COMPRESSED AIR ENERGY STORAGE FOR VEHICLE START-STOP AND LAUNCH ASSIST

A Thesis

Submitted to the Graduate School
of the University of Notre Dame
in Partial Fulfillment of the Requirements
for the Degree of

Master of Science
in Mechanical Engineering

by

Andrew N. Balhoff,

James P. Schmiedeler, Director

Graduate Program in Aerospace and Mechanical Engineering
Notre Dame, Indiana
December 2012
Vehicle hybridization is an increasingly pursued strategy for improving fuel economy and lowering emissions. In an effort to provide a low cost and environmentally friendly system, compressed air energy storage (CAES) was pursued as an alternative to current hybrid systems. A driving simulator was created to predict areas for potential performance gains by use of the CAES system. Engine start-stop (ESS) and launch assist (LA) were considered as two potential methods for energy reintroduction. A pilot study was conducted over a range of vehicle speeds and traffic patterns that are provided for standard government testing purposes. Based on this study, the modeled CAES system was optimized to provide for necessary power flow and storage for the previously identified beneficial scenarios. Results show ESS benefits for schedules with numerous vehicle stops and LA benefits over all schedules considered.
To my parents, Michael and Frances
CONTENTS

FIGURES ................................................................. v

TABLES ................................................................. vi

ACKNOWLEDGMENTS .................................................... vii

CHAPTER 1: INTRODUCTION ........................................... 1
1.1 Motivation .......................................................... 1
1.2 Literature Review .................................................. 5
1.3 Objectives ........................................................... 9
1.4 Organization ......................................................... 9

CHAPTER 2: COMPRESSED AIR ENERGY STORAGE AND DRIVING SIMULATOR ........................................... 11
2.1 System Design and Experimental Apparatus ....................... 11
2.2 CAES Simulation ................................................... 15
2.3 Driving Simulator and Schedules ................................... 18
2.4 Summary ............................................................. 24

CHAPTER 3: ENERGY CYCLING INVESTIGATION ...................... 26
3.1 Introduction ......................................................... 26
3.2 Drivetrain Organization ............................................. 27
3.3 Start-stop ............................................................ 29
3.3.1 Advantages and Disadvantages .............................. 29
3.3.2 Implementation Requirements ............................... 32
3.3.3 System Performance ........................................... 34
3.4 Launch Assist ....................................................... 42
3.4.1 Advantages and Disadvantages .............................. 42
3.4.2 Implementation Requirements ............................... 44
3.4.3 System Performance ........................................... 46
3.5 Summary ............................................................. 49
<table>
<thead>
<tr>
<th>FIGURES</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>2.1 System Configuration</td>
<td>12</td>
</tr>
<tr>
<td>2.2 Cross sectional view of compressor/expander and rotary valve assembly. [41]</td>
<td>13</td>
</tr>
<tr>
<td>2.3 Compressor/Expander valve configuration [42]</td>
<td>14</td>
</tr>
<tr>
<td>2.4 Experimental System</td>
<td>15</td>
</tr>
<tr>
<td>2.5 Cruze MPG Changes with Payload Increases</td>
<td>25</td>
</tr>
<tr>
<td>3.1 Parallel Hybrid Drivetrain</td>
<td>28</td>
</tr>
<tr>
<td>3.2 Range Extender Drivetrain</td>
<td>29</td>
</tr>
<tr>
<td>3.3 Air Motor</td>
<td>34</td>
</tr>
<tr>
<td>3.4 Air Motor Torque and Power</td>
<td>35</td>
</tr>
<tr>
<td>3.5 Drive Schedules Color Coded by Drivetrain Requirement</td>
<td>37</td>
</tr>
<tr>
<td>3.6 Cruze Energy per ESS. Blue indicates braking energy. Red line indicates energy to start the engine. Note that this line is just above the axis.</td>
<td>38</td>
</tr>
<tr>
<td>3.7 Tahoe Energy per ESS. Blue indicates braking energy. Red line indicates energy to start the engine. Note that this line is just above the axis.</td>
<td>39</td>
</tr>
<tr>
<td>3.8 Cruze and Tahoe LA Fuel Economies. The blue represents the results from the simulated vehicle with no hybridization. The red represents the results from the simulated vehicle with the sized hybrid system implemented.</td>
<td>48</td>
</tr>
<tr>
<td>4.1 Ideal PV diagram</td>
<td>54</td>
</tr>
<tr>
<td>4.2 Valve timing diagram for compressor/expander</td>
<td>60</td>
</tr>
<tr>
<td>4.3 Mechanical efficiency of tuned valving in compression mode</td>
<td>63</td>
</tr>
<tr>
<td>4.4 Mechanical efficiency of tuned valving in expansion mode</td>
<td>64</td>
</tr>
<tr>
<td>4.5 Power of tuned valving in compression mode</td>
<td>64</td>
</tr>
<tr>
<td>4.6 Power of tuned valving in expansion mode</td>
<td>65</td>
</tr>
<tr>
<td>4.7 Mechanical efficiency of tuned valving in compression mode</td>
<td>67</td>
</tr>
<tr>
<td>4.8 Power of tuned valving in compression mode</td>
<td>67</td>
</tr>
<tr>
<td>4.9 Initially sized CAES air tank charge over NYCC cycle</td>
<td>69</td>
</tr>
<tr>
<td>4.10 Optimally sized CAES air tank charge over NYCC cycle</td>
<td>70</td>
</tr>
<tr>
<td>4.11 Optimized valving efficiency in compression mode</td>
<td>75</td>
</tr>
<tr>
<td>4.12 Optimized valving efficiency in expansion mode</td>
<td>75</td>
</tr>
</tbody>
</table>
TABLES

2.1 VEHICLE PARAMETERS [34] ........................................... 22
2.2 MPG VALIDATION (EPA MPG VALUES [5]) .......................... 22
2.3 ENERGY CONSUMPTION BREAKDOWN BY CYCLE FOR CRUZE 24
2.4 ENERGY CONSUMPTION BREAKDOWN BY CYCLE FOR TAHOE 24

3.1 IDLE TIME PER SCHEDULE ................................. 30
3.2 VEHICLE PRICE COMPARISON BETWEEN HYBRID AND NON-HYBRID TRIM .............................................. 31
3.3 FUEL ECONOMY IMPROVEMENTS DUE TO ESS ................. 41
3.4 MODEL PRICE AND FUEL ECONOMY COMPARISONS .......... 43
3.5 AVAILABLE DISSIPATED POWER - CRUZE ......................... 45
3.6 AVAILABLE DISSIPATED POWER - TAHOE ......................... 45
3.7 VEHICLE POWER DEMANDS - CRUZE ............................. 46
3.8 VEHICLE POWER DEMANDS - TAHOE ............................. 46
3.9 AVERAGE DISCHARGE TIME PER ACCELERATION/BRAKING CYCLE .................................................. 47

4.1 FMINCON PARAMETER BOUNDS .............................. 61
4.2 AVERAGE DISCHARGE TIME PER ACCELERATION CYCLE ABOVE ENGINE OPTIMIZATION THRESHOLD ............... 71
4.3 ESTIMATED BSFC SCALING FACTOR ............................ 73
4.4 LAUNCH ASSIST UPDATED MPG ESTIMATES FOR CRUZE .... 76
4.5 LAUNCH ASSIST UPDATED MPG ESTIMATES FOR TAHOE .... 76
ACKNOWLEDGMENTS

I thank my adviser, James Schmiedeler, for his guidance, support, and patience which made this work possible. His intellectual rigor and enthusiasm serves as inspiration to me.
CHAPTER 1

INTRODUCTION

1.1 Motivation

Internal combustion engines (ICEs) almost universally serve as the primary power source for modern road-based transportation. This is true despite their inefficient energy conversion from fuel to mechanical energy when operated over the wide range of environmental conditions encountered by vehicles. A thermodynamically ideal ICE can convert approximately 45% of the chemical energy of fuel into mechanical energy. However, while operating over the range required during day-to-day driving, real ICE implementations only achieve half that efficiency. Additionally, they have no way of recapturing or storing the energy they produce once the fuel source is combusted. Despite these issues, ICEs are regularly implemented due to their fuel’s high energy density, as well as the flexibility of recharging their fuel source.

Alternatives to ICE-only powered vehicles have been pursued since before the turn of the 20th century with the goal of addressing ICE shortcomings. Attempts have been made to replace the ICE wholly or to use an auxiliary system to augment the ICE. These attempts have been met with limited success, largely due to the monetary investment necessary to realize significant efficiency gains. However, with rising fuel prices as well as increased governmental pressure on car manufacturers...
to produce higher efficiency vehicles, hybrid systems research has increased over the past few decades.

The colloquial “hybrid vehicle” refers to two different energy pathway configurations. One category of vehicles is composed of true hybrid vehicles, which employ an alternate energy delivery system in conjunction with an ICE. The other category of “hybrid” operates solely on an alternative energy system, without an ICE. True hybrids on the market today include the Toyota Prius and the Ford Fusion Hybrid. This type of vehicle typically uses an electric energy storage system. Attempts to use compressed air, flywheels, thermal energy, and springs have all failed to realize significant market penetration.

Hybrid vehicles have great flexibility as to their reliance on the alternate energy delivery system, which defines the split into weak and strong hybrids. Weak hybrid vehicles rely heavily on the ICE for driving power and use the alternate system as a light supplement. These weak systems often are used for engine start-stop (ESS) applications and sometimes also for light launch assist (LA). ESS works by stopping the engine when the vehicle is stopped and restarting it when the engine is required to drive the vehicle. This system saves the wasted fuel used during idle, a problem in city driving and stop-and-go traffic. LA works by propelling the car forward, either to assist the engine or without ICE assistance. Strong hybrids operate by relying heavily on a large alternate power delivery system, often with the capability of propelling the vehicle with no engine assistance. These strong hybrid vehicles almost universally employ both LA and ESS.

Pure alternative energy vehicles have been attempted in a variety of configurations using electricity, solar power, and hydrogen, to name a few. A modern day, commercial market example of an alternative energy vehicle is the Nissan Leaf, which is powered by electric motors using energy stored in a battery. These vehicles have
had mixed success and have been plagued by issues with both short range compared to ICE competitors and lack of fuel availability. Both true hybrids and alternative energy systems attempt to decrease fuel consumption, lower harmful emissions, and recapture wasted energy, addressing key issues with ICE-powered vehicles. While benefits of alternative energy systems do exist, ICE-coupled hybrid vehicles have seen more significant success in the consumer market.

Battery electric hybrid vehicles have gained prevalence as the hybrid energy storage system of choice compared to other hybrid systems. Battery electric hybrids benefit from high round trip efficiency in energy storage and reintroduction compared to many mechanical systems and are capable of providing large torques to assist the ICE, a problem with many mechanical hybrid systems. However, these battery systems are not without downsides. Battery hybrids incur significant additional expenses over the entirety of their life cycles. Initially, these hybrid vehicles cost the consumer between $2000 and $10000 dollars extra when compared to their non-hybrid equivalents. While a portion of this is often offset by government tax credits, the cost for the consumer is still significant. Additionally, there are costs associated with the training and safety necessary for technicians to service the vehicles. Finally, there are increased costs associated with the proper disposal of the batteries to ensure that their harmful rare earth metals are not introduced into the environment.

The batteries used in hybrid vehicles all use a variety of rare earth metal. These materials are largely imported to the United States. In addition to batteries, the demand for rare earth minerals extends to all electronics which drives up cost and induces scarcity. The nation’s dependence on foreign suppliers could prove disadvantageous should hostile international relations arise. According to an EPA report [35] on mining of rare earth elements, every ton of rare earth element produces
8.5 kilograms of fluorine and 13 kilograms of flue dust. During the production and refinement of these metals, an additional 8,600 to 12,000 cubic meters of gas containing sulfuric acid, hydrofluoric acid, flue dust, and sulfur dioxide are produced. Additionally, approximately 75 cubic meters of acidic waste water and a ton of radioactive waste residue are produced per ton of rare earth element. The disposal of these materials involves either costly recycling or placement in specially lined landfills.

Non-battery hybrid systems are capable of addressing some of the issues with battery electric hybrid systems. One of these in particular is the Compressed Air Energy Storage (CAES) hybrid. Compressed air for energy storage has been explored since the 1800s, with early attempts to power railways [12]. However, issues with sealing and other technical challenges prevented success. Implementation to provide power for other types of vehicles has also been pursued and is currently implemented successfully in volatile environments such as for mining applications [27]. However, the low energy density of air has limited attempts to implement air-only vehicles in commercial markets. Additionally, the round trip efficiency of a CAES system is poor relative to a battery energy storage system when taking into account the compression and expansion processes. Creutzig et al. investigated issues of cycle efficiency and performance of a compressed air vehicle [17]. They identified the losses present due to rapid charging, but indicated that potential gains could be achieved through heat addition to the compressed air. With this modification, the CAES system could be competitive with battery electric hybrids. This stemmed from the low cost for the CAES hybrid system and its ability to more rapidly earn its initial investment than comparable battery hybrid systems. In addition to its low cost, CAES also has a low environmental impact, as the energy storage medium does not have harmful byproducts, as in battery hybrid systems.
1.2 Literature Review

Most work concerning CAES vehicles has focused on some hybridization rather than a pure CAES configuration. This is largely due to the low energy density and efficiency of such systems, which lead to large air storage tanks. Work concerning an open CAES system integrated with an ICE was conducted by Schechter et al. [39], [40]. These systems incorporate a variable valve timing (VVT) approach to use the ICE as the compressor capable of operating over a wide pressure range through alterations to the engine valve profiles. Additional work by Ivanco et al. [8] looked to optimization techniques to control the valve timing intelligently for better efficiency. The systems investigated in these papers use complicated active valve systems that are costly to implement. In order to provide some of the same benefits, Brown [41] investigated implementation of a rotary valve. This approach provided a low cost solution and permitted more efficient operation over a large pressure range than a fixed timing system. A rotary valve assembly was experimentally validated to be capable of providing the necessary valve timing at a fraction of the cost of a traditional VVT system.

