the disturbance torque models.

14. Re-calculate optimal PMEP during DFSO base either on an improved fit, or on more variables.

15. Calculate RMS velocity error during disturbance torque activation to partially quantify drivability issues.