In order to model vehicle performance over a drive schedule, Markel investigated driving simulator design [43]. He categorized simulators in two general categories – forward and backward simulators. The downsides of the forward simulator, including the more intensive iterative process and lack of comparability with public vehicle data, led Markel to the design of a backward simulator. The backward simulator does not provide the potential for active human input; however, this functionality is more desired in physical drive simulator units rather than for system modeling. Work to minimize drive simulation computational effort was conducted by Froberg [18]. Similarly to Markel, Froberg determined that a backward simulation was necessary for expedient schedule analysis, especially for detailed engine
dynamics modeling. Montazeri-Gh and Nagizadeh [32] employed drive simulation analysis to study partitioning of drive cycles by energy flow for use by hybrid vehicles. They used this partitioning to analyze drive schedules and to note potential areas for improvement with hybrid systems over ICE-only operation. In order to easily compare vehicle performance to current, publicly available values, work done by Gillespie [20] provided basic equations to model vehicle dynamics. Gillespie used standard EPA schedules, and city and highway fuel economies were generated for comparison [4]. Once the driving simulator was executed using these schedules, verified standard vehicle performance values were obtained from the EPA Office of Transportation and Air Quality [5]. Gillespie determined that the error of the simple simulation was often only marginally worse than more complex and computationally costly simulations.

Fellini et al. investigated optimal design of hybrid powertrain systems, comparing hybrid vehicle drivetrain configurations [36]. Sensitivity and optimization of the systems over a wide range of cycles compared derivative-based and derivative-free optimization techniques for tuning an appropriate hybrid system. The derivative-free techniques avoid local minima, but they converge much more slowly. Therefore, a derivative-based approach allowing for rapid convergence is desirable with appropriate initial conditions to speed optimization. Ciccarelli and Toossi [13] highlighted the benefits of a power assist parallel hybrid for the purposes of city driving, as it offers the chance to use the ICE to power the vehicle while affecting efficiency by the hybrid system changing engine load. Karbowski et al. [15] then performed a comparison of powertrain configurations for hybrid vehicles, specifically contrasting systems that funnel all power through the hybrid system and those that operate with the hybrid system and ICE in parallel to drive the vehicle. They studied the capability to meet demands on performance and the hybrid storage system’s state
of charge (how much energy was stored in the energy reservoir). In addition, the issue of hybrid power storage sizing was addressed, with both gradient-based and dynamic programming algorithms proposed to appropriately size a system based on drive schedule requirements. It was determined that, similar to previous research, the optimal powertrain design was dependent on vehicle trip optimization, with smaller storage capacity for city or low acceleration cycles and larger capacity for highway and high acceleration cycles. They also noted the benefits of using a parallel design with a smaller hybrid for increased driveability and reducing the direct load on the hybrid system. They noted that this would allow for occasional use of the hybrid system for purposes such as ESS, while not requiring the system to provide constant LA capability.

Research into powertrain design by Aziz et al. studied series, parallel, and power split hybrids [7]. Series hybrids operate by using a single path for energy flow, often from the ICE to the hybrid system to the drivetrain. This is in contrast to parallel and power split hybrids which allow for both power plants to act on the drivetrain. Aziz identified a power split hybrid drive train as an improved solution over the series and parallel drive trains, as the engine operation is not directly coupled with the hybrid system as it is in the series case. This allows for the engine to power the vehicle forward without having to deliver energy through the hybrid system, where the power is decreased through efficiency losses in the hybrid system. However, the engine in a power split hybrid can operate and power the car while simultaneously charging the hybrid storage system, providing simple engine load optimization to increase fuel economy. This is in contrast to the parallel hybrid which lacks the ability to both power the car and charge the hybrid system. The power split system allows for energy to be stored in the hybrid system more readily and for improved launch assist performance.
Work done by Bishop et al. [22] studied the benefits and challenges of an ESS system. The paper successfully justified that a weak hybrid system not capable of providing launch assist could still provide fuel economy benefits to a vehicle. Launch assist hybrids, as noted by Karbowski [15], tend to be larger in size, with increased fuel economy benefits over a simple ESS system. An et al. addressed the hybrid options for light duty vehicles and noted the efficiency gains of using regenerative braking and engine optimization for ESS and LA [16]. They found only marginal benefits of using ESS alone, with greater fuel economy increases using both ESS and LA. Miller addressed the potential benefits of ESS and LA systems [31], computing a potential increase in fuel economy of 15% for ESS technology and a 91% increase for LA. However, he does note the increased complexity and weight required to implement the LA system.

Controlling adequate power delivery and cycling are well documented issues in hybrids. Paganelli et al. addressed a two-energy-source hybrid vehicle and the optimization necessary for minimum power delivery [19]. They found a potential 17.5% improvement in fuel economy with a weak hybrid system in launch assist. Phillips et al. compared hybrid optimization for electric vehicles and found similar results using a dynamic controller to control two subcontrollers, one for each power system [9]. Delprat approached the issue of control by using dynamic programming and heuristic controllers, finding control of a system over drive cycles challenging due to large computational requirements associated with dynamic programming [37]. Jalil [33] addressed issues associated with subsystem efficiency rather than simply targeting increased fuel economy and included optimization of the power subsystems for a series hybrid. This work additionally addressed the issue of state of charge, which is required by law to stay within an acceptable range. This is important because without such safeguards, the hybrid system could run dangerously low and poten-
tially leave the vehicle stranded. Won addressed a similar issue with parallel hybrids and used a vehicle-mode-based compensator to ensure appropriately sustained charge [23]. Here, Won analyzed the complex nonlinear problem and controlled it with a single linear optimization formulation. Liu and Peng addressed the issues of control of the separate systems subject to multiple constraints and found a number of control schemes to be successful, especially dynamic programming [28]. However, the potential benefits of this system could be offset by the large computational cost, as mentioned by Delprat [37].

Work done by Kang et al. investigated energy recovery with compressed air systems interacting with ICEs [21]. Valve timing optimization was conducted with the valve ported to the engine cylinder. Alternatively, Huang et al. used a pneumatic power system with ICEs to load the engine more nearly optimally [26], [24]. This system was successful in providing high efficiency, but had issues with regulating power output over cycles due to challenges with control of the compressed air.

1.3 Objectives

The purpose of this work is to investigate powertrain integration of a CAES system with a modern ICE-powered vehicle. The objectives are: 1) to create a valid simulation of vehicular performance for CAES optimization, 2) to study powertrain integration and potential methods for energy reintroduction, and 3) to optimize Brown’s [42] system for implementation in a consumer vehicle over a variety of drive schedules.

1.4 Organization

The remainder of the thesis is organized as follows. Chapter 2 provides a detailed explanation of Brown’s work which was used as a basis for the computer modeling
of the physical CAES system. It then transitions to a description of the dynamic vehicle driving simulator developed in the present work to provide a real world performance estimation of the CAES system, as well as how this system works with Brown’s model. Chapter 3 investigates ESS and LA power delivery systems available for hybrid vehicle operation over a variety of drive schedules using Brown’s model and conducts a pilot study to investigate potential gains to be made through system optimization. Additionally, the chapter explores possibilities for energy recapture and their effects on the system capability. Initially, ESS is investigated for its simple implementation requirements. Next, LA capability is pursued for its greater potential for use over all driving schedules. Chapter 4 also investigates how changing CAES parameters alters the compression and expansion capabilities of the system. It then uses this information to tune the CAES system to maximize returns over a variety of EPA driving schedules keeping in mind rules set in Chapter 3. Chapter 4 investigates the potential for optimizing the valve assembly and system parameters for Brown’s system to fit the drive schedule needs as described in Chapter 2. Chapter 5 concludes the thesis and details future work to be conducted based on the solutions obtained as well as insights into the mechanical system.
The compressed air energy storage system considered here consists of a compressor/expander and an air tank that provide a single path for energy conversion and storage. The system will be briefly described here, with greater detail available in Brown’s article [42]. The component parts were designed and physically assembled by Brown and used for validation of his computer models describing the system. This physical validation lent credibility to optimizations of the valve assembly that were performed using the computer models in order to increase the CAES system’s efficiency. A driving simulator was created by the author to better map the CAES system performance over the EPA drive cycles used to compare hybrid performance.

2.1 System Design and Experimental Apparatus

The experimental system can be broken down into three main parts: an IVT, the compressor/expander, and an air tank. A basic overview of the design can be seen in Figure 2.1, where the driveshaft of the IVT is connected to the vehicle on the left of the diagram to handle energy flow and the mechanical energy is transmitted through it to the compressor/expander and finally into the tank as compressed air.

The compressor, a Champion B1 compressor pump with a 2.375-inch bore and 2-inch stroke, was modified to handle operation as both a compressor and an expander.
This added functionality lightens the system by eliminating the need for separate mechanisms for fluid inflow and outflow.

However, this modification lowered the efficiency of the original compressor, an issue addressed later. The compressor/expander was also altered to accommodate a rotary valve, the piece necessary to allow for the multidirectional flow. This rotary valve provides the fixed valve timing, which becomes an issue in the expander case while operating at different tank pressures. The rotary valve is a low cost, simple solution in contrast to VVT and variable compression ratio systems. The rotary valve was machined out of a 1 inch diameter steel shaft, with ports cut through to accommodate for both expansion and compression. The high side port was designed with a width of 0.4 inches and a length of 0.2 inches. The corresponding valve hole also has a length of 0.2 inches. The low side port has a width of 0.2945 inches and a length of 0.375 inches. The corresponding valve hole is also 0.375 inches. The center
of the high side sweep occurs at 22 degrees during the rotation, and the low side center occurs at 134 degrees. A diagram of the rotary valve can be seen in Figure 2.2.

![Figure 2.2: Cross sectional view of compressor/expander and rotary valve assembly.](image)

The rotary valve is assisted by reed valves on both the high and low sides to increase efficiency below design pressure. Figure 2.3 details the valve pathways for airflow, showing two rotary valves instead of a single one for clarity of those paths. The reed valves with flow hole diameters of 0.375 are passively open at 100 psi. When the system is in compression, the air flows from the compressor to the air tank. This tank was designed with an 11 gallon capacity and is safe to operate to 125 psi. This tank was chosen by Brown to allow for inexpensive and safe low pressure system validation.

The CAES system was designed with an IVT to meter energy in and out of the
Figure 2.3: Compressor/Expander valve configuration [42].

system through a single mechanism. The IVT envisioned mates an electronically actuated CVT with a planetary gearbox. This would allow for gearing changes and power flow direction to be handled between the vehicle drive train and the compressor/expander. However, this IVT is not in place in the experimental setup, but rather an electrical system is used for handling the energy flow. This electric system provides a simple, single path for energy flow in and out of the system, while allowing for straightforward measuring of that energy flow through voltage and current monitoring. This system consists of a high current power supply that powers a 1 HP DC motor to provide the input energy for the compressor. The energy outflow during expansion is handled by dissipating the load through five 20 Ω load resistors.

The electronic system is controlled by an on-board computer, which actuates a servo drive to control the DC motor. The computer also monitors the pressure and
temperature of the tank, as well as the shaft position and shaft speed. A picture of the experimental apparatus is shown in Figure 2.4.

![Figure 2.4: Experimental System.](image)

The benchtop CAES system weight is approximately 130 pounds. The base compressor weighs 31 pounds, and approximately 40 with the rotary valve head. The air tank weighs 18 pounds, and the CVT weighs approximately 30 pounds. The planetary gearbox and the system infrastructure such as piping and sensors were conservatively estimated to take up another 30 pounds. This system estimate allowed for payload adjustments later in the drive simulator.

2.2 CAES Simulation

Brown created a computer model of the CAES system using MATLAB to allow for accurate simulation of system performance. The code determines whether the CAES unit is compressing air, expanding air, or not operating and then calls a
sub-script to calculate the position of the compressor. From this information, the main script advances the compressor and calculates various properties of the air pushed into the air tank, as well as updates a number of tank parameters. The model updates and records the energy and temperature changes of the tank and the volume of air in it. The total mass flow at each time step is calculated, taking into account the possibility of choked flow, which occurs when the pressure ratio between the compressor chamber and the tank pressure is greater than 1.89. Additionally, MATLAB functions handle compressor, rotary valve, and reed valve geometries specific to the model.

Mass flow is calculated at each time step. The flow was modeled using orifice equations capable of handling choked and non-choked equations. Standard non-choked flow was modeled using

\[ \dot{M} = A Y C \sqrt{2 \rho (P_{\text{high}} - P_{\text{low}})}, \quad (2.1) \]

where \( A \) is the area of an orifice, \( Y \) is the expansion factor, \( C \) is the discharge coefficient, \( \rho \) is the upstream air density, \( P_{\text{high}} \) is the upstream pressure, and \( P_{\text{low}} \) is the downstream pressure. The expansion factor is given by

\[ Y = \sqrt{r^{2/k} \left( \frac{k}{k-1} \right) \left( 1 - r^{(k-1)/k} \right) \frac{1 - \beta^4}{1 - \beta^4 r^{2/k}}}, \quad (2.2) \]

where \( r = P_{\text{high}}/P_{\text{low}} \), \( k \) is the specific heat ratio, and \( \beta \) is the discharge coefficient. Here, the discharge coefficient was ignored due to the complex geometry present in the valve flow channels. In the case of choked flow,

\[ \dot{M} = A C \sqrt{k \rho P_{\text{high}} \left( \frac{2}{k+1} \right)^{\frac{k+1}{k-1}}}, \quad (2.3) \]

was used.
The piston work was calculated by integrating

$$\dot{W} = \dot{V} P_c,$$  \hspace{1cm} (2.4)

where \( V \) is the volume of the cylinder and \( P_c \) is the cylinder pressure. The thermal conduction by the piston mechanism was calculated by

$$\dot{Q} = -\kappa A (T_c - T_{metal}),$$  \hspace{1cm} (2.5)

where \( \kappa \) is the thermal conductivity coefficient, \( A \) is the exposed area, \( T_c \) is the temperature in the cylinder, and \( T_{metal} \) is the temperature of the piston chamber walls. The change in energy of the system was then calculated, with the energy flow due to mass exchange between the cylinder and the tank computed by

$$\Delta E_{flow} = \begin{cases} 
  h_c \dot{M}_{high} & \text{if } \dot{M}_{high} < 0 \\
  h_t \dot{M}_{high} & \text{if } \dot{M}_{high} > 0.
\end{cases}$$  \hspace{1cm} (2.6)

The energy change for mass flow between the cylinder and the atmosphere was calculated by

$$\Delta E_{flow} = \begin{cases} 
  h_c \dot{M}_{low} & \text{if } \dot{M}_{low} < 0 \\
  h_0 \dot{M}_{low} & \text{if } \dot{M}_{low} > 0.
\end{cases}$$  \hspace{1cm} (2.7)

Therefore, the total energy change for the control volume was calculated by

$$\frac{dE}{dt} = \dot{W} - \dot{Q} + \Delta E_{flow}.$$  \hspace{1cm} (2.8)

In order to determine the rates of change of pressure and temperature, the rate of change per mass of internal energy was determined. The total energy was calculated as equal to the internal energy. The rate of temperature change was calculated for
ideal gases using $cv$, the specific heat of air at constant volume.

$$\dot{T} = \frac{du}{dt}/cv. \quad (2.9)$$

Thus, the rate of change of pressure was found by differentiating the ideal gas law with respect to time.

$$\dot{P} = \frac{R(T\dot{M} + M\dot{T})}{V} - \frac{P}{V}\dot{V}. \quad (2.10)$$

This set of differential equations was then integrated using 4th order Runge Kutta to advance the system variables through time. This was done using the inputs of the pressure and temperature differential between the tank and the atmosphere and the position of the compressor crank at the timestep. The equations returned the energy, temperature, and pressure rate of change of the tank.

2.3 Driving Simulator and Schedules

A driving simulator was required to model the CAES system in a useful way for implementation over a drive schedule. The simulator studied vehicle motion by calculating the vehicle’s energy demands and integrated the CAES system with the engine power plant to meet those needs. To produce results easily comparable to industry standards, the program was designed to follow EPA-specified drive schedules which provide speeds for the vehicle to match. This is in contrast to throttle-based designs that take input of throttle and output the velocity. While a program such as this would readily simulate a realistic driving experience, results would not be repeatable due to dependency on the “driver.” In addition, useful comparison to current vehicles would be impossible because such comparisons are standardized across the EPA schedules. The program, therefore, operated in an inverse manner, taking the desired velocity and returning the energy input necessary to drive the vehicle as provided by the engine. The EPA profile design was chosen due to its
ability to mimic dynamometer testing, the industry comparison method.

The EPA began issuing drive schedules to fulfill requirements of the Energy Tax Act of 1978. The act was passed to promote fuel efficiency and alternative energy sources. One portion of the act created the gas guzzler tax, which taxes cars with low fuel economy. In order to standardize the application of this tax, the EPA mandated standard highway and city driving schedules. Over the years, these driving schedules have been updated to better account for driving conditions. These cycles cover multiple city and highway profiles and test for a variety of factors, such as air conditioning use, vehicle size, and aggressive driving. From the selection of profiles, four were chosen to cover a range of possibilities. The UDDS, or urban dynamometer driving schedule, was selected for its common use simulating city driving conditions. The HWFET, or highway fuel economy driving schedule, was chosen to simulate highway driving under 60 mph. The NYCC, or New York City cycle, was chosen for its use in low speed stop-and-go traffic. The US06 is a high acceleration aggressive driving schedule. These four schedules allow for scenarios of high braking and acceleration, as well as moderate breaking and acceleration. By studying these four, high and low bounds of the energy requirements for CAES implementation can be found and compared to publicly available data for almost any vehicle.

EPA driving schedules are provided as dynamometer-based profiles, where the car is required to match provided speeds under controlled conditions and then the fuel economy is recorded upon completion of the cycle. The speeds are listed at 1 Hertz and publicly available at the EPA website. In order to calculate the fuel economy of a vehicle, first the acceleration and velocity at each timestep were computed.

$$a = \frac{(V_i - V_{i-1})}{t} \quad (2.11)$$
Here \( a \) is the time averaged acceleration, \( V_i \) is the velocity at the \( i \)th timestep, and \( t \) is the time. From these values, the force at each timestep necessary to move the vehicle at the desired speed was computed. This was done by summing the force required to overcome the rolling resistance of the vehicle, \( F_{RR} \), the force to overcome the aerodynamic forces on the vehicle, \( F_A \), and the force required to overcome the inertia of the drivetrain, \( F_I \), as seen in Equations 2.13, 2.14, 2.15, and 2.16.

\[
V = \frac{(V_i + V_{i-1})}{2}. \tag{2.12}
\]

\[
F_{tot} = F_{RR} + F_A + F_I \tag{2.13}
\]

\[
F_{RR} = m g C_{RR} \tag{2.14}
\]

\[
F_A = \frac{1}{2} \rho C_D A_F (V)^2 \tag{2.15}
\]

\[
F_I = m r a + m g \sin(\theta) \tag{2.16}
\]

\[
P_{tractive} = F_{tot} V. \tag{2.17}
\]

Here, \( C_{RR} \) is the coefficient of rolling resistance of the tires, \( g \) is gravity, \( m \) is the mass of the vehicle, \( C_D \) is the coefficient of drag of the vehicle, \( \rho \) is the density of air, \( A_F \) is the frontal area of the vehicle, \( \theta \) is the slope of the road, \( r \) is the rotational inertia compensation factor, and \( P_{tractive} \) is the tractive power. The rotational inertia compensation factor added imaginary weight to the vehicle in order to approximate the effects of the rotating vehicle components. This was set to be 1.03. The value of \( \rho \) was set to be \( 1.2 \frac{kg}{m^3} \). These equations were based on simplified free body diagrams of vehicles from Miller [31].

Based on these equations, five power requirement cases were considered: acceleration, cruising, coasting, powered deceleration, and braking. Acceleration consisted of \( F_{tot} \) meeting the demands of the aerodynamic force, the rolling resistance, and
the force required to overcome the inertia of the vehicle. Cruising occurred when velocity was constant and therefore $F_I = 0$. Coasting occurred when $F_{RR} + F_A = -F_I$ and no input from the drivetrain was necessary. The car in this case is decelerating due to the lack of force input. Powered deceleration occurred when $F_I < 0$, but engine input was still necessary to maintain the speed required by the driving schedule. Finally, braking was considered where external force was required to slow the car to match the driving schedule. These five cases were then grouped into three categories of braking, powered deceleration, and engine input required. By grouping the possible cases in this way, the drive schedule can be organized by energy demand.

Once the tractive power required was calculated, the total energy required for the drive cycle was calculated by analyzing the total force required over the cycle,

$$E_{tot} = \sum P_{tractive} \, dt,$$

where $E_{tot}$ is the total energy required in Joules. Then based on the heating value of the fuel used, the MPG could be calculated

$$MPG = \frac{D}{E_{tot}} \frac{hv \, e_{eng} \, e_{dt}}{BTU}.$$ 

Here, $D$ refers to the distance traveled, $hv$ is the heating value of the fuel in $BTU/\text{GAL}$, $e_{eng}$ is the engine efficiency, and $e_{dt}$ is the drivetrain efficiency. The fuel heating value was set to 114132 $BTU/\text{GAL}$.

Calculating the $MPG$ values allowed for validation of the drive simulator by studying the values produced and comparing them to EPA-provided values. Five vehicles were compared, the Chevrolet Tahoe, Silverado, Malibu, and Cruze, as well as the Cadillac Escalade. These vehicles were chosen to span a range of vehicle types, including SUVs, the Escalade and Tahoe, sedans, the Malibu and Cruze, and pickup
trucks, the Silverado. A table of the parameters necessary to model each vehicle is listed in Table 2.1. In this table, $C_D$ is the coefficient of drag, $A_F$ is the frontal area, $D_W$ is the wheel diameter, and $e_{eng}$ is the engine efficiency. Additionally, these vehicles were chosen for comparison with their production hybrid versions. A comparison of the simulator’s estimated values and EPA-provided values in MPG can be seen in Table 2.2.

### TABLE 2.1

**VEHICLE PARAMETERS [34]**

<table>
<thead>
<tr>
<th></th>
<th>Mass [kg]</th>
<th>$C_D$</th>
<th>$A_F$ [m$^2$]</th>
<th>$D_W$ [m]</th>
<th>$e_{eng}$</th>
<th>$e_{dr}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tahoe</td>
<td>2530.5</td>
<td>0.39</td>
<td>3.4556</td>
<td>0.457</td>
<td>0.34</td>
<td>0.78</td>
</tr>
<tr>
<td>Silverado</td>
<td>2427.7</td>
<td>0.41</td>
<td>3.3445</td>
<td>0.457</td>
<td>0.32</td>
<td>0.77</td>
</tr>
<tr>
<td>Escalade</td>
<td>2520.5</td>
<td>0.36</td>
<td>3.4071</td>
<td>0.457</td>
<td>0.28</td>
<td>0.78</td>
</tr>
<tr>
<td>Malibu</td>
<td>1560.0</td>
<td>0.29</td>
<td>2.5898</td>
<td>0.432</td>
<td>0.32</td>
<td>0.82</td>
</tr>
<tr>
<td>Cruze</td>
<td>1410.0</td>
<td>0.29</td>
<td>2.6501</td>
<td>0.406</td>
<td>0.34</td>
<td>0.82</td>
</tr>
</tbody>
</table>

With the vehicle engine and drivetrain efficiencies tuned, the errors are relatively small, with the largest errors occurring in city driving cases with larger veh-
cles. These are likely due to overestimations of drivetrain efficiencies necessary to match highway values. The small errors present, especially in the case of the sedans modeled, lent credibility to the simulator for further implementation of the CAES system model.

The benefit of the driving simulator is the amount of information generated to aid in analysis. The simulator returns the net power dedicated to meet the aerodynamic, rolling and inertia force at each time step, as well as the amount of energy lost in each category. It calculates the total amount of force at each timestep, allowing study of the wheel demands during the drive schedule. It also calculates both the instantaneous and estimated average for the cycle. All of this information leads to more productive focus on cycles with potentially recapturable energy losses. The cycle energy losses due to aerodynamic forces cannot be stored, but the energy losses due to braking provide potential for improvement.

The cycles were analyzed for percent losses based on aerodynamic drag energy, rolling resistance energy, and braking. Two vehicles were compared, the Cruze and the Tahoe, with individual results presented in Tables 2.3 and 2.4. Based on braking losses, the UDDS and NYCC cycles offer significant chances for improvements in both vehicles, while the US06 schedule also offers some potential benefits, and the HWY cycle offers little room for improvement. Additionally, it is evident that the Tahoe suffers from greater aerodynamic losses due to its larger frontal area. However, both the Tahoe and Cruze follow the same patterns concerning cycle losses.

Weight also plays an important role in the system design. As the system weight is increased, significant changes occur in the vehicle MPG. This change can be seen in Figure 2.5. The MPG changes per payload increase were most pronounced in the high acceleration and low speed profiles, as these profiles suffered most of their
TABLE 2.3

ENERGY CONSUMPTION BREAKDOWN BY CYCLE FOR CRUZE

<table>
<thead>
<tr>
<th>Schedule</th>
<th>Aero. Drag Energy %</th>
<th>Rolling Resistance %</th>
<th>Braking %</th>
</tr>
</thead>
<tbody>
<tr>
<td>UDDS</td>
<td>20.3</td>
<td>29.87</td>
<td>49.81</td>
</tr>
<tr>
<td>HWY</td>
<td>53.89</td>
<td>33.57</td>
<td>12.52</td>
</tr>
<tr>
<td>NYCC</td>
<td>4.03</td>
<td>21.79</td>
<td>74.17</td>
</tr>
<tr>
<td>US06</td>
<td>47.14</td>
<td>19.74</td>
<td>33.1</td>
</tr>
</tbody>
</table>

TABLE 2.4

ENERGY CONSUMPTION BREAKDOWN BY CYCLE FOR TAHOE

<table>
<thead>
<tr>
<th>Schedule</th>
<th>Aero. Drag Energy %</th>
<th>Rolling Resistance %</th>
<th>Braking %</th>
</tr>
</thead>
<tbody>
<tr>
<td>UDDS</td>
<td>24.63</td>
<td>28.63</td>
<td>46.72</td>
</tr>
<tr>
<td>HWY</td>
<td>60.17</td>
<td>29.62</td>
<td>10.19</td>
</tr>
<tr>
<td>NYCC</td>
<td>5.07</td>
<td>21.65</td>
<td>73.27</td>
</tr>
<tr>
<td>US06</td>
<td>53.93</td>
<td>17.84</td>
<td>28.22</td>
</tr>
</tbody>
</table>

losses to rolling and inertial forces. This indicated that the system needed to be lightweight enough to justify its implementation, especially in city profiles. Should the system not provide a fuel economy benefit of approximately 3 MPG per 100 pounds, the system would not justify itself.

2.4 Summary

A design for a CAES system is chosen here, based on the work of Brown. This CAES system is composed of an IVT, a compressor/expander, and an air tank to provide power flow, energy conversion, and energy storage. A validated computer model of the system allows for the modeling necessary for modification without
construction of new hardware. A drive simulator was created, which accurately models vehicle energy demand over a number of EPA drive schedules. This energy demand analysis indicates areas of energy loss through dissipation and therefore potential areas for CAES system incorporation and overall vehicle fuel economy improvement.

Figure 2.5: Cruze MPG Changes with Payload Increases.
CHAPTER 3

ENERGY CYCLING INVESTIGATION

3.1 Introduction

The energy stored in the CAES system can be reintroduced to assist the vehicle in a multitude of ways. For the purposes of this research, two main propulsion pathways were considered: engine start-stop (ESS) and launch assist (LA). Neither pathway involves significant alterations to the engine or its combustion cycle.

ESS is a technology that is being increasingly implemented by automotive manufacturers as a simple alternative to a strong hybrid, while still reducing emissions and increasing fuel economy [30]. ESS has been implemented by many popular automotive manufacturers, including Ford, Volvo, Audi, BMW, and GM. LA requires more alterations to the vehicle drivetrain, but allows for load adjustments to be made to assist the engine. This allows for energy reintroduction during the entire drive schedule, rather than requiring the vehicle to come to a stop to assist. This technology has been implemented by many manufacturers also, including Toyota, Ford, and Hyundai.

For the purposes of this research, the Chevy Cruze, a compact sedan, was chosen as the target sedan, and the Chevrolet Tahoe was chosen as the target SUV due to their sizes and popularity. The Cruze is similar in size, engine displacement, and weight to other high volume sedans, such as the Toyota Camry and the Honda Accord. In fact, in 2011, seven of the ten best selling vehicles were sedans, all falling
in a size and weight range similar to the Cruze [14]. The Cruze was the ninth best selling car itself, with 231,732 units sold. The Chevrolet Tahoe was a leading seller in the SUV category and is representative of many full size SUVs on the market. Despite this focus, the principles explored in the following research transfer to other vehicles, including those out of the Cruze or Tahoe’s classes.

The Cruze and the Tahoe both have significant room for improvement through vehicle hybridization. The benefits are more pronounced for the Cruze, as a larger percentage of energy is dissipated through braking; however, there are still gains available for the Tahoe. For each vehicle, questions need to be asked as far as the potential gains for either an LA or ESS system. This investigation looks at the weight addition to the vehicle and the challenges associated with the implementation for both ESS and LA.

3.2 Drivetrain Organization

In order to implement a hybrid propulsion system, the vehicle drivetrain must be adapted to handle the energy flow. The first energy flow requirement is dependent on the level of hybridization desired. Many hybrids today use the alternate system to provide all of the force to push a car forward at low speeds. These systems output thousands of watts of power to launch the car. Alternatively, a system can be used in parallel with the ICE and simply add force to help the launch. This option allows for weaker systems to still help cycle energy.

A hybrid system can be coupled with the drivetrain in a multitude of ways. Two commonly implemented designs that both require extensive modifications to the vehicle drivetrain are detailed here. The first is a power assist parallel hybrid, as detailed in Figure 3.1. In this diagram, C/E represents the compressor/expander, AM represents the air motor, Trans represents the transmission, and E represents
the engine. The power assist hybrid vehicle is primarily driven by an engine, with the hybrid drive mostly for ESS and high load regions. This allows for the ICE to operate in more efficient regions during load optimization. Regenerative power is reintroduced separately from the engine pathway or is used to restart the engine.

Another possible design detailed in Figure 3.2 is a range extender or engine assist parallel. This design has the energy flowing through the hybrid system as a primary source, and the engine used to either re-fill the tank or directly augment the power delivery during high loads. This hybrid design requires significantly more modification to the standard vehicle, as well as a very powerful and efficient hybrid system.

The CAES system was designed to be implemented through a parallel hybrid drivetrain, as this organization allowed for heavy reliance on the ICE, lessening the efficiency demands placed on the CAES subsystem. Additionally, less overall modification to the vehicle is required for this implementation, reducing the need for redesign of the original vehicle.
3.3 Start-stop

3.3.1 Advantages and Disadvantages

Engine start-stop is a simple solution to a glaring problem that vehicles encounter in slow moving traffic — the idling engine. Small gasoline engines consume approximately 0.15 grams of gasoline every second spent idling. By eliminating this unnecessary fuel usage, gains can be seen primarily in city driving schedules. Table 3.1 indicates the total amount of time spent idling in each of the listed EPA schedules. As seen in the NYCC case, a typical city schedule, over a third of the time in the cycle is spent idling. By taking advantage of this wasted fuel, gains can be simply achieved with no post-engine drivetrain integration.

The implementation of this system is relatively simple. Rather than a full drivetrain restructure, ESS can use a direct connection with the driveshaft, coupled with the alternator belt. This integration with already existing structure increases the desirability of ESS due to the simple mechanical alterations necessary for implementation. This is in contrast with other hybrid systems, such as strong parallel or
### Table 3.1

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>UDDS</td>
<td>259</td>
<td>1370</td>
<td>18.9</td>
</tr>
<tr>
<td>HWY</td>
<td>6</td>
<td>766</td>
<td>0.91</td>
</tr>
<tr>
<td>NYCC</td>
<td>210</td>
<td>599</td>
<td>35.1</td>
</tr>
<tr>
<td>US06</td>
<td>45</td>
<td>601</td>
<td>7.59</td>
</tr>
</tbody>
</table>

split drivetrain hybrids, that require mechanical modifications post-engine or post-transmission. Modifications such as these are often disruptive to the chassis of the vehicle and require drivetrain re-organization. Additionally, a weak hybrid system takes up less space. As indicated before, strong hybrid systems add hundreds of pounds of weight to vehicles and can take up large amounts of space. These space losses manifest themselves in smaller backseats and trunks and also eliminate spare tires in many vehicles. A weak hybrid system such as the CAES could potentially be contained in the engine bay without altering the passenger compartment of the vehicle.

The lack of complication associated with vehicle modification additionally leads to reduced costs compared to strong hybrids. The cost of strong hybrid vehicles can significantly increase the vehicle base price. Table 3.2 compares some vehicles with their hybrid counterparts. These vehicles experience a significant increase in price over their non-hybrid counterparts. Additionally, the vehicles available as hybrids are not offered as base models, making them even more expensive in comparison to a base vehicle. Therefore, it is necessary to compute the price associated with the hybrid system by comparing the hybrid system with a comparable non-hybrid trim. This significantly increases the cost of entry to the hybrid market. A weak hybrid
system such as the CAES would only add a few hundred dollars for the cost of the materials, and the simplicity of an ESS system might allow for its implementation on any trim level, rather than restricting the fuel savings to more expensive models.

The ESS system is a lightweight, inexpensive method to address some, but not all, of the issues with ICE implementation. ESS only assists driving schedules with frequent braking. Because this energy delivery pathway is dependent on the car coming to rest, this system does not aid vehicles following highway schedules. In fact, it hurts the fuel economy of these vehicles, as the weight of the system causes a decrease in fuel efficiency as noted in the previous chapter. Strong hybrid systems can augment the engine during the driving cycle, allowing the engine to operate more efficiently while maintaining the power required by the driver of the vehicle. ESS can only assist the engine efficiency when the load is below optimal level, as it cannot add power to the drivetrain while the vehicle is moving. Additionally, in cold weather driving, ESS can allow the engine to cool down during stops, forcing the engine to operate less efficiently. This additional side effect can harm the engine’s performance enough to offset any fuel savings provided by the ESS system.

ESS also requires rapid engine turnover. Early ESS implementations were plagued
with lurching at start, as the motor started the engine and the engine then rapidly tried to meet accelerator demand [1]. This issue was overcome by more rapid engine starts to allow the engine to increase torque at a lesser rate. However, this rapid engine start requires a higher power ESS system, often increasing size and weight.

ESS is a desirable system for almost every consumer ICE vehicle. However, ESS cannot be implemented through the standard flooded lead-acid battery configuration present in cars today. ESS systems crank the engine much faster than a standard engine start and therefore require a standalone system. Standard batteries cannot provide the required current for ESS capability and will also lose their ability to cycle at a necessary rate within a few hundred charge cycles [6]. Therefore, a separate system is required to allow for ESS while not compromising basic operation of the vehicle. Typically, this is done with a battery energy storage system (BESS), coupled with an electric motor/generator to control energy conversion in and out of the system. This BESS is sized to be smaller than a strong hybrid system, so it requires a smaller battery. Similarly, a CAES system would be small and lightweight while still meeting the demands of the ESS functionality.

3.3.2 Implementation Requirements

Sizing an engine start-stop system is dependent on engine size, as well as expected braking cycles. These two factors dictate the amount of power the system must deliver to adequately start the engine without disrupting the driver and ensure that enough energy is stored in the tank so that the vehicle will be able to shut off and subsequently restart at every stop. For the case of an 1.8L inline 4-cylinder engine, similar to that of the Cruze, an 80 ft-lb torque at 250 RPM for 0.5 seconds is required for a guaranteed start. However, the time per start can be as low as 0.3 seconds. This information was provided by GM (personal communication, Prasad
Atluri) and is used for current BESS ESS configurations. Upon initial investigation, the current CAES system was determined to be inadequate, as the output of the expander was approximately 8 ft-lb at 500 RPM in the benchtop computer model. While the torque could be increased after passing through the IVT to decrease the RPM, the system would still not be powerful enough. Therefore, alternative power delivery options were pursued.

An air motor was chosen as an alternative power delivery system. Air motors take advantage of the high power of compressed air and are capable of transmitting large amounts of power with little spin up. Additionally, air motors are readily available and a proven technology, decreasing cost and simplifying procurement. While motors existed that operated more closely in line with the base requirements, a factor of safety of at least 1.1 was chosen to ensure that the required torque was provided. This was because air motors lose top end power rapidly while operating in sub-optimal pressure ranges, and the ESS system had to guarantee reliable rapid starts. The air motor chosen was a Parker P1V-A260-B0060, a large vane air motor. A drawing of the motor with planetary gearing can be seen in Figure 3.3. This motor was chosen because of its capability to deliver necessary torque while not being overly heavy. This particular air motor weighs 35 pounds. The motor can output approximately 92.2 ft-lbs at 250 RPM. This air motor was the smallest air motor found that satisfied the requirements with at least a factor of safety of 1.1. At this speed and torque, the motor is applying approximately 2100 Watts of power. It requires 21.13 gal/sec of flow at full power, and the power decreases nearly linearly with air flow. This leads to the determination of a 12.33 gal flow at the air motor’s working pressure of 87 psi with a start time of 0.5 seconds. By knowing the air consumption of the motor, the number of charges per tank can be calculated for a given tank size. The performance of the motor can be seen in the torque and power...
graph in Figure 3.4 [2].

The power requirements for a Tahoe starter motor are approximately 80% more than that of the Cruze, based on starter motor sizing currently in place in the vehicles [3]. Therefore, it was determined that two air motors could be used for the ESS implementation for the Tahoe.

As a limitation, the size and weight of the system must always be offset by the fuel saving gains produced by the hybridization. The volume of the air motor and its additional weight therefore must be offset by the ESS gains.

3.3.3 System Performance

In order to implement an ESS pathway using an air motor, the CAES system increases weight to appropriately assume the infrastructure for the air motor’s operation. The air motor requires a pressure regulator, flow control valve, air treatment
unit, and a silencer for proper use. These parts do not add much weight or expense to the overall system, and in fact, weigh less than the air motor itself. As indicated before, a strength of the ESS system is the ability to add alternate functionality without requiring significant redesign of the base CAES system. Therefore, the air motor simply needs to be connected to the air tank and does not alter the single pathway design by Brown [42]. In total, the air motor assembly adds approximately 50 pounds to the CAES system. The fuel economy losses due to weight for the total CAES system is then approximately equal a 3% loss of base fuel economy for the Cruze and a 2% loss for the Tahoe, as indicated by altering the payload weight of the driving simulator. These losses must be offset by the gains of the ESS.

It is important to size the energy reservoir appropriately to prevent loss of functionality due to low energy reserves. By understanding that the air motor uses at most 12 gallons of air at 87 psi for each start, the CAES system was sized to take
in at least that amount of air per ESS occurrence for the EPA city schedules.

As discussed above, there are two main pathways for energy recapture: braking and engine load optimization. The driving schedules were analyzed with this in mind, and the possibilities for energy recapture per engine start were noted. In Figure 3.5, each drive cycle is split into braking periods (green), engine load optimization periods (yellow), and periods in which no energy recapture was possible (black). The red circles at the top of the plots indicate ESS occurrences. These four graphs detail the potential uses for ESS technology and the potential gains for each cycle. For the UDDS and NYCC cycles, a significant number of ESS occurrences exist. In the US06 and HWYFET cycles, there are very few vehicle stops. With this understanding, the ESS optimization in Chapter 4 will focus on the two cycles with high potential gains, while illustrating the minor gains present in the other cycles.

In order to ensure consistent engine start, an appropriate amount of energy needs to be stored in the system. The amount of energy per ESS was calculated by

\[ J = \frac{2 \pi T \, RPM \, t}{44.235} \]  

(3.1)

where \( J \) is the energy required in Joules, \( T \) is the torque necessary in Newtons, \( RPM \) is the speed of the motor, and \( t \) is the time of application in seconds. Based on the time of 0.3 to 0.5 seconds with torque of 80 Nm and speed of 250 RPM, the energy requirement was determined to be between 840 and 1400 J. The total amount of braking energy for each ESS incidence is shown in Figure 3.6 the Cruze and Figure 3.7 for the Tahoe.

The energy change is very similar between the Cruze and the Tahoe. The only variations between the two cycles occur in the higher speed portions where a greater proportion of the vehicle energy is lost to air resistance rather than braking. How-
Figure 3.5: Drive Schedules Color Coded by Drivetrain Requirement.
Figure 3.6: Cruze Energy per ESS. Blue indicates braking energy. Red line indicates energy to start the engine. Note that this line is just above the axis.
Figure 3.7: Tahoe Energy per ESS. Blue indicates braking energy. Red line indicates energy to start the engine. Note that this line is just above the axis.
ever, these differences are insignificant for the purposes of designing the ESS system. Of the four cycles studied, the braking per incidence of ESS is significantly higher for the UDDS cycle. This is because of the higher speeds attained during the UDDS cycle, as well as the long periods between ESS occurrences. However, even with the lower braking power available during the NYCC cycle, the minimum required energy is stored over almost all of the ESS cycles without requiring energy carry over from previous cycles. This leads to a design with a tank capable of holding only a few charges to maximize weight savings.

An 11 gallon tank for the Cruze, similar to the one currently in place on the CAES hardware, cycling from a high of 300 psi to a low of 90 psi could hold between two and four ESS charges depending on the start time required, varying between 0.3 and 0.5 seconds. A 20 gallon tank for the Tahoe cycling from a high of 300 psi to a low of 90 psi could hold between two and four ESS charges, based on the increased power requirements of the Tahoe for ESS. This charge count was estimated using the air consumption rate of the air motor as provided by Parker. This is sufficient for all of the EPA driving schedules provided, based on braking and ESS patterns. This was also only absorbing energy from the braking of the car and not using any engine optimization. With such engine optimization, the tank could conceivably be smaller, holding only one full charge at a time. In fact, the energy available due to engine optimization is significantly greater than that due to braking.

Here in Figures 3.6 and 3.7, the braking energy is shown in blue, with the red line indicating the amount of energy necessary to start the engine. The energy from engine optimization is almost always larger than the braking energy, however, this energy is not necessary for implementation in an ESS only system. The red line indicating the necessary energy is almost indistinguishable from the axis, indicating that the energy available through braking far exceeds the needed energy. With the
hybrid system only focused on ESS, the compressor/expander can be optimized for compression, leading to greater efficiency while compressing the air. Also, the IVT would not have to operate in both a positive and negative direction, reducing that subsystem to a CVT. The entire hybrid system would weigh approximately of 200 pounds for the Cruze due to adding the 60 pound air motor system to the 130 pound CAES system. The system for the Tahoe would weigh approximately 240 pounds, as needed for the inclusion of a second air motor.

By adjusting the simulator’s vehicle weight estimate appropriately and implementing an energy reintroduction subroutine into the driving simulator, the effects of the ESS were determined. This subroutine charged an enlarged model of the benchtop air compressor when the car was braking and subtracted the amount of air necessary to start the engine when the vehicle accelerated again. This benchtop model was enlarged to be able to intake the necessary energy to ensure ESS. The subroutine then subtracted the fuel that was consumed in the simulator during engine idle, to account for the benefits of the ESS. The efficiency gains over the four cycles can be seen in Table 3.3.

**TABLE 3.3**

FUEL ECONOMY IMPROVEMENTS DUE TO ESS

<table>
<thead>
<tr>
<th>Schedule</th>
<th>Cruze Percent Improvement [%]</th>
<th>Tahoe Percent Improvement [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>UDDS</td>
<td>27.55</td>
<td>15.22</td>
</tr>
<tr>
<td>HWY</td>
<td>0.26</td>
<td>0.15</td>
</tr>
<tr>
<td>NYCC</td>
<td>25.14</td>
<td>12.52</td>
</tr>
<tr>
<td>US06</td>
<td>4.83</td>
<td>2.70</td>
</tr>
</tbody>
</table>

These improvements are worthwhile in the cases of UDDS and NYCC despite the vehicle weight gain. The US06 cycle is marginally beneficial in the case of the Cruze
and slightly less beneficial for the Tahoe. The benefit in the HWYFET schedule is marginal and only exists due to assumptions of initial tank charge capable of an ESS, as well as simulating the beginning of the schedule as a warm start. The most drastic changes in fuel economy were realized in the UDDS and the NYCC cycles, which follows from the ESS incidence frequency. This indicates that system implementation would be useful in any vehicle not exclusively designed for highway use. As most consumer transportation vehicles experience some start-stop driving in cities or in heavy traffic, there is a strong argument for universal implementation in new vehicles.

3.4 Launch Assist

3.4.1 Advantages and Disadvantages

Launch assist is an alternate pathway for energy reintroduction to a vehicle. The LA system assists the existing ICE drivetrain when engine load demand is high, allowing for more responsive acceleration while the engine operates more efficiently. LA systems are often coupled with a load optimization control system. By permitting the ICE to operate more efficiently at low load points, the fuel is not only better converted to mechanical energy, but excess energy is also recaptured to be introduced during high load periods. At cruising speeds often characterized by suboptimal performance, the engine is capable of operating at the best load point, storing the extra energy in the LA system for use later. This capability allows it to increase energy efficiency over the whole vehicle driving schedule, rather than relying only on vehicle stops to provide energy reintroduction points. Finally, the LA system lessens the waste of energy during braking periods by storing energy that would normally be dissipated by the brakes. LA is a more comprehensive solution to vehicle efficiency improvements than ESS, but LA is not without challenges. An
LA system needs to be larger than the typical ESS system due to the high energy requirements associated with accelerating a vehicle. It also must be capable of continuous power delivery for extended periods. The system’s response time must be as fast or faster than an ICE.

Although LA systems are costly, their benefits far outweigh their costs in periods of high fuel prices. For early LA hybrids, gasoline prices of $1 or $2 per gallon led to slowly gained or even unachievable returns on the hybrid cost increase as indicated in Table 3.4. Studying current LA hybrids with increased fuel prices, the gains become more apparent. To use the Cruze as an example, the hybrid attains modest fuel economy benefits over the comparable non-hybrid trim line, as shown in Table 3.4.

### TABLE 3.4

MODEL PRICE AND FUEL ECONOMY COMPARISONS

<table>
<thead>
<tr>
<th>Vehicle</th>
<th>Price [$]</th>
<th>City MPG</th>
<th>HWY MPG</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cruze (Baseline)</td>
<td>17130</td>
<td>26</td>
<td>38</td>
</tr>
<tr>
<td>Cruze (comp. trim)</td>
<td>18560</td>
<td>26</td>
<td>38</td>
</tr>
<tr>
<td>Cruze (hybrid)</td>
<td>19680</td>
<td>28</td>
<td>42</td>
</tr>
<tr>
<td>Tahoe (Baseline)</td>
<td>39080</td>
<td>15</td>
<td>21</td>
</tr>
<tr>
<td>Tahoe (comp. trim)</td>
<td>45225</td>
<td>15</td>
<td>21</td>
</tr>
<tr>
<td>Tahoe (hybrid)</td>
<td>53290</td>
<td>20</td>
<td>23</td>
</tr>
</tbody>
</table>

At $1.30 per gallon as it was in 2000, the Cruze earns its price difference over its comparable model in just over 320,000 miles [11]. This was calculated by

\[ \frac{Miles}{\$} = (City \times 0.55 + Hwy \times 0.45) \times \frac{Gal}{\$}, \]

using the standard EPA combined fuel economy calculation. This equation was then multiplied by the price difference to calculate the range necessary to meet
the increased cost. This distance is significantly farther than the average distance traveled during the life of a sedan. With the Tahoe, it would take 686,000 miles to make up the price difference. However, at $4 per gallon, the Cruze earns that same $1120 in 104,480 miles. The Tahoe takes 222,960 miles at the same price point. If gas prices increase as they have done in Europe to over $10 per gallon, the vehicle earns its initial investment back in a mere 41,000 miles for the Cruze and 89,000 miles for the Tahoe. Additionally, LA vehicles have lower emissions, easing the process of passing increasingly stringent EPA standards. LA systems offer the potential for large gains in fuel economy by addressing the issue of energy cycling throughout the whole drive schedule of a vehicle.

3.4.2 Implementation Requirements

In order to properly implement a CAES LA hybrid system, the system must be able to handle large amounts of energy flow in and out, as well as store an appropriate amount to account for periods of high acceleration without any energy recapture. This energy flow is significantly greater than the flow present in an ESS system, as the energy required to accelerate a vehicle is much greater than that necessary to turn over the engine. This energy flow can be accomplished by using multiple compressor/expanders or more efficient single compressor/expanders. In order to appropriately size a CAES system to fulfill these needs, studies of the various energy demands were conducted. The average demand, largest demand, and variance of demand placed on the system during the drive cycles were calculated. By understanding these values, an appropriately sized system can be tailored to handle most of the overall vehicle demands without being overly bulky. The power flow averages, peaks, and standard deviations are listed in Tables 3.5, 3.6, 3.7, and 3.8.
Capturing all of the dissipated energy would be impractical and would lead to an unnecessarily large system. However, by studying the average dissipated power available for recapture by the CAES system over the UDDS, HWYFET, US06, and NYCC cycles, a general picture of compressor/expander requirements can be formed. Finally, a tank needs to be sized to appropriately absorb this energy, store it for later use, and handle extended discharges. A list of the average discharge times per braking and accelerating cycle is shown in Table 3.9.

Based on these average discharge times, a general understanding of the total energy capacity required of the tank is acquired and can be applied to design of the optimized system. Assuming a discharge time of approximately 10 seconds, based
TABLE 3.7

VEHICLE POWER DEMANDS - CRUZE

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>UDDS</td>
<td>13423</td>
<td>50938</td>
<td>11996</td>
</tr>
<tr>
<td>HWYFET</td>
<td>14724</td>
<td>70917</td>
<td>15832</td>
</tr>
<tr>
<td>NYCC</td>
<td>9087</td>
<td>44563</td>
<td>10740</td>
</tr>
<tr>
<td>US06</td>
<td>35673</td>
<td>97909</td>
<td>25563</td>
</tr>
</tbody>
</table>

TABLE 3.8

VEHICLE POWER DEMANDS - TAHOE

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>UDDS</td>
<td>17649</td>
<td>95179</td>
<td>15194</td>
</tr>
<tr>
<td>HWYFET</td>
<td>23980</td>
<td>78137</td>
<td>13300</td>
</tr>
<tr>
<td>NYCC</td>
<td>11248</td>
<td>72768</td>
<td>13650</td>
</tr>
<tr>
<td>US06</td>
<td>49080</td>
<td>213310</td>
<td>36390</td>
</tr>
</tbody>
</table>

on the average acceleration duration as determined by the simulator, the tank for the Cruze should be at approximately 100,000 Joules and the tank for the Tahoe 230,000 Joules.

3.4.3 System Performance

The designed launch assist energy delivery path incorporates the basic system overview as detailed in Figure 3.1. This energy path allows for the fuel savings of ESS while also providing benefits on higher speed cycles, an area in which the ESS significantly lacks. A pure LA system does not make sense for implementation, as the benefits of ESS with marginal weight gain lead to substantial fuel savings as indicated during the ESS system study.
Table 3.9

Average Discharge Time Per Acceleration/Breaking Cycle

<table>
<thead>
<tr>
<th>Cycle</th>
<th>Time [sec]</th>
</tr>
</thead>
<tbody>
<tr>
<td>UDDS</td>
<td>20</td>
</tr>
<tr>
<td>HWYFET</td>
<td>40</td>
</tr>
<tr>
<td>NYCC</td>
<td>11</td>
</tr>
<tr>
<td>US06</td>
<td>21</td>
</tr>
</tbody>
</table>

In order to implement the LA system, the compressor/expander needs to be significantly more powerful than the benchtop model. The current benchtop model is capable of a maximum power output of approximately 200 W, an order of magnitude short of what is necessary to effect substantial change for a full size vehicle’s performance. With optimization, the current system could see an increase in power output; however, the final system would most likely need multiple compressor/expanders or a significantly larger compressor/expander to handle the power flow. The assumption was made that the weight penalty for the larger compressor/expander would be the same as for multiple compressor/expanders, and therefore the system was scaled based on power requirements. For a system using similarly sized base compressors, 3 compressor/expanders and reasonably increasing the power output as detailed though the optimization in Chapter 4, the system could output approximately 1000 W. While falling short of a strong hybrid’s capability, the CAES LA/ESS system would still see marked benefits over a non-hybrid system. The weight estimations of the two vehicles were done by increasing the number of compressor/expanders linearly based on the power requirements. This was determined by scaling the 31 pound compressor/expander by a factor of 3 for the Cruze and by a factor of 7 for the Tahoe. The vehicle payload was increased by 300 pounds for the Cruze and 400
pounds for the Tahoe due to their respective CAES systems. The Cruze was assumed to have a tank size of 110 gallons, as sized to handle discharges listed before, storing 100,000 Joules. The Tahoe was assumed to have a 250 gallon tank, similarly sized to handle the idealized power storage. Assuming round trip efficiency of 46% (maximum predicted by Brown [42] to be achievable), the driving simulator was run over the drive schedules. The results can be seen in Figure 3.8.

![Figure 3.8: Cruze and Tahoe LA Fuel Economies. The blue represents the results from the simulated vehicle with no hybridization. The red represents the results from the simulated vehicle with the sized hybrid system implemented.](image)

In Figure 3.8, the Chevy Cruze is compared in the top plot, and the Chevy Tahoe in the bottom. As is evident, the benefits of a CAES LA/ESS system are most pronounced for low speed cycles, where much of the energy is lost to braking as compared to the higher speed schedules where most of the energy loss is due to air and rolling resistance. However, this implementation leads to an overall
improvement of approximately 50% for the averaged city/hwy mileage. The effects were more pronounced in the case of the Cruze, due to lesser energy losses due to air and rolling resistance than the large faced SUV. This provides greater energy recapture possibilities for the Cruze. However, further work into optimizing the final system over the benchtop model needs to be conducted in order to realize these improvements. These optimizations will be handled in Chapter 4.

3.5 Summary

Here a pilot study was conducted to determine the viability of the CAES system. Drivetrain organization was investigated, and a system that allowed for separate engine and hybrid input was chosen. Additionally, braking and engine load optimization were investigated as methodologies for energy recapture. Braking recapture was promising, as this method only involves storing energy that would normally be dissipated by the brakes. Engine load optimization was determined to be more promising, especially for highway cycles. However, the control for this is more complex, and its implementation requires more drivetrain alterations.

Basic requirements on a start-stop system were detailed, based on the drive cycle analysis afforded by the simulator. The requirements ruled out implementation of the initial compressor/expander system and led to implementation of an air motor to provide the torque necessary to start the engine. ESS benefits were significant in the city profiles analyzed and less pronounced in highway cycles. This is due to the lower rate of ESS occurrence during highway cycles. LA hybridization was also investigated, and system benefits were analyzed. This system was found to have large power requirements which could prove problematic for the low energy density of CAES. However, unlike ESS, LA offers benefits during all of the driving schedules due to its ability to reintroduce energy at more points during the vehicle’s motion.
CHAPTER 4

SYSTEM OPTIMIZATION

4.1 Introduction

Implementation of the systems described in the pilot study requires alterations to the compressor/expander and air tank of the benchtop design in order to achieve better efficiency and meet system demands. The ESS-only system requires a small air tank and high compression efficiency, as the expansion functionality is not needed at all. Additionally, the energy demands of the ESS system are met on a cyclic basis, meaning that the air tank can be shrunk due to the closely related braking energy dissipation and the occurrences of start-stop points. Prior to each engine stop, the car itself comes to a stop, so the braking energy to accomplish that can be stored for the subsequent engine restart. The ESS/LA system proposed requires the dual functionality of robust compression and expansion due to the system’s energy cycling requirements. For this system, large amounts of power are necessary to propel the car forward, so the energy intake must also be capable of large amounts of power flow at a reasonable efficiency.

In order to model the required alterations to the benchtop system, the experimentally verified system model was adapted based on a number of assumptions. First, the modified systems were limited to the use of a reciprocating type compressor as in the existing system, but with variations in the design parameters of stroke, bore, and compression ratio to achieve the adjusted design pressures. Secondly, a
modified rotary valve was used to provide valve actuation, rather than exploring the possibility of alternate VVT designs. Finally, the system power was assumed to scale linearly as the number of compressor/expander units was increased or the compressor/expander unit was increased in size to account for different required power flows. This assumption was made as initial tests involving scaling the system parameters led to linear power improvements. In the case of multiple compressor/expanders, the individual units’ power should approximately sum in a linear fashion. This assumption allowed for optimization of a nominal system based on a power independent objective function. Once this optimization was completed, the system parameters were linearly scaled to a system capable of meeting the power requirements set by the drive schedule and hybrid system demands for a particular vehicle.

Brown [42] identified a number of ways in which the efficiency of the benchtop system could be improved, and many of these were implemented in the modified CAES systems prior to detailed optimization. The design pressure was increased to 300 psi in order to increase both the power capacity of the compressor/expander. 300 psi was chosen because it allowed for higher energy density in the storage tank, while remaining in the range of pressures for commercially available low pressure air tanks. This helped limit system costs, as well as decreased the risk of dangerous tank rupture in the case of a collision. The valve diameter was increased by a factor of 100% in order to allow for higher flow rates in and out of the compressor/expander due to larger area for valve openings. The compression ratio of the compressor was increased by reducing the residual volume of the compressor at top dead center by 50%. This allowed for the higher design pressure increased the efficiency of the compressor. Finally, the air tank was heated to 500 K. This would be achieved by using engine exhaust, which exits the combustion chambers at temperatures between
600 K to 1200 K.

Once these changes were made, a nominal system was designed. Once this nominal system was designed a parameter sensitivity analysis was completed to explore the possible solutions the drive schedule specific optimization could lead to evaluate how robust the system is to errors in modeling. Finally, an optimization was conducted to tweak the chosen system parameters to an optimal solution for a particular cycle.

4.2 Nominal System Design and Parameter Sensitivity

An initial system was designed to provide a nominal starting point for later optimization. This system model was a modified benchtop model following the suggestions by Brown [42]. Three portions of the CAES system were modified: the tank, the compressor/expander, and the rotary valve assembly. The tank volume and operating pressure range were altered. The compression chamber bore, stroke, and clearance were modified. The rotary valve diameter and speed were changed. The valve opening length, width, and mating port length were changed, as well as the opening and closing angles of the valves.

The energy storage tank must stay within a reasonable state of charge. This is necessary to ensure that the hybrid system does not deplete itself and leave the user stranded. The standard accepted range for charge is between 30% and 80% charge. However, as mentioned by Same [10], the state of charge of most hybrid systems fluctuate only a few percent to provide a nearly constant power supply, as well as to avoid damaging the batteries with heavy cycling. While this small range of charge would be conducive to optimizing the compressor/expander, the tank required to allow for this would be very large or have a much higher pressure to achieve the energy density necessary. Therefore, the system was designed for variable pressure
ranges and smaller tank sizes.

The efficiency of the system was dependent on the tank pressure. The pressure difference between the tank and the atmosphere drives the flow equations modeling the air movement in the system. The pressure is dependent on the tank volume as it fluctuates between charging and discharging. This volume was directly related to the total energy storage capacity of the tank which was dependent on the vehicle and the drive schedule. For the purposes of the nominal system, the original 11 gallon tank was used. This was because the efficiency of the compressor/expander unit was independent of the tank size. The efficiency at different pressures could be studied regardless of tank size, and the final tank could then be sized to permit operation in the desired pressure range.

The compressor was modified to change the compression ratio. This was done by altering the bore, stroke, and clearance height. The compressor was designed to operate at a compression ratio of 40 in order to allow for the 300 psi maximum design pressure to be achieved just after the halfway point of the compression stroke, as recommended by Kerr [38]. The clearance height was then chosen to be 5% of the bore, also recommended by Kerr. Kerr indicated an optimal piston volume-to-crank-length ratio of approximately 5. Therefore, based on Eqs. 4.1, 4.2, and 4.3 from Kerr, the bore was calculated to be 2.5 inches, with a stroke of 2 inches and a clearance volume of 0.245 cubic inches.

\[ V_C = \frac{\pi}{4}b^2c, \quad (4.1) \]

\[ \frac{0.25 \times \pi \times b^2}{\frac{s}{2}} = 5, \quad (4.2) \]

\[ CR = \frac{\frac{\pi b^2s + V_C}{V_C}}. \quad (4.3) \]
Here $CR$ represents the compression ratio, $c$ is the clearance height, $b$ is the bore of the compressor piston, $s$ is the compressor stroke, and $V_C$ is the clearance volume. These values were within a reasonable range for off the shelf compressor units.

The rotary valve assembly played a critical role in overall system efficiency. Because system efficiency and power were dependent on thermodynamic losses in the flow between the compressor/expander and the air tank, the valve sizing and operational speed had significant effects on the overall round trip efficiency (RTE). An ideal pneumatic system follows a path generally indicated by Figure 4.1. This compression cycle begins at 1 with the compression chamber at bottom dead center and the pressure and temperature at atmospheric levels. The air is then compressed adiabatically until point 2. There, the air flows through the high side valve to the tank until the compression chamber reaches top dead center. At this point, the high side valve closes, and the air expands to atmospheric levels. Subsequently, the low

![Figure 4.1: Ideal PV diagram](image)

54
side valve opens, and the compression chamber is recharged. This process is reversed for ideal expansion operation. While both of these ideal cycles would lead to 100% efficiency, this cannot be practically achieved. Because the compression cannot be achieved adiabatically or isothermally, the polytropic compression process suffers from efficiency losses. Additionally, because valve timing will not be ideal for a full range of pressures and speeds, additional effects will detract from the efficiency.

With fixed valve timing approaches, the valving is designed to operate at a specific design pressure. In cases where the operating tank pressure is above or below the design pressure, there are inefficiencies. If the system is operating above design pressure in compression mode, air will blow from the tank back into the cylinder due to early high-side valve opening, requiring extra work to pump the air back to the tank. In the case of expansion mode, there will be blowback from 4→3 in Figure 4.1, and the air will be under-expanded during stage 2→1. If the tank is below design pressure, the compressor will over-compress the air in the cylinder, wasting energy.

This efficiency loss can be prevented when operation is limited to below design pressure by introducing reed valves. These reed valves were implemented both in the benchtop model and the nominal model and can be seen in Figure 2.3. These reed valves open passively when the pressure differential exceeds a value in one direction. These valves do not open when this value is reached for the reverse pressure gradient. Mechanically, these valves assist the system during compression by opening to prevent over-compression at point 2 in Figure 4.1 by venting from the compression chamber to the air tank and to prevent over-expansion at point 4 by permitting air to come in from the atmosphere. The valves assist expansion by preventing over-compression from 4→3 and over-expansion from 2→1. This is not the case for pressures above the design pressure because the reed valves will remain
inactive as the pressure gradient is reversed. This means that the additional flow required will not be provided by the closed valves. The only potential benefit of the reed valves in the above design pressure operation would be to open to augment flow area for the active rotary valves during the compression stroke. The reed valves do not help prevent the detrimental over-compression and over-expansion effects to any extent for either compression or expansion.

This use of reed valves led to asymmetric RTE over the input-output cycles. The compression cycles experience high thermodynamic and volumetric efficiency, while the expansion cycles suffer from volumetric efficiency losses, as defined for a compressor in Eq. 4.4 and for an expander in Eq. 4.5 [25].

\[
\eta_V = \frac{V_1 - V_4}{V_1 - V_3} 
\]

(4.4)

\[
\eta_V = \frac{V_3 - V_2}{V_3 - V_1} 
\]

(4.5)

Here, \(\eta_V\) is the volumetric efficiency, and \(V_n\) refers to the volume at a particular point in the PV diagram. The volume \(V_2\) is close to \(V_1\) as specified for the ideal compression cycle, leading to issues with volumetric efficiency. For cycles requiring high compression efficiency, this is an acceptable compromise. However, for balanced use of the compressor/expander system, the valve timing was modified to combat this effect. This was done by opening the low side active valve just after top dead center, which significantly increases the volumetric efficiency in the expansion mode by decreasing \(V_2\). This is the valve that is open between points 1 and 4. However, this creates losses in thermodynamic efficiency. Due to a free expansion event that occurs when the high side valve opens and air flows to fill the chamber. This leads to an increase in power input requirements during the compression mode because residual air is vented rather than expanded in the cylinder. This loss can be mitigated by decreasing the clearance volume of the compressor.
In order to maximize system efficiency by following an ideal PV cycle, the high side valve was opened when the design pressure was reached and closed at top dead center. To improve the expander volumetric efficiency, the low side was opened almost directly after top dead center and kept open until bottom dead center to allow for interchange with the atmosphere. It was not opened directly at top dead center to allow for some balance between thermodynamic losses of the compressed gas remaining in the cylinder and the volumetric efficiency losses. This opening occurred 4 degrees after top dead center. The necessary cylinder pressure of 300 psi was achieved 3 degrees after the piston moved halfway up its stroke as calculated by the cylinder volume change.

The valve and mating hole shape and width were then chosen to allow complete cylinder flow through the valve during the cycle. This allowed for the necessary air to flow between the cylinder and the tank or atmosphere, without closing before the necessary air could be exchanged. Hole shape design seeks to maximize the amount of time spent at maximum area while minimizing the amount of time transitioning from open to closed. Therefore, a circular valve opening with a circular hole does not mate well, as the overlapping area increases faster than linearly until maximum area and then decreases rapidly also. The ideal valve is a wide, short valve mated with an oversized rectangular hole. Both choked and unchoked flow were considered for port design. The flow was determined to be choked or unchoked by Eq. 4.6 [42],

\[
\frac{P_{\text{high}}}{P_{\text{low}}} > \left(\frac{k + 1}{2}\right) k^{\frac{k}{k-1}},
\]

where \(k\) is the specific heat ratio, \(P_{\text{high}}\) is the pressure on the higher pressure side and \(P_{\text{low}}\) is the pressure on the lower pressure side. If the pressure ratio is higher than the constant determined by the specific heat ratio, then the flow is choked. For
unchoked flow,
\[ \dot{M} = AYC \sqrt{2\rho(P_{\text{high}} - P_{\text{low}})}, \]  
(4.7)

where \( A \) is the area of the orifice, \( Y \) is the expansion factor, \( C \) is the discharge coefficient, \( \rho \) is the upstream air density, \( P_{\text{high}} \) is the upstream pressure, and \( P_{\text{low}} \) is the downstream pressure. The expansion factor is given by

\[ Y = \sqrt{\frac{r^{2/k}}{(k-1)(1-r^{(k-1)/k})} \frac{1 - \beta^4}{1 - \beta^4 r^{2/k}}}, \]  
(4.8)

where \( r = P_{\text{high}}/P_{\text{low}} \) and \( \beta \) is the discharge coefficient. Here, the geometry-based discharge coefficient was ignored and set to zero due to the complex geometry present in the valve flow channels. In the case of choked flow,

\[ \dot{M} = AC \sqrt{k\rho P_{\text{high}} \left( \frac{2}{k+1} \right)^{\frac{k+1}{k}}}. \]  
(4.9)

Here, \( C = 0.8 \), \( P_{\text{low}} = 90 \text{psi} \), \( k = 1.4 \), and \( \rho \) was calculated by

\[ \rho = \frac{P_{\text{high}}}{R \ast T_{\text{tank}}}, \]  
(4.10)

where \( R = 287 \) and \( T_{\text{tank}} = 500 \). Eqs.4.7, 4.8, and 4.9 with the rotary valve diameter of 2” led to a high side valve length of 0.4 inches centered 19 degrees before top dead center. This valve area was sized to meet mass flow requirements of 0.000207 kg of air, while the length was sized to stay open for the intended portion of the compression or expansion cycle as indicated by the PV analysis. Therefore, the width was 1.54 inches which was sized to meet mass flow requirements. This hole was deemed acceptable to exhaust the air from the cylinder for flow scenarios at 900 RPM, the chosen maximum speed. This speed was selected because it was the maximum speed of the validated model. More importantly, as the speed increased, the valve sizing increased in order to provide adequate flow, and beyond 900 RPM, the valve lengths added together were nearly the bore of the compressor. At higher
speeds such as 1000 RPM, the valving would not fit across the compression chamber bore. The high side valves could theoretically handle 0.000972 kg of air flow for the choked flow scenario as computed from Eq 4.9. For the non-choked case, the system could handle 0.000293 of air flow as computed from Eq. 4.7.

For the low side valves, the same process was followed. The valve length was designed to have a hole length of 0.74 inches, centered 47 degrees from bottom dead center based on the PV requirements. The valve opening was oversized in order to ensure that enough atmospheric air to fill the compression chamber is taken in for the compression cycle and that all of the air is exhausted in the expansion cycle. Square holes were designed and found to provide sufficient flow to exhaust the cylinder for the non-choked case with a potential 0.00101 kg of air for the valve duration. There was no choked flow for the low side port, as the pressure ratios were always below the choked boundary. A diagram of the valve open and close areas is shown in Figure 4.2. The high side valve opens at 300 psi, just past halfway up the compression stroke. The low side valve opens right after top dead center, and stays open until bottom dead center.

*Fmincon*, the MATLAB gradient-based optimization tool, was used to check for an optimal valve timing and sizing arrangement. Since the tool is prone to local minima, care must be taken to achieve a global minimum solution. In the case of this optimization, the system found only one local minima. However, further search of the design space was not conducted, as the nominal system provided a starting point that was designed to be close to the final solution. The system was constrained to optimize based on the power multiplied by the mechanical efficiency of both the compression and expansion cycles following Eq. 4.11, effectively maximizing the
amount of energy that will be cycled through the system.

\[
\frac{\text{Energy}_{\text{out}}}{\text{Energy}_{\text{in}}} \times \text{Power}_{\text{in}}
\]  

(4.11)

Only optimizing cycle efficiency was not sufficient, as certain systems allowed for greater power return, despite their lower efficiencies due to their ability to handle larger volumetric flow. Fmincon optimized with the parameters of valve shaft diameter and high and low side valve length, valve width, and angle. These parameters were chosen because they allowed for the greatest variance in the PV cycle which determined efficiency and power. The valve length and width were multiplied by a factor to scale with the diameter of the valve. The bounds of these variables can be seen in Table 4.1. The shaft diameter maximum bound was chosen based on the size of the compressor bore, and the minimum by the benchtop system. The valve width bound was set so that the openings would fit on the compressor head,
TABLE 4.1

FMINCON PARAMETER BOUNDS

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Low Bound</th>
<th>High Bound</th>
</tr>
</thead>
<tbody>
<tr>
<td>Valve Angle</td>
<td>NOM -20</td>
<td>NOM + 20</td>
</tr>
<tr>
<td>Valve Width</td>
<td>NOM * 0.5</td>
<td>NOM * 1.5</td>
</tr>
<tr>
<td>Valve Length</td>
<td>NOM * 0.5</td>
<td>NOM * 1.5</td>
</tr>
<tr>
<td>Shaft Diameter</td>
<td>1</td>
<td>2</td>
</tr>
</tbody>
</table>

and the length of the valve was limited by the the open time for a full compressor stroke. The valve angle bounds were set to allow for the shortest valve length open limit to extend to either the beginning or end of the piston’s bottom-to-top stroke.

Given the starting point of the nominal system and operated over a single full charge and discharge cycle, the \textit{fmincon} optimization determined that the high side valve should be opened 2 degrees later, past top dead center. This is most likely due to the effect of the inertia of the air as described by the choked and non choked flow equations. The low side valve timing was not changed, and neither of the valve hole sizes were changed. This is likely due to the valve holes being oversized for the cases, and the low side timing being nominally chosen based on preliminary efficiency tuning. The calculated RTE of the \textit{fmincon} optimized model for a full charge and discharge cycle was calculated to be 42.40%. This number is a significant improvement over the original benchtop system, which had a maximum efficiency of 16%. However, these are reasonable with a friction reduction estimate of 50% as recommended by Brown [42], increasing the tank temperature by way of heating, and properly sizing the valves for a high pressure system. The tank heating accounted for 51% of the improvement, friction for 26%, and valve sizing for 23% of the overall improvement.
From the optimized solution, the valve diameter, length and width, and valve angle were adjusted to test for system sensitivity. This was done by varying the parameters over the range conducted for the optimization and then running the system model over the same charge-discharge cycle as for the optimization. Any decrease in valve diameter led to smaller areas for flow, leading to efficiency losses. It was found that delaying the high side valving caused an 8% decrease per degree in RTE. This makes sense, as there would then be overlap between the low side and high side valves in this case. By increasing the valving there was a 4% decrease in RTE per degree. This is due to extra work spent compressing air that is not forced into the air tank. For the low side valving, delaying the opening caused a 2% decrease per degree in efficiency, most likely due to losses in volumetric efficiency during expansion. By opening the valving early, there was a 9% decrease in efficiency per degree, as this effect is the same as delaying the high side valving. For the valve sizing, efficiency was lost by decreasing the length of the valve holes, as this would result in over-compression during the compression cycle and under-expansion on the expansion cycle. By increasing the high side valve lengths, there were benefits for lower pressure operation, but losses at design pressure. The low side valve length increases resulted in efficiency losses due to overlap with the high side valving or opening past bottom dead center and losing efficiency due to decreased volumetric efficiency. No efficiency changes were achieved by increasing the width of the holes as there was no further air to flow through, and efficiency was lost as the holes decreased in size because the air could not fully enter or exit the compression chamber. This would not be the case if the compression ratio was increased for the system, as the necessary width would increase to handle the increased flow.

The CAES system was operated over a range of speeds and tank pressures. The efficiency of the system as well as the power handling capability were computed and
can be seen in Figures 4.3, 4.4, 4.5, and 4.6.

These plots indicate a maximum power of approximately 2000 W. The power of the model in expansion is slightly better than that of the model in compression at lower tank pressures. This is due to the long low side valve opening causing the expansion cycle to be less constrained by valve open times than the compression cycle. This power gain is at a loss of efficiency, however, as the compressor achieves approximately 60% efficiency and the expander approximately 70% efficiency at maximum speed and pressure. However, as the speed and pressure decrease, the overall efficiency increases for the expander, as seen in Figures 4.3 and 4.4. This follows from the above observations, that the efficiency and power are inversely related in the system. Therefore, during vehicle operation, there has to be some control balancing the power and efficiency of the system.
Figure 4.4: Mechanical efficiency of tuned valving in expansion mode

Figure 4.5: Power of tuned valving in compression mode
4.3 Engine start-stop System Optimization

The engine start-stop system was designed to be a small, simple system to add on to any vehicle with only sizing modifications. Based on previously noted drive cycle analysis, the system can operate solely based on braking power. Therefore, engine load optimization work was not incorporated in this design. The tank was sized to have 3 full start-stop charges in it at all times. This was to provide a buffer in case the first start did not work or if there was not enough braking energy dissipated to fully prepare for another ESS occurrence. This meant the tank needed to be 14.9 gallons to allow for 3 full starts at 0.5 seconds per start at the air motor’s 87 psi working pressure. In a best case scenario, this tank would have 5 starts at 0.3 seconds. Additionally in this design, as mentioned above, there would be no use for the IVT, but rather only a CVT. This is because there is no energy flow direction control needed.
The compressor/expander valving for this system was optimized solely for compression by removing the changes in timing made to account for volumetric efficiency losses during the expansion cycle. This was because there was no energy outflow through the compressor/expander, as the energy was reintroduced using the air motor. This led to a system that did not have the same RTE as the previously defined system, but outperformed the previously tuned system in compression efficiency. This system was based off ideal compressor design, as detailed by [38]. The system changed by opening the low side valve later in the down stroke, once the residual air in the chamber at pressure after top dead center had fully expanded to atmospheric levels. This occurred at just after halfway down the stroke identical to the high side valve, rather than the original timing of opening immediately after top dead center. Therefore, the high side valve and low side valve dimensions and angle were set as equal, with the high side opening before top dead center and the low side before bottom dead center. This design was optimized by `fmincon` with an objective function to maximize

\[
Eff_{\text{compressor}} = \frac{\text{Power}_{\text{stored}}}{\text{Power}_{\text{in}}},
\]

(4.12) comparing the power applied to the compressor to the power stored in the air tank, but the optimization confirmed the previous system design for valve sizing, confirming the recommendations by Kerr. No changes were made by the optimization to the resized valving.

Figures 4.7 and 4.8 detail the efficiency and power capabilities of this tuned system.

Figure 4.8 indicates power to be consistent with the tank pressure and shaft speeds of the previously tuned system. However, Figure 4.7 indicates greater efficiency at higher speeds and pressures than in the previously optimized case. This
Figure 4.7: Mechanical efficiency of tuned valving in compression mode

Figure 4.8: Power of tuned valving in compression mode
An optimized compressor design was then implemented in the drive schedule simulator. The control for the model involved charging the system at maximum power if the power was provided by the brakes and then stopping the charge when the system was full. If the system was not able to charge at maximum power, the system would charge at a lesser power level and speed. Modulation in system speed was assumed to be handled through use of the CVT to allow for the power to be delivered to the air tank at the maximum allowable speed for that power level. The vehicle weight was updated to include the CVT, a 15 gallon air tank, the air motor, and the compressor. This added an approximate 150 pounds to the vehicle due to the 50 pound air motor and equipment, the 50 pound compressor/expander and valve head, the 20 pound tank, and an estimated 30 pounds of additional infrastructure. This was appropriately updated through the payload variable in the simulator. The system was estimated to consume approximately 3.5 cubic feet. The majority of this size is the volume of the 15 gallon air tank, which consumes 2.01 cubic feet of space. The compressor and valve head footprint along with the air motor with the appropriate valving consumes approximately 0.5 cubic feet of space, and the CVT is estimated to take up a maximum of 1 cubic foot. This system should comfortably fit in an engine bay, as it can be rearranged to fit around the engine. The only potential challenge would be placement of the air tank.

As seen in Figures 4.9 and 4.10, the system was adequate to charge the air tank during braking and start an engine again for the NYCC cycle. This cycle had stops with the lowest braking energy available per stop and therefore, was chosen for the validation. Figure 4.9 shows the system from the beginning of the chapter run over the NYCC cycle. Here, the system is large enough to rapidly charge the air tank for all of the ESS stops. Figure 4.10 displays the sized compressor/expander performance. Here, the compressor takes more time to charge the air tank, as...
indicated by longer charging times. This compressor was sized to be \( \frac{1}{6} \) the size of the original compressor volume and still comfortably met the performance requirements. This compressor was oversized to be capable of 1.5 times the power of the minimum sized unit necessary to intake enough energy to ensure start-stop for all of the NYCC cycle. While the air compressor was oversized for the amount of energy necessary, this was deemed appropriate to account for even shorter braking cycles that could exist during a consumer’s driving experience. The overall fuel economy improvement was the same over all cycles as estimated in Chapter 3, as this metric is only dependent on the idle time and not system efficiency as long as the air tank is adequately charged for every start-stop. The volume of the system is small enough to be fit in a vehicle, and the weight does not cause significant negative effects on the efficiency that are not offset by the ESS.

![Figure 4.9: Initially sized CAES air tank charge over NYCC cycle](image)

This confirms the conclusions of the pilot study in Chapter 3 for the ESS system. This conclusion is that there are achievable and desirable benefits of implementation
of an ESS system for cycles with high amounts of braking. Therefore, a CAES hybrid system could be implemented on a car that currently uses a battery hybrid system to perform engine start-stop. The optimization allowed for a smaller, more efficient compressor/expander system to be implemented, taking up less vehicle volume and adding less weight to the total vehicle payload.

4.4 Launch Assist System Optimization

The launch assist CAES hybrid system was designed to be larger than the ESS system and capable of inputting a reasonable amount of power to propel the vehicle. Additionally, this system had to store more energy to provide this power and have a more complex control strategy to dictate how the power would be delivered. To handle energy input, both braking energy and engine load optimization were considered separately to charge the system, although many of their design objectives are similar. The braking only system will be referred to as BOS, and the engine op-
The optimization system will be referred to as EOS. For power delivery to the drivetrain, a basic controller was used to meter energy back to the vehicle for use. The compressor/expander was chosen to have the best possible RTE and therefore, incorporated the valving designed by the \textit{fmincon} optimization. The tank was sized to be able to handle draw for 7 seconds. This was chosen to meet the average discharge time for both the UDDS and the NYCC cycles and to be able to assist in the US06 and HWYFET cycles. Table 4.2 displays the average time between accelerations in each cycle that reach above the engine optimization threshold of 40% of maximum power.

**TABLE 4.2**

<table>
<thead>
<tr>
<th>Cycle</th>
<th>Time [sec]</th>
</tr>
</thead>
<tbody>
<tr>
<td>UDDS</td>
<td>6.7</td>
</tr>
<tr>
<td>HWYFET</td>
<td>20.5</td>
</tr>
<tr>
<td>NYCC</td>
<td>4.9</td>
</tr>
<tr>
<td>US06</td>
<td>9.6</td>
</tr>
</tbody>
</table>

The compressor/expander was initially sized to be able to provide approximately one standard deviation’s worth of power, as calculated in Tables 3.7 and 3.8. In order to provide this power, the Cruze tank would have to be approximately 60 gallons, and the Tahoe tank would be 160 gallons in addition to the 15 gallon allotment for ESS. This tank size was deemed unacceptable and therefore reevaluated. In order to still provide meaningful power, the tank sizes were halved to 30 and 80, respectively. Although these tanks were still large, they were deemed appropriate considering early strong battery hybrids. In these hybrids, often the trunk space
was filled with a power storage unit. These tank sizes fall into a similar size range. The tank sizing for both the braking and engine optimization cases was the same, with the rate of cycling of the system being varied between the two to account for power availability.

4.4.1 System Control

Two different controllers were required to control the different LA systems considered. The BOS used a simple controller that charged the tank with any available braking energy and would provide power to the system when it operated at over the engine’s average power per cycle. The system would stop providing power when the tank charge fell to a 2 ESS limit in order to ensure continued ESS capability. However, the control for the EOS case was more complicated. Pseudocode of the controller is included in the appendix.

The high and low thresholds for determining power flow were set by estimations from brake-specific fuel consumption (BSFC) maps. When the vehicle was operating below the low threshold of power, the system would allow for an increase in engine power to charge the CAES system. When the vehicle was above the high threshold, the CAES would operate to reintroduce the energy to partially mitigate the high engine load. When the engine was operating between the high and low power threshold, the CAES system did not operate to input or output energy. Following these power flow rules, the code updated the driving simulator’s estimate of the power consumed by use of BSFC maps, a standard of measurement of the amount of fuel used to generate power at a particular engine load point. These maps are useful in complex simulators that model engine speed and load and can generate precise estimates of the amount of fuel consumed by the engine at each point. However, by using these maps, generalized statements about fuel consumption were made based
on engine size for use in the simplified simulator. BSFC maps were obtained from an online repository [29] and used to generate general fuel consumption guidelines.

The engine sweet spot was estimated to be approximately between 40 and 60% for a Cruze-sized engine and between 35 and 65% of maximum load for a Tahoe-sized engine. These values were set as the high and low power thresholds for their respective vehicles. In addition, ratios of the BSFC values provided scaling factors to estimate the fuel consumption for operation on both the high and low side of the engine power spectrum. It was assumed that the BSFC decreased linearly from the low power operation to its sweet spot and then again linearly from the upper end of the sweet spot to the maximum vehicle power, as

\[ E_{\text{new}} = F \times E_{\text{current}} \times (L_{\text{current}}\% - L_{\text{EOS}}\%), \] (4.13)

where \( E \) is the energy needed to propel the vehicle forward, \( F \) is the corresponding vehicle factor from Table 4.3, and \( L \) is the load in percent of the maximum load. By using this formula, the extra fuel burned to load the engine or the fuel saved

<table>
<thead>
<tr>
<th></th>
<th>High Side</th>
<th>Low Side</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cruze</td>
<td>9.16</td>
<td>2.93</td>
</tr>
<tr>
<td>Tahoe</td>
<td>7.5</td>
<td>2.57</td>
</tr>
</tbody>
</table>
4.4.2 Optimization

The system as originally organized was composed of a 30 gallon tank for the Cruze and a 80 gallon tank for the Tahoe. These tanks were attached to the compressor/expander valve design as optimized at the beginning of the chapter, as well as an air motor. The weight of the system was estimated to be approximately 300 pounds for the Cruze and 450 pounds for the Tahoe. The control system as described above was implemented to control the energy reintroduction. An \texttt{fmincon} optimization was run with the valve timing set as before, varying over the range set in Table 4.1. This was run over the US06 cycle, as the power demands varied the most in this cycle. The sum of the total power reintroduced through LA was the objective function to be maximized, as the ESS capability disrupted the estimates of total system efficiency. This is due to the much more efficient air motor skewing the discharge efficiency estimate with the expander. \texttt{Fmincon} returned a slightly altered system, more optimized to higher efficiency on the expansion side. The optimized system had a slightly longer high side valve, with the same width as the nominal system but open from 4.2 degrees below the half stroke until top dead center. This aided the efficiency during expansion by allowing more air to flow to the expansion cylinder while slightly harming the efficiency during compression. The low side valve did not change from the nominal system, as this valving eliminated the possibility of inefficient compression during the upstroke. This valving is likely due to the engine operating for the majority of the cycle while below the optimal engine load point. This would lead to an excess of available energy for charging the tank and would push the system to be more efficient in its energy use in expansion during the high vehicle demand load points to maximize the tank charge. Efficiency maps of the new valve system in compression and expansion can be seen in Figures 4.11 and 4.12. This system sees lower efficiency than the nominal system in
Figure 4.11: Optimized valving efficiency in compression mode

Figure 4.12: Optimized valving efficiency in expansion mode
compression, but significantly greater expansion efficiency. The optimized system as executed over the drive cycle generated new fuel efficiencies, seen in Tables 4.4 and 4.5.

TABLE 4.4

LAUNCH ASSIST UPDATED MPG ESTIMATES FOR CRUZE

<table>
<thead>
<tr>
<th>Cycle</th>
<th>Original MPG</th>
<th>BOS MPG</th>
<th>EOS MPG</th>
</tr>
</thead>
<tbody>
<tr>
<td>UDDS</td>
<td>32.9</td>
<td>44.2</td>
<td>48.2</td>
</tr>
<tr>
<td>HWYFET</td>
<td>38.1</td>
<td>39.4</td>
<td>44.1</td>
</tr>
<tr>
<td>NYCC</td>
<td>26.0</td>
<td>45.9</td>
<td>51.2</td>
</tr>
<tr>
<td>US06</td>
<td>23.2</td>
<td>27.7</td>
<td>47.3</td>
</tr>
</tbody>
</table>

TABLE 4.5

LAUNCH ASSIST UPDATED MPG ESTIMATES FOR TAHOE

<table>
<thead>
<tr>
<th>Cycle</th>
<th>Original MPG</th>
<th>BOS MPG</th>
<th>EOS MPG</th>
</tr>
</thead>
<tbody>
<tr>
<td>UDDS</td>
<td>18.2</td>
<td>24.9</td>
<td>28.5</td>
</tr>
<tr>
<td>HWYFET</td>
<td>21.4</td>
<td>22.5</td>
<td>24.1</td>
</tr>
<tr>
<td>NYCC</td>
<td>14.3</td>
<td>25.6</td>
<td>30.0</td>
</tr>
<tr>
<td>US06</td>
<td>13.0</td>
<td>15.8</td>
<td>29.5</td>
</tr>
</tbody>
</table>

These values are a significant improvement over the base values and are only slightly less than the original values estimated in Figure 3.8. These results validate the use of an LA system in any vehicle in terms of fuel economy benefits. However, this benefit comes at a cost. With this size LA system in place, the vehicle loses a large amount of interior space. This space corresponds to approximately 7 cubic feet for the Cruze and 13 cubic feet for Tahoe. This is a serious limiting factor for any sizeable CAES LA hybrid system. Because the energy density is so low with low
pressure air, the storage tank must be very large to store a meaningful amount of energy. While the CAES LA works, implementation and sales to customers would prove more troublesome while competing with smaller battery hybrids.

The benchtop system required significant redesign in order to operate efficiently. Therefore, the nominal system was designed to incorporate reasonable updates as suggested by Brown [42], as well as changes to the compressor/expander and rotary valve to account for the increased operating pressure. This nominal system allowed for further optimization in order to better suit the needs of vehicles in particular drive schedules. For the ESS system, a compressor-efficient design was followed in order to increase the energy available for the air motor. This led to a small system that was capable of producing reliable ESS functionality and decent fuel economy benefits for drive schedules with regular vehicle stops. The LA system was optimized for greater expander efficiency and therefore, did not compress as well as the ESS-only cycle. While the LA system was better at improving fuel economy for all cycles, the size of the system was significant and would be challenging to market to a consumer.
5.1 Conclusions

Vehicle hybridization is a strategy used to help address the shortcomings of modern ICE-powered vehicles. These systems work by recapturing wasted energy and reintroducing that energy later in an attempt to increase overall fuel efficiency. Compressed air energy storage provides a simple, low cost alternative to electric hybrid systems. These hybrids avoid the downsides of the battery systems currently in place, providing a safer alternative to the costly systems currently available to consumers. A driving simulator was created and validated in order to test these systems. This simulator provided computationally inexpensive estimations of vehicular performance while still achieving reasonably accurate results. These results were accurate within 5% of the values provided by the governmentally regulated testing for the vehicles chosen for investigation. Analysis of vehicle performance over EPA test schedules indicated benefits for two methods of energy reintroduction: launch assist and engine start-stop. ESS was capable of providing benefits for cycles with high numbers of vehicle stop points by eliminating extended idling at these stops. LA was found to be beneficial in almost all of the cycles, as it provided opportunities for power reintroduction independent of vehicle stop occurrences. The power necessary to drive these cycles could also be recovered using two methods. Recapturing braking energy was initially pursued, as the energy here is otherwise
dissipated in the form of heat as the vehicle decelerates. This method was found to recover energy sufficient for the needs of an ESS system or a small LA system. The second involves engine load optimization in order to allow the engine to operate at a more efficient load point and stores the excess energy for reintroduction later.

This initial study provided a clear picture of two different potential systems for vehicle hybridization. The first system to emerge is a pure ESS system that operates only using brake power as a source and an air motor to start the engine. This system was optimized for compression due to the role of expansion being shifted to the air motor. The second system is a LA/ESS capable system that charges using braking and engine load optimization. This system was optimized for efficient energy storage and reintroduction, as high efficiency was required in order to operate the compressor/expander for both purposes. This second system is larger and requires more vehicle alteration than the first, but provides the benefit of greater fuel economy improvements.

In the system optimization, the CAES hybrid system was sized for the two energy delivery pathways. The ESS system operated solely by collecting energy from braking and using that to restart the engine after a vehicle stop. This system was successful with a small implementation that could reasonably fit inside either the engine bay of a car or in the spare wheel well, a common hybrid system placement used today. The benefits from this system justify its implementation in vehicles that will undergo lots of start-stop driving, such as in a city environment. This system should be competitive in vehicles with battery ESS hybrids that are currently on the consumer market. CAES LA hybrids are also beneficial in terms of fuel economy gains. However, the systems as sized are significantly larger than current competitive hybrid systems. The size of these systems would serve as a deterrent to any serious implementation in the consumer marketplace. CAES hybridization would see the
greatest benefits for small vehicles with ESS system implementation.

5.2 Future Work

In order to provide higher fidelity estimations, a more rigorous driving simulator should be pursued. The rudimentary simulator used for the purposes of this study does not take into account engine load dynamics and therefore cannot appropriately define the varying fuel consumption of different engine load points. With such detail, a better picture of the effects of engine load optimization as well as load adjustment from the hybrid system could be modeled.

Gradient-based optimizations such as \texttt{fmincon} suffer from local minima and often do not successfully find the globally optimal solution. Full investigation of the system design space could lead to a better solution that exists far outside the expected system designed in this paper.

Additionally, the controller used for the LA hybrids systems is very simplistic. Use of a two-part controller to optimize the loading on both the engine and the hybrid system would be ideal and has been promising in other implementations [9]. An additional method for system control would use a dynamic programming algorithm. Dynamic programming methods have been especially useful in determining solutions that maximize performance over vehicle drive cycles.

Implementation of better control would most likely lead not only to a more efficient system, but perhaps to a smaller system that would more intelligently cycle the energy from waste to reintroduction.

The largest losses in efficiency for the CAES system occur in the compressor/expander assembly as it operates over a wide pressure range. Investigation into alternate, more efficient compressor systems could lead to efficiency gains for the overall CAES system. Additionally, the rotary valve assembly used to approximate
variable valve timing is a system with very limited history, and future work should be conducted to explore additional benefits and issues with the implementation of rotary valves.

Additional work should be completed to validate the IVT concept as proposed by Brown [42] for system implementation. Validating this IVT is important to the system model for the purposes of power control. This IVT design is the final step in the process of constructing the newly designed system for a full concept validation. This validation is necessary to confirm the physical responses of the heavily modified computer model. With this confirmation of the system model, implementation of the CAES system could occur for a vehicle to provide real world testing over the previously used driving schedules.
A.1 Pseudocode for LA controller

\[
\text{if Velocity} \neq 0 \\
\text{if tank} < \text{tank -max} \times 0.8 \\
\text{if wheel-demand} < \text{threshold-low} \\
\text{activate compressor} \\
\text{end} \\
\text{end} \\
\text{end} \\
\text{if tank} < \text{tank-full} \\
\text{if braking} \\
\text{activate compressor} \\
\text{end} \\
\text{if wheel-demand} > \text{threshold-high} \\
\text{if tank} > 2 \text{ ESS charge} \\
\text{Run expander} \\
\text{end} \\
\text{end} \\
\text{if ESS point} \\
\text{execute air motor} \\
\text{end}
\]
BIBLIOGRAPHY